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KENT'S
MECHANICAL ENGINEERS' HANDBOOK

Power

Founded by the late

WILLIAM KENT, M.E., Sc.D.

ELEVENTH EDITION, REWRITTEN

BY

ROBERT THURSTON KENT, M.E.
Editor-in-Chief

With the Collaboration of a Staff of Specialists

WILEY ENGINEERING
HANDBOOK SERIES

NEW YORK : JOHN WILEY & SONS, INC.
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PUBLISHER'S PREFACE

In making plans for new editions of our handbooks in mechanical engineering and in electrical engineering, it soon became clear that engineering science and practice had developed to such an extent that handbooks were growing beyond all practical bounds. They had become both bulky and inconvenient and contained much duplicated material. In order to solve the problems presented by these conditions, the editors of our various handbooks were asked to serve as an advisory editorial board.

This board recommended, first, that the fundamental material underlying all engineering be published in a separate volume, and, second, that the existing handbooks as they are revised be issued in several volumes containing material closely related to the specialized branches of engineering. As a result of these recommendations, the Wiley Engineering Handbook Series has been initiated, which in the beginning will comprise the following: Eshbach's "Handbook of Engineering Fundamentals"; Kent's "Mechanical Engineers' Handbook" in two volumes, viz., "Power" and "Design and Shop Practice"; Pender's "Electrical Engineers' Handbook" in two volumes, viz., "Electric Power" and "Communication and Electronics."

This division has also made it possible to devote more space to the various topics so that the entire new series of handbooks contains more complete information on all topics than heretofore has been possible. It is our hope that this new plan will give engineers information that is more useful, more complete, and in more convenient form.

JOHN WILEY & SONS, INC.

PREFACE TO THE ELEVENTH EDITION,

July 1936.

In the preparation of the eleventh edition of this book, the principle that has governed the ten previous editions has constantly been kept in mind—namely, that it must represent the *practice* of mechanical engineering. The changes that have taken place in engineering in the 13 years that have elapsed since the 10th edition was published have been many and great. To conform to these changes, a complete rewriting of the book has been necessary.

Advantage has been taken of this to change the arrangement, form and typography, to make a book that would better fill its field and be more usable. The larger page permits the use of larger type and illustrations. The separation of the book into volumes, each covering more or less closely related subjects, makes it more convenient for the engineer interested in only a limited field.

A further radical change is the inclusion in a separate volume under the editorship of O. W. Eshbach of the discussion of the fundamental sciences that underlie all engineering, thus permitting more space to be given to the practice in the other volumes.

It is believed that these changes will meet with the approval of the engineering profession, and that the book will continue to render the high degree of service that has characterized it for the past 42 years.

The editor-in-chief expresses his thanks to the collaborators whose co-operation has rendered this work possible. He also desires to thank Dr. Charles E. Lucke for many helpful suggestions. Thanks also are due to the many industrial organizations whose criticism and compilation of data have been of material assistance.

Robert Thurston Kent.

ABSTRACT FROM PREFACE TO THE FIRST EDITION, 1895.

More than twenty years ago the author began to follow the advice given by Nystrom: "Every engineer should make his own pocket-book, as he proceeds in study and practice, to suit his particular business." The manuscript pocket-book thus begun, however, soon gave place to more modern means for disposing of the accumulation of engineering facts and figures, viz., the index rerum, the scrap-book, the collection of indexed envelopes, portfolios and boxes, the card catalogue, etc. Four years ago, at the request of the publishers, the labor was begun of selecting from this accumulated mass such matter as pertained to mechanical engineering, and of condensing, digesting, and arranging it in form for publication. In addition to this, a careful examination was made of the transactions of engineering societies, and of the most important recent works on mechanical engineering in order to fill gaps that might be left in the original collection, and insure that no important facts had been overlooked.

Some ideas have been kept in mind during the preparation of the Pocket-book that will, it is believed, cause it to differ from other works of its class. In the first place it was considered that the field of mechanical engineering was so great, and the literature of the subject so vast, that as little space as possible should be given to subjects which especially belong to civil engineering.

Another idea prominently kept in view by the author has been that he would not assume the position of an "authority" in giving rules and formulæ for designing, but only that of compiler, giving not only the name of the originator of the rule, where it was known, but also the volume and page from which it was taken, so that its derivation may be traced when desired. When different formulæ for the same problem have been found they have been given in contrast, and in many cases examples have been calculated by each to show the difference between them. In some cases these differences are quite remarkable. Occasionally the study of these differences has led to the author's devising a new formula, in which case the derivation of the formula is given.

Much attention has been paid to the abstracting of data of experiments from recent periodical literature, and numerous references to other data are given. In this respect the present work will be found to differ from other Pocket-books.

William Kent.

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SYMBOLS AND ABBREVIATIONS

Greek Letters

Α	α	Alpha	Η	η	Eta	Ν	ν	Nu	Τ	τ	Tau
Β	β	Beta	Θ	ϑ θ	Theta	Ξ	ξ	Xi	Υ	υ	Upsilon
Γ	γ	Gamma	Ι	ι	Iota	Ο	ο	Omicron	Φ	ϕ	Phi
Δ	δ	Delta	Κ	κ	Kappa	Π	π	Pi	Χ	χ	Chi
Ε	ϵ	Epsilon	Λ	λ	Lambda	Ρ	ρ	Rho	Ψ	ψ	Psi
Ζ	ζ	Zeta	Μ	μ	Mu	Σ	σ ς	Sigma	Ω	ω	Omega

Mathematical Signs and Abbreviations

$+$ plus (addition).
$+$ positive.
$-$ minus (subtraction).
$-$ negative.
$\pm$ plus or minus.
$\mp$ minus or plus.
$=$ equals.
$\geqq$ equals or greater than.
$\leqq$ equals or is less than.
$\approx$ approximately equals.
$\times$ multiplied by.
ab or $a.b = a \times b$.
$\div$ divided by.
/ divided by.
$\frac{a}{b} = a/b = a \div b$. $15/16 = \frac{15}{16}$.
$0.2 = \frac{2}{10}$; $0.002 = \frac{2}{1000}$.
$\sqrt{\ }$ square root.
$\sqrt[3]{\ }$ cube root.
$\sqrt[4]{\ }$ 4th root.
: is to, : : so is, : to (proportion).
2 : 4 : : 3 : 6, 2 is to 4 as 3 is to 6.
: ratio; divided by.
2 : 4, ratio of 2 to 4 = 2/4.
$>$ greater than.
$<$ less than.
° degrees, arc or thermometer.
′ minutes or feet.
″ seconds or inches.
′ ″ ‴ accents to distinguish letters, as a', a'', a'''.
a_1, a_2, a_3, a_b, a_c, read a sub 1, a sub b, etc.
( ) [ ] { } —— parentheses, brackets, braces, vinculum; denoting that the numbers enclosed are $\overline{\text{to be}}$ taken together; as, $(a + b)c = 4 + 3 \times 5 = 35$.
a^2, a^3, a squared, a cubed.
a^n, a raised to the nth power.
$a^{2/3} = \sqrt[3]{a^2}$, $a^{3/2} =$.
$a^{-1} = \frac{1}{a}$, $a^{-2} = \frac{1}{a^2}$.
$10^9 = 10$ to the 9th power $= 1{,}000{,}000{,}000$.
$\sin a =$ the sine of a.
$\sin^{-1} a =$ the arc whose sine is a.
$\sin a^{-1} = \frac{1}{\sin a}$.
log = logarithm.
$\log_e$ or hyp log = hyperbolic logarithm.
% per cent.
$\angle$ angle.
$\triangle$ triangle.
sin, sine.
cos, cosine.
tan, tangent.
sec, secant.
versin, versed sine.
cot, cotangent.
cosec, cosecant.
covers, co-versed sine.

In Algebra, the first letters of the alphabet, a, b, c, d, etc., are generally used to denote known quantities, and the last letters, w, x, y, z, etc., unknown quantities.

Abbreviations and Symbols commonly used.

d, differential (in calculus).
$\int$, integral (in calculus).
, integral between limits a and b.
Δ, delta, difference.
Σ, sigma, sign of summation.
π, pi, ratio of circumference of circle to diameter = 3.14159.
g, acceleration due to gravity = 32.16 ft. per second per second.

Abbreviations

abs. . . . . absolute
A.C. . . . . alternating-current
A.I.E.E. . . . . . American Institute of Electrical Engineers
A.I.M.E. . . . . American Institute of Mining and Metallurgical Engineers
amp. . . . . amperes
A.I. and S.I. . American Iron and Steel Institute
approx. . . approximate
A.P.I. . . . . . . American Petroleum Institute
A.R.A. . . . . . . American Railway Association
A.R.E.A. . . . . American Railway Engineering Association
A.S.A . . . . . . American Standards Association
A.S.C.E. . . . . American Society of Civil Engineers

SYMBOLS AND ABBREVIATIONS

A.S.H.V.E.	American Society of Heating and Ventilating Engineers
A.S.M.E.	American Society of Mechanical Engineers
A.S.R.E.	American Society of Refrigerating Engineers
A.S.S.T.	American Society for Steel Treating
A.S.T.M.	American Society for Testing Materials
atmos.	atmosphere
avoir.	avoirdupois
A.W.G.	American wire gage
B., *b.*	breadth
B. & S.	Brown & Sharpe wire gage
bar.	barometer, barometric
bbl.	barrel
Be.	Baumé
B.G.	Birmingham gage (tube and sheet)
B.Hp.	brake horsepower
b.m.e.p.	brake mean effective pressure
Bur. of Std.	Bureau of Standards
B.t.u.	British thermal unit(s)
Bu.	bushel(s)
Bull.	bulletin
B.W.G.	Birmingham wire gage
C	one hundred
C., cent.	centigrade
c.c.	cubic centimeter(s)
c.f.m.	cubic feet per minute
c.g.s.	centimeter-gram-second
c.m.	circular mil
cal.	calorie(s)
cg.	centigram(s)
cir	circular
cl	centiliter(s)
col	column
cu. cm.	cubic centimeter(s)
cu. ft.	cubic feet
cu. in.	cubic inch(es)
cu. m.	cubic meter(s)
cu. yd.	cubic yard(s)
cyl	cylinder
D., *d.*	diameter, depth
D.C.	direct current
def.	definition
deg.	degree(s)
diam	diameter
e	base of Naperian system of logarithms
E.E.I.	Edison Electrical Institute
E. Hp.	electric horsepower
e.m.f.	electromotive force
eff.	efficiency
evap.	evaporation
ext.	external
F.	Fahrenheit
F	force
f	coefficient of friction
f.Hp	friction horsepower
f.p.m.	feet per minute
ft.	foot, feet
ft.-lb.	foot-pound(s)
g	acceleration due to gravity
g.p.m.	gallons per minute
gal.	gallon(s)
gm.	gram(s)
gm.-cal.	gram-calorie(s)
gr.	grain(s)
H, *h*	height
H.P.C.N.A.	Heating Piping Contractors' National Association
h.p.	high pressure
Hg.	mercury
hor.	horizontal
Hp.	horsepower
Hp.-hr.	ho
hr.	hour(s)
hyd.	hydraulic
hyp.	hyperbolic
hyp. log.	hyperbolic or Napierian rithm
I	current, electric; moment of inertia
I.Hp.	indicated horsepower
i.p.	intermediate pressure
Imp.	Imperial
in.	inch(es)
in.-lb.	inch-pound(s)
Inst. C.E.	Institute of Civil Engineers
Inst. M.E.	Institute of Mechanical Engineers
int.	internal
J	mechanical equivalent of heat
j	joule(s)
*k*B	1000 B.t.u.
kg.	kilogram(s)
kg.-cal.	kilogram-calorie(s)
kg.-m	kilogram-meter(s)
kl.	kiloliter(s)
km	kilometer(s)
kva	kilovolt-ampere(s)
kw	kilowatt(s)
kw.-hr.	kilowatt-hour(s)
L, *l*	length
l	liter
l.p.	low pressure
lat	latitude
lb.	pound(s)
lin	linear
log	logarithm
$\log_e$	Naperian logarithm
$\log_{10}$	common logarithm
M	one thousand
m	meter(s)
m.e.p.	mean effective pressure
m.g.d.	million gallons per day
m.i.p.	mean indicated pressure
m.-kg.	meter-kilogram
m.m.f.	magnetomotive force
m.p.h.	miles per hour
max	maximum
mi	mile(s)
min.	minimum; minute
mm.	millimeter
M.S.S.	Manufacturers' Standardization Society of Valve and Fittings Industry
N	number (in mathematical tables)
N.A.C.A.	National Advisory Council on Aeronautics
nat	natural
N.E.L.A.	National Electric Light Association (now Edison Electrical Institute)
N.E.M.A.	National Electrical Manufacturers Association
No.	number
O.D.	outside diameter
oz.	ounce(s)
P	pressure
p., pp.	page, pages
p.f.	power factor
Proc.	Proceedings
pt	pint(s)
Q	quantity or volume
qt	quart(s)

R, *r*.......... radius
r.m.s......... square root of mean square
r.p.m., or rev. per min.... revolutions per minute
Ry.......... railway
S.A.E........ Society of Automotive Engineers
S.W.G....... Standard (British) wire gage
sat.......... saturated
sec.......... second(s)
sp........... specific
sp. gr........ specific gravity
sp. ht........ specific heat
sq........... square
sq. cm....... square centimeter(s)
sq. ft........ square foot, square feet
sq. in........ square inch(es)
sq. km....... square kilometer(s)
std.......... standard
T.S.......... tensile strength
temp........ temperature
Trans........ Transactions
U.S.......... United States
U.S.S.G...... United States Standard gage
ult.......... ultimate
V, *v*.......... velocity; volume
vel.......... velocity
vol.......... volume
vs........... versus
W, *w*......... weight
W. & M..... Washburn and Moen wire gage
W.I......... wrought iron
wt.......... weight
yd.......... yard(s)
y.p.......... yield point
yr.......... year(s)

Periodicals and Publications

Aerologist, Chicago.
Am. Gas-light Jour., American Gas Journal, New York.
Am. Mach., American Machinist, New York.
Annales des Mines, Annales des Mines, Paris.
Archiv. f. Warme und Damf, Archiv für wärmewirtschaft und damfkesselwesen, Berlin.
A.S.H.V.E. *Guide*, American Society of Heating and Ventilating Engineers Guide, New York.
Auto. Engr., Automobile Engineer, London.
Auto. Ind., Automotive Industries, New York.
Aviation, Aviation, New York.
Blast Furnace and Steel Plant, Blast Furnace and Steel Plant, Pittsburgh.
Brown Boveri *Review*, Brown Boveri Review, Baden, Switzerland.
Bull. Am. Locomotive, Bulletin, American Locomotive Co., Philadelphia.
Bull., Am. Ry. Eng. and Maint. of Way Assoc., Bulletin American Railway and Maintenance of Way Association, Chicago.
Bull. Cornell Univ. Engg. Expt. Station, Bulletin Cornell University Engineering Experiment Station, Ithaca, N. Y.
Bull. Engg. Expt. Sta. Univ. of Ill., Bulletin Engineering Experiment Station, University of Illinois, Urbana, Ill.
Bull. Penna. State College, Bulletin Pennsylvania State College, State College, Pa.
Bull. U. S. Dept. Agriculture, Bulletin, U. S. Department of Agriculture, Washington.
Bull., U. S. Geol. Survey, Bulletin United States Geological Survey, Washington, D.C.
Bull. Univ. of Ill., Bulletin University of Illinois, Urbana, Ill.
Bull. Univ. of Minn., Bulletin of University of Minnesota, Minneapolis, Minn.
Bull. Univ. of Texas, Bulletin University of Texas.
Bull. Univ. of Wis., Bulletin University of Wisconsin, Madison, Wis.
Bulletin Canadian Dept. of Mines, Bulletin Canadian Department of Mines, Ottawa, Canada.
Bulletin Iowa State College, Bulletin Iowa State College, Ames, Iowa.
Bulletin Louisiana Agricultural Expt. Station, Bulletin Louisiana Agricultural Experiment Station, Baton Rouge, La.
Bulletin U. S. Bureau of Mines, Bulletin United States Bureau of Mines, Washington.
Bur. Stds. Circular, Bureau of Standards Circular, Washington.
Chem. Soc. Jour., Chemical Society Journal, London.
Coal Age, Coal Age, New York.
Combustion, Combustion, New York.
Comp. Air. Mag., Compressed Air Magazine, Phillipsburg, N. J.
Comptes Rendu, Comptes Rendu Hebdomadaires des seances de l'Academie des Sciences, Paris
Compressed Air, Compressed Air, now Comp. Air Magazine.
Diesel Power, Diesel Power, New York.
Elec. Eng., Electrical Engineer, London.
Elec. Jour., Electrical Journal, Pittsburgh.
Elec. Wld., Electrical World, New York.
Elektricitats Wirt, Elektrizitätwirtschaft, Berlin.
Ency. Brit., Encyclopædia Britannica, New York,
Eng. Digest, Engineering Digest.*
Eng. News, Engineering News, New York.
Engg., Engineering, London.
Eng. Mechanics, En ineering Mechanics.*
Engr., The Engineer, London.
Gen. Elec. Rev., General Electric Review, Schenectady, N. Y.
Heat and Vent. Mag., Heating and Ventilating Magazine, New York.
Heating, Piping and Air Conditioning Magazine, Heating, Piping and Air Conditioning Magazine, Chicago.
Ice and Cold Storage, Ice and Cold Storage, London.
Ice and Refrigeration, Ice and Refrigeration, Chicago.
Ind. and Engg. Chem., Industrial and Engineering Chemistry, New York.
J.C.I.W., Journal Charcoal Iron Workers Association.*
Jour. A.S.H.V.E., Journal American Society of Heating and Ventilating Engineers, New York.
Jour. Am. Ceramic Soc., Journal American Ceramic Society, Columbus, Ohio.
Jour. Chemical Society, Journal Chemical Society, London.
Jour. Franklin Inst., Journal Franklin Institute, Philadelphia.
Jour. Inst. Metals, Journal Institute of Metals, London.

* Out of Print.

Jour. Royal Aeronautical Soc., Journal Royal Aeronautical Society, London.

Jour. S.A.E., Journal Society of Automotive Engineers, New York.

Mech. Engg., Mechanical Engineering, New York.

Mech. Sect. British Assoc. for Advancement of Science, Mechanical Section British Association for Advancement of Science, London.

Mitteil u. Forsch, Mitteilungen über Forschungsarbeiten, Berlin.

New Zealand Jour. of Science and Technology, New Zealand Journal of Science and Technology, Wellington, N. Z.

Power, Power, New York.

Power Plant Engg., Power Plant Engineering, Chicago.

Proc. Am. Gas Inst., Proceedings American Gas Institute, New York.

Proc. Am. Gas Light Assoc., Proceedings American Gas-Light Association, New York.

Proc. Am. Ry. M.M. Assoc., *Proc. Amer. Ry. Master Mech. Assoc.*, Proceedings American Railway Master Mechanics Association, Chicago.

Proc. Am. Water Wks. Assoc., Proceedings American Water Works Association.

Proc. Engrs. Club of Phila., Proceedings Engineers Club of Philadelphia, Philadelphia.

Proc. Engrs. Soc. West. Penna., Proceedings Engineers Society of Western Pennsylvania, Pittsburgh.

Proc. Inst. C.E., *Proc.* Inst. Civ. Eng., Proceedings Institute Civil Engineers, London.

Proc. Inst. M.E., Proceedings Institute of Mechanical Engineers, London.

Recommended Practice (Am. Ry. Assoc.), Recommended Practice of American Railway Association.

Refrig. Engg., Refrigerating Engineering, New York.

Refrig. Wld., Refrigerating World, New York.

Revue de Mecanique, Revue de Mecanique, Paris.

Ry. Age, Railway Age, New York.

Ry. Age-Gazette, Railway Age-Gazette, New York.

Ry. Engr., Railway Engineer, London.

Ry. Mech. Engr., Railway Mechanical Engineer, New York.

Southern Power Jour., Southern Power Journal, Atlanta.

The Engr., The Engineer, London.

The Shipbuilder, The Shipbuilder, London.

Trans. A.I.M.E., Transactions American Institute of Mining & Metallurgical Engineers, New York.

Trans. A.P.I., Transactions American Petroleum Institute, New York.

Trans. A.S.H.V.E., Transactions American Society of Heating and Ventilating Engineers, New York.

Trans. A.S.M.E., Transactions American Society of Mechanical Engineers, New York.

Trans. A.S.R.E., Transactions American Society of Refrigerating Engineers, New York.

Trans. Am. Elect. Chem. Soc., Transactions American Electrochemical Society, New York.

Trans. Am. Ry. Master Mechanics Assoc., Transactions American Railway Master Mechanics Association, Chicago.

Trans. Chem. Soc., Transactions Chemical Society, London.

Trans. Int. Engg. Congress, Transactions International Engineering Congress.

Trans. So. African Inst. of Engrs., Transactions South African Institute of E
g

Trans. Soc. Nav. Arch. and Marine
Transactions Society of Nav
Marine Engineers, New York.

Turbines N.E.L.A., *Turbines*, N.E.L.A. and E.E.I., Turbines, National Electric Light Association (now Edison Electrical Institute).

U. S. Bureau of Standards Journal of Research, United States Bureau of Standards Journal of Research, Washington.

Van Nostrands Magazine,* Van Nostrands Engineering Magazine.

Yearbook, Am. I. and S. Inst., Yearbook, American Iron and Steel Institute, New York.

Zeits f. ang. Math. u. Mech., Berlin

Zeit. für Physikal. Chemie, Zeitschrift für Physikalische Chemie, Leipzig.

Zeit, V.D.I., Zeitschrift des Vereines deutscher Ingenieure, Berlin.

* Out of Print.

This book is divided into sections, each section carrying its independent sequence of page numbers. For example, 3–15 indicates Section 3, page 15.

Section 1

AIR

COMPRESSED AIR

By Robert Peele

FANS AND BLOWERS

By Robert Thurston Kent

TABLE OF CONTENTS

AIR

1. PROPERTIES OF AIR

Air is a mechanical mixture of the gases oxygen and nitrogen, with about 1% by volume of argon. Atmospheric air of ordinary purity contains about 0.04% of carbon dioxide. The composition of air according to various authorities is given in Table 1.

Table 1.—Composition of Air, According to Various Authorities

	By Volume			By Weight		
	N	O	Ar	N	O	Ar
1	79.3	20.7	....	77	23	....
2	79.09	20.91	....	76.85	23.15	....
3	78.122	20.941	0.937	75.539	23.024	1.437
4	78.06	21	0.94	75.5	23.2	1.3

(1) Values formerly given in books on physics. (2) Average results of several determinations, Hempel's Gas Analysis. (3) Sir Wm. Ramsay, *Bull. U. S. Geol. Survey*, No. 330. (4) A. Leduc, *Comptes Rendus*, 1896, *Jour. Franklin Inst.*, Jan., 1898. Leduc gives for the density of oxygen relatively to air 1.10523; for nitrogen 0.9671; for argon, 1.376.

The weight of pure air at 32° F. and a barometric pressure of 29.92 in. of mercury, or 14.6963 lb. per sq. in., or 2116.3 lb. per sq. ft., = 0.080728 lb. per cu. ft.; volume of 1 lb. = 12.387 cu. ft. At any other temperature and barometric pressure its weight in lb. per cubic foot is $W = \frac{1.3253 \times B}{459.6 + T}$, where B = height of the barometer, T = temperature, deg., F. and 1.3253 = weight in lb. of 459.6 cu. ft. of air at 0° F. and 1 in. barometric pressure. Air expands 1/494.6 of its volume at 32° F. for every increase of 1° F., and its volume varies inversely as the pressure.

Table 2.—Conversion Table for Air Pressures

	Lb. per sq. ft.	In. of Water	Oz. per sq. in.	Ft. of Water	In. of Mercury	Lb. per	Ft. of Air at 62° F.*	$V = \sqrt{2g}$ ft. per sec.†
1 lb. per sq. ft...		0.19245	1/9	0.01604	0.01414	1/144	13.14	29.1
1 in. water at 62° F........	5.196	1	0.5774	1/12	0.07347	0.036085	68.30	66.3
1 oz. per sq. in..	9	1.732	1	0.1443	0.1272	1/16	118.3	87.2
1 ft. water at 62° F....	62.355	12	6.928	1	0.8816	0.43302	819.6	230
1 in. mercury at 32° F.......	70.73	13.612	7.859	1.1343	1	0.49117	929.6	245
1 lb. per sq. in..	144	27.712	16	2.3094	2.036	1	1893	349
1 atmosphere...	2116.3	407 27	......	33.94	29.921	14.6963	27,815	1338

* The figures in this column show the head in feet of air of uniform density at atmospheric pressure and 62° F. corresponding to the pressure in the preceding columns.

† The figures in this column show the theoretical velocities corresponding to these heads, or the velocities of a jet flowing from a frictionless conical orifice whose flow coefficient is unity.

THE AIR-MANOMETER consists of a long, vertical glass tube, closed at the upper end, open at the lower end, containing air, provided with a scale. It is immersed, together with a thermometer, in a transparent liquid, such as water or oil, contained in a strong glass cylinder which communicates with the vessel in which the pressure is to be ascertained. The scale shows the volume occupied by the air in the tube.

Let v_0 = volume, at 32° F., p_0 = the mean pressure of the atmosphere, v_1 = volume of the air at the temperature t, and under the absolute pressure to be measured p_1; then

$$p_1 = \{(t + 459.6)\ p_0\, v_0\} \div (491.6\, v_1).$$

ATMOSPHERIC PRESSURE AT DIFFERENT ALTITUDES.—The pressure of the atmosphere in lb. per sq. in. at any altitude may be closely approximated for ordinary temperatures by multiplying the height of the mercury barometer, in inches, by 0.491. The pressure increases with the depth below the earth's surface, and is equal to about 1 in. rise in the barometer for each 900 ft. of depth. This is an approximate rule for ascertaining the depth of mine shafts. See Fig. 1.

LEVELING BY BAROMETER.—Many conditions combine to render the results of barometric leveling unreliable where accuracy is required. It is difficult to read an

aneroid (the barometer commonly used by engineers) to within, say, 2 to 6 ft., depending on its size. The results are affected by moisture or dryness of the air, winds, vicinity of mountains, and the daily atmospheric tides, all causing barometric fluctuations. A barometer hanging quietly in a room may vary 0.1 in. within a few hours, corresponding to nearly 100 ft. difference in elevation. No formula can embrace these sources of error.

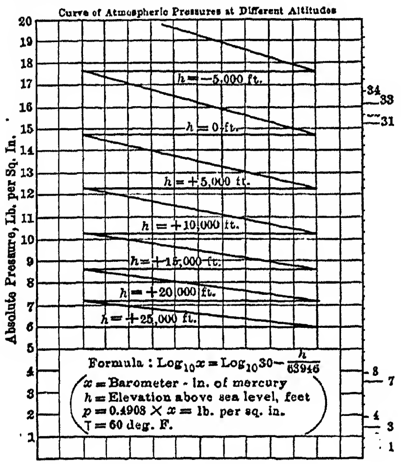


Fig. 1. Variation of Pressure with Altitude

LEVELING BY BOILING POINT OF WATER.—Tables 3 and 4 with explanation below are sufficiently accurate for many purposes.

Take from Table 4 the altitudes opposite the respective boiling temperatures, or the barometer readings. Subtract the one opposite the lower reading from that opposite the higher reading. The remainder will be the approximate required height. To correct this, add together the two thermometer readings, and divide by 2, for their mean. From table of temperature corrections take the number under this mean. Multiply by this number the approximate height just found.

At 70° F. pure water will boil at 1° lower temperature for an average of about 550 ft. of elevation above sea level, up to a height of ½ mile. At a height of 1 mile, 1° boiling temperature will correspond to about 560 ft. of elevation. In the table the mean of the temperatures at the two stations is assumed to be 32° F., at which no correction for temperature is necessary.

MOISTURE IN THE ATMOSPHERE.—Atmospheric air always contains a small quantity of carbon dioxide (see Ventilation), and a varying quantity of aqueous vapor or moisture. The relative humidity of the air at any time is the percentage of moisture

Table 3.—Boiling Point of Water

Temperature in degrees F., barometer in inches of mercury.

In.	.0	.1	.2	.3	.4	.5	.6	.7	.8	.9
28	208.7	208.9	209.1	209.2	209.4	209.5	209.7	209.9	210.1	
29	210.5	210.6	210.8	210.9	211.1	211.3	211.4	211.6	211.8	212.0
30	212.1	212.3	212.4	212.6	212.8	212.9	213.1	213.3	213.5	213.6

Table 4.—Barometer and Boiling Point of Water at Different Altitudes

Boiling Point, deg. F.	Barometer, in.	Altitude above Sea Level, ft.	Boiling Point, deg. F.	Barometer, in.	Altitude above Sea Level, ft.	Boiling Point, deg. F.	Barometer, in.	Altitude above Sea Level, ft.
184	16.79	15,221	195	21.26	9,031	206	26.64	3,115
185	17.16	14,649	196	21.71	8,481	207	27.18	2,589
186	17.54	14,075	197	22.17	7,932	208	27.73	2,063
187	17.93	13,498	198	22.64	7,381	208.5	28.00	1,809
188	18.32	12,934	199	23.11	6,843	209	28.29	1,539
189	18.72	12,367	200	23.59	6,304	209.5	28.56	1,290
190	19.13	11,799	201	24.08	5,764	210	28.85	1,025
191	19.54	11,243	202	24.58	5,225	210.5	29.15	754
192	19.96	10,685	203	25.08	4,697	211	29.42	512
193	20.39	10,127	204	25.59	4,169	211.5	29.71	255
194	20.82	9,579	205	26.11	3,642	212	30.00	S.L. = 0
						212.5	30.30	−261

Corrections for Temperature

Mean temp. F. in shade.....		10°	20°	30°	40°		60°	70°	80°	90°	100°
Multiply by ..	0.933	0.954	0.975	0.996	1.016	1.036	1.058	1.079	1.100	1.121	1.142

Table 5.—Atmospheric Pressure at Different Barometric Readings

a.—Barometer, in. × 0.4912 = lb. per sq. in.; lb. per sq. in. × 144 = lb. per sq. ft.

Barometer, in.	Pressure per sq. in., lb.	Pressure per sq. ft., lb.*	Barometer, in.	Pressure per sq. in., lb.	Pressure per sq. ft., lb.*
28.00	13.75	1980	29.75	14.61	2104
28.25	13.88	1998	30.00	14.74	2122
28.50	14.00	2016	30.25	14.86	2140
28.75	14.12	2033	30.50	14.98	2157
29.00	14.24	2051	30.75	15.10	2175
29.25	14.37	2069	31.00	15.23	2193
29.50	14.49	2086			

* Decimal omitted

For lower pressures see table of the Properties of Saturated Steam, page 5-04.

Table 6.—Weight of Air per Cubic Foot at Different Pressures and Temperatures

Formula: $W = 0.080728 \times (P/14.6963) \times \{491.6/(T + 459.6)\}$.

Temperature		Absolute Pressure P, lb. per sq. in.										
Deg F	Ab.	14.6963	15.696	16.696	19.696	24.696	34.696	54.696	74.696	94.696	114.696	134.696
0	459.6	.086349	.09222	09810	.11573	14511	.20385	.32137	.43888	.55639	.67391	.79141
32	491.6	.080728	.08622	09171	.10819	.13566	.19059	.30045	.41031	.52017	.63004	.73990
42	501.6	.079119	.08450	08989	.10604	.13295	.18679	.29446	.40213	.50980	.61748	.72515
52	511.6	.077572	.08285	.08813	.10396	13035	.18314	.28871	.39427	.49984	.60541	.71097
62	521.6	.076085	.08126	08644	.10197	.12786	.17963	.28317	.38671	.49026	.59380	.69734
70	529.6	.074936	.08004	08513	.10043	.12592	.17691	.27887	.38087	.48285	.58483	.68681
80	539.6	.073547	.07855	.08356	.09857	.12359	.17364	.27372	.37381	.47390	.57399	.67408
90	549.6	.072209	.07712	.08204	.09678	.12134	.17048	.26874	.36701	.46528	.56355	.66182
100	559.6	.070918	.07574	.08057	.09504	.11937	.16743	.26394	.36045	.45697	.55348	.64999
120	579.6	.068471	.07313	.07779	09177	11506	16165	.25483	.34802	.44120	.53438	.62756
140	599.6	.066187	.07069	.07519	08871	.11122	.15626	24633	.33641	.42648	.51656	.60663
160	619.6	.064051	.06841	.07277	.08584	.10763	.15122	.23838	.32555	.41272	.49988	.58705
180	639.6	.062048	06627	.07049	.08316	.10427	.14649	.23093	.31537	.39981	.48425	.568[illegible]
200	659.6	.060167	.06426	.06835	.08064	.10111	.14205	22393	.30581	.38769	.46957	.5514[illegible]
250	709.6	.055297	.05973	.06354	.07496	.09398	.13204	.20815	.28426	.36037	.43649	.51259
300	759.6	.052245	.05580	05936	.07002	.08779	.12335	.19445	.26555	.33665	.40775	.47885
350	809.6	.049019	.05236	05569	.06570	.08237	.11573	18244	.24915	.31586	.38257	.44925
400	859.6	.046168	.04931	.05245	.06188	.07758	.10900	.17183	.23466	.29748	.36032	.42314
450	909.6	.043630	04660	.04957	.05847	07332	.10301	.16238	.22176	.28113	.34051	.39988
500	959.6	.041357	04417	04699	.05543	.06950	09764	.15392	.21020	.26648	.32277	.37905
550	1009.6	.039309	04198	.04466	.05268	.06606	.09280	.14630	.19979	.25329	.30678	.36028
600	1059.6	.037454	.04000	.04255	.05020	.06294	08842	.13939	.19037	.24133	.29230	.34327
650	1109.6	.035766	.03820	.04063	.04793	06010	08444	.13311	.18179	.23046	.27913	.32781
700	1159.6	.034224	03655	.03888	.04587	.05751	.08080	.12737	.17395	.22052	.26710	.31367
800	1259.6	.031507	03365	.03579	.04223	.05294	.07438	.11726	.16014	.20301	.24589	.28877
900	1359.6	029190	.03118	.03316	03912	.04905	.06891	.10864	.14836	.18808	.22781	.26753
1000	1459.6	.027190	.02904	.03089	.03644	.04569	06419	.10119	.13830	.17519	.21220	.24920

Table 7.—Atmospheric Pressure and Barometer Readings, Metric and English

Altitude above Sea Level		Atmospheric Pressure		Barometer Reading	
Feet	Meters	Lb per sq. in.	Kg. per sq. cm	In. of Mercury	Millimeters
0	0.00	14.7	1.0335	30.0	762.00
500	152.40	14.4	1.0124	29.4	746.76
1000	304.80	14.2	0.9984	28.8	731.52
1500	457.20	13.9	.9773	28.3	718.82
2000	609.60	13.6	.9562	27.7	703.58
2500	762.00	13.4	.9421	27.2	690.88
3000	914.40	13.1	.9210	26.7	678.18
3500	1066.80	12.9	.9070	26.2	665.48
4000	1219.20	12.6	.8859	25.7	652.78
4500	1371.60	12.4	.8718	25.2	640.08
5000	1524.00	12.1	.8507	24.7	627.38
5500	1676.40	11.9	.8367	24.3	617.22
6000	1828.80	11.7	.8226	23.8	604.52
6500	1981.20	11.5	.8085	23.4	594.36
7000	2133.60	11.2	.7874	22.9	581.66
7500	2286.00	11.0	.7734	22.5	571.50
8000	2438.40	10.8	.7593	22.1	561.34
9000	2743.20	10.4	.7312	21.2	538.48
10000	3048.00	10.0	.7031	20.4	518.16

8.— t o: Differe ssures and Temperatures

n A ric of 14.7 . in., absolute, at Sea Level

	[illegible]	120				970	4580	.5190	.5800	.6410	.702	.7635	.82	[illegible]	948	.010	1.165	1.318	1.465	1.625	1.930
	.0882	118				3880	4478	.5076	.5674	.6272	.687	.747	.80	86	928	.989	1.139	1.288	1.438	1.588	1.890
	.0864	116				3800	4385	.4970	.5555	.6140	.672	.731	.79	84	908	.968	1.114	1.260	1.406	1.553	1.[illegible]0
	.0846	113				3720	4292	.4863	.5433	.6006	.656	.716	.77	83	889	.947	1.090	1.233	1.376	1.520	1. 0
	.0828	111				3645	4205	.4770	.5330	.5890	.645	.701	.75	81	869	.927	1.067	1.208	1.348	1.489	1. 0
	.0811	08				3570	4121	.4672	.5221	.5771	.632	.687	.74	79	852	.908	1.046	1.184	1.322	1.460	1. 5
	.0795	06				3503	4038	.4576	.5114	.5652	.619	.673	.72	78	835	.890	1.025	1.161	1.296	1.431	1. 1
	.0780	04		[illegible]		3432	3960	.4487	.5014	.5541	.607	.660	.71	76	819	.873	1.006	1.139	1.271	1.403	1. [illegible]
	.0764	02		1803		3362	3882	.4402	.4927	.5447	.596	.649	.70	75	804	.856	.988	1.116	1.245	1.376	1.
70	.0750	.1005		1770	2791		3808	.4316	.4824	.5332	.584	.635	.68		788	.839	.967	1.095	1.223	1.350	1.604
80	.0736	.0988	.	1738	2739		3738	.4234	.4729	.5224	.572	.622	.67		774	.824	.949	1.074	1.199	1.325	1.573
90	.0723	.0970	1	1707	2688		3670	.4154	.4639	.5122	.561	.611	.66		759	.809	.932	1.054	1.177	1.300	1.544
100	0710	0954	1	1676		[illegible]	3602	.4079	.4555	.5033	.551	.599	.64		745	.794	.914	1.035	1.155	1.276	1.517
110	0698	0937	1	1645		070	3542	.4011	.4481	.4950	.542	.589	.63		732	.780	.899	1.017	1.135	1.254	1.491
120	0686	0921	1	1618		018	3481	.3944	.4403	.4866	.533	.579	.62		720	.767	.884	1.001	1.118	1.234	1.465
130	0674	0905	1	590		966	3446	.3924	.4296	.4770	.524	.570	.61		708	.754	.869	.984	1.099	1.214	1.440
140	0663	0889	1	565		915	3364	.3813	.4262	.4711	.516	.561	.60		[illegible]	.742	.855	.968	1.081	1.194	1.416
150	0652	0874	10	541		865	3308	.3751	.4193	.4636	.508	.552	.59			.730	.841	.953	1.064	1.175	1.[illegible]
175	0626	0840	10	482	535	755	3181	.3607	.4033	.4450	.488	.531	.57			.701	.808	.914	1.021	1.128	1.
200	0603	0809	10	427	248	655	3054	.3473	.3882	.4291	.470	.511	.55			.674	.776	.879	.982	1.084	1.
225	0581	0779	0[illegible]	373	163	555	2949	.3344	.3738	.4129	.452	.491	.53			.649	.747	.846	.944	1.043	1.
250	0560	0751		323	085	466	2845	.3223	.3602	.3981	.436	.474	.51		589	.627	.722	.817	.912	1.007	1.
275	0541	0726		278	011	378	2745	.3111	.3478	.3844	.421	.458	.49		568	.605	.697	.789	.881	.972	1.
300	0523	07[illegible]		237	[illegible]	[illegible]	2654	.3008	.3362	.3716	.407	.442	.47		549	.585	.673	.762	.852	.940	1.
350	0491	06[illegible]	0.	160	8		2492	.2824	.3156	.3488	.382	.415	.44		516	.549	.632	.715	.799	.883	1.
400	0463	06[illegible]	0	090	7		2348	.2661	.2974	.3287	.360	.391	.42		[illegible]	.517	.596	.674	.753	.831	.
450	0437	05[illegible]	0	033	6.		2220	.2515	.2810	.3105	.340	.369	.39			.488	.562	.637	.711	.786	.
	041	0555			5		2100	.2380	.2660	.2940	.322	.351	.37			.463	.534	.604	.675	.746	.
	039	0528			4		1996	.2262	.2528	.2794	.306	.333	.35			.440	.507	.573	.641	.709	.
	037	0504			3		1904	.2158	.2412	.2668	.292	317	.34			.419	.483	.547	.611	.675	.

contained in it, as compared with the amount it is capable of holding at the same temperature.

The degree of saturation or relative humidity of the air is determined by the use of the dry and wet bulb thermometer. The degree of saturation for a number of different readings of the thermometer is given in Table 9, condensed from the Hygrometric Tables of the U. S. Weather Bureau.

Below 32° F. the pressure of saturated vapor in contact with ice is given. Values in the last column do not include the heat of the liquid. Below 32° F. the heat of sublimation of ice instead of the latent heat of vaporization is used.

Table 9.—Relative Humidity

Dry-bulb Thermometer, deg. F.	Difference Between Dry- and Wet-bulb Thermometers, deg. F.																									
	2	3	4	5	6	7	8	9	10	11	12	13	14	15	16	17	18	19	20	21	22	23	24	26	28	30
	Percentage of Relative Humidity, Saturation being 100. Barometer = 30 in.																									
32								11																		
35								19																		
40								29	23	15																
45								38	31	25	18															
50								43	38	32	27															
55								49	43						14											
60								53	48																	
65								56	52		44	39	35	31												
70								59	55		48															
75								62	58	54	51											15				
80								64	61	57	54							32	29	26	23	20				
85								66	63	60	56				44	41		36	33	30	27	25				
90								68	65	61	58								36	34	31	29	26	22		
95								69	66	63	60										34	32	30	25		
100								70	68	65	62										37	35	33	28		
110								73	70	67	65						50	48	46	44	42	40	38	34		
120								74	72	69	67		62	60							45	43				
140								77	75	73	70		66	64	62	60					51	49				

MOISTURE IN AIR AT DIFFERENT PRESSURES AND TEMPERATURES.—(Saunders and Hirschberg.)—When air is termed dry or wet, it means only that it is dryer or wetter than air elsewhere. An important function of the atmosphere is the conveyance and distribution of water over the earth. Examples of mean percentages of atmospheric humidity are (Tables U. S. Weather Bureau): Galveston, 85; New York, 73; Walla Walla, Wash., 65; Rapid City, S. D., 60; Salt Lake City, 63; Yuma, Ariz., 42; El Paso, Tex., 39.

Moisture intermixed with air is in the form of transparent and invisible vapor, until a humidity of 100% or the point of saturation (dew point) is reached. This always occurs in the compression and transmission of air at ordinary working pressures, say, 75 lb. per sq. in. gage. When this point is passed, the excess moisture, though still mixed with the air, condenses into actual water. The supersaturated air then appears as fog or mist. In quiescent air the freed water slowly precipitates.

The varying point of saturation of air is determined by its pressure and temperature, especially the latter. At a fixed temperature, a given volume of air is saturated when it contains a certain quantity of water vapor. If the absolute pressure is doubled, reducing the volume one-half, the moisture capacity is reduced in the same proportion; so if the humidity of the free air is 50%, it becomes 100% when the air is compressed to 2 atmospheres or to 15 lb. per sq. in. gage, and if it is compressed to 6 atmospheres or 75 lb. per sq. in. gage, a common working pressure, the humidity becomes 300%, the excess over 100% being deposited. This is the case if the temperature remains constant. But as air is compressed, its temperature rises rapidly, and with each rise of 20° F. its capacity for moisture is doubled. If free air at 60° F. is compressed to 75 lb. per sq. in. gage, its temperature at delivery, however perfectly the compressor cylinder be water-jacketed, will be above 300° F. Due to this increase of temperature, the capacity of air for moisture will have been doubled so many times that, when it leaves the cylinder, the relative humidity of the air will be quite low, although all the original moisture is still present.

The conditions under which the compressed air has the lowest capacity for moisture are high pressure and low temperature. The temperature of the air leaving the compressor is reduced in its flow through the pipes, so that at the point where its work is to be done, the air should be at a low temperature and carry a minimum of moisture, if the condensed moisture has been removed.

To determine what percentage of moisture pure air can contain at various pressures and temperatures, to ascertain how low the "relative humidity" of the atmosphere must be to prevent deposition of water in any part of a compressed-air system and to find to what temperature air drawn from a saturated atmosphere must be cooled to cause

the deposition of moisture to begin, the following formulas and tables (by H. M. Murphy, *Eng. News*, June 18, 1908) are used, based on Dalton's law of gaseous pressures.

Dalton's Law.—The total pressure exerted against the interior of a vessel by a given quantity of a mixed gas enclosed in it is the sum of the pressures which each of the component gases would exert separately if it were enclosed alone in a vessel of the same volume and at the same temperature.

Table 10.—Mixtures of Air and Saturated Vapor
(From Goodenough's Tables)

Temp., deg. F.	Pressure of Saturated Vapor		Weight of Saturated Vapor		Volume in cu. ft.		B.t.u. per lb. of Dry Air above 0° F.	Latent Heat of Vapor, B.t.u.	of 1 lb. of Dry Air with Vapor to Saturate It
	In. of Mercury	Lb. per sq. in.	Per cu. ft.	Per lb. of Dry Air	Of 1 lb. of Dry Air	Of 1 lb. of Dry Air + Vapor			
0	0.0375	0.0184	0.0000674	0.000781	11.58	11.59	0.0	0 964	0.964
10	.0628	.0308	.0001103	.001309	11.83	11.86	2.411	1.608	4.019
20	.1027	.0504	.000177	.002144	12.09	12.13	4.823	2.623	7.446
32	.1806	.0887	.000303	.003782	12.39	12.47	7.716	4.058	11.783
35	.2036	.1000	.000340	.004268	12.47	12.55	8.44	4.57	13 02
40	.2478	.1217	.000410	.005202	12.59	12.70	9.65	5.56	15.21
45	.3003	.1475	.000492	.00632	12.72	12.85	10.86	6.73	17.59
50	.3624	.1780	.000588	.00764	12.84	13.00	12.07	8.12	20.19
55	.4356	.2140	.000699	.00920	12.97	13.16	13.28	9.76	23.04
60	.5214	.2561	.000829	.01105	13.10	13.33	14.48	11.69	26 18
65	.6218	.3054	.000979	.01323	13.22	13.50	15.69	13.96	29 65
70	.7386	.3628	.001153	.01578	13.35	13.69	16.90	16.61	33.51
75	.8744	.4295	.001352	.01877	13.48	13.88	18.11	19.71	37.81
80	1.0314	.5066	.001580	.02226	13.60	14.09	19 32	23 31	42.64
85	1.212	.5955	.001841	.02634	13.73	14.31	20.53	27.51	48.04
90	1.421	.6977	.002137	.03109	13.86	14.55	21.74	32.39	54.13
95	1.659	.8148	.002474	.03662	13.98	14.80	22.95	38.06	61.01
100	1.931	.9486	.002855	.04305	14.11	15.03	24.16	44.63	68.79
105	2.241	1.1010	.003285	.0505	14.24	15.39	25.37	52.26	77 63
110	2.594	1.274	.003769	.0593	14.36	15.73	26.58	61.11	87 69
115	2.994	1.470	.004312	.0694	14.49	16.10	27.79	71.40	99.10
120	3.444	1.692	.004920	.0813	14.62	16.52	29.00	83 37	112.37
130	4.523	2.221	.006356	.1114	14.88	17.53	31.42	113.64	145.06
140	5.878	2.887	.008130	.1532	15.13	18.84	33.85	155.37	189.22
150	7.566	3.716	.01030	.2122	15.39	20.60	36.27	214 03	250 3
160	9.649	4.739	.01294	.2987	15.64	23.09	38.69	299.55	338 2
170	12.20	5.990	.01611	.4324	15.90	26.84	41.12	431.2	472 3
180	15.29	7.51	.01991	.6577	16.16	33.04	43.55	651.9	695 5
190	19.01	9.34	.02441	1.0985	16.41	45.00	45.97	1082.3	1128 3
200	23.46	11.53	.02972	2.2953	16.67	77 24	43.40	2247 5	2296

FORMULAS FOR WEIGHT OF AIR, STEAM AND SATURATED VAPOR.—(Copyright, 1908, by H. M. Prevost Murphy.) Formulas for the weight of 1 cu. ft. of dry air, of 1 cu. ft. of saturated steam or water vapor and the maximum weight of water vapor that 1 lb. of pure air can carry at any pressure and temperature are given below.

The values K and H being given in Table 11 for various temperatures, t, the formulas are:

$$\text{Weight of 1 cu. ft. saturated steam} = 1.325271\,KH/(459.2 + t). \qquad [1]$$

H = elastic force or tension of water vapor or saturated steam, in inches of mercury corresponding to the temperature t (Fahr.) = 2.036 × (gage pressure + atmospheric pressure, in pounds per square inch). K = the ratio of the weight of a volume of saturated steam to an equal volume of pure dry air at the same temperature and pressure, $= 0.6113 + [0.092\,t/(850 - t)]$. Values of K and H corresponding to the various temperatures t are given in Table 11.

$$\text{Weight of 1 cu. ft. pure dry air} = \frac{1.325271M}{459.2 + t} = \frac{2.698102P}{459.2 + t}. \qquad [2]$$

M = absolute pressure, in. of mercury. P = absolute pressure, lb. per sq. in. W = maximum weight, in lb., of water vapor, that 1 lb. of pure air can contain, when the temperature of the mixture is t, and the total or observed, absolute pressure in pounds per square inch is $P = KH/(2.036\,P - H)$.

NOTE.—The results obtained by the use of any of the above formulas agree exactly

with the average data for air and steam weights as given by the most reliable authorities and careful experiments, for all pressures and temperatures; the value of K being correct for all temperatures up to the critical steam temperature of 689° F.

Table 11.—Values of K and H Corresponding to Temperatures from −30° to 434° F.

t	K	H	t	K	H	t	K	H	t	K	H	t	K	H
−30	.6082	.0099	64	.6188	.5962	158	.6323	9.177	252	.6501	62.97	344	.6739	254.2
−28	.6084	.0111	66	.6190	.6393	160	.6326	9.628	254	.6505	65.21	346	.6745	261.0
−26	.6086	.0123	68	.6193	.6848	162	.6330	10.10	256	.6510	67.49	348	.6751	268.0
−24	.6088	.0137	70	.6196	.7332	164	.6333	10.59	258	.6514	69.85	350	.6757	275.0
−22	.6090	.0152	72	.6198	.7846	166	.6336	11.10	260	.6518	72.26	352	.6763	282.2
−20	.6092	.0168	74	.6201	.8391	168	.6340	11.63	262	.6523	74.75	354	.6770	289.6
−18	.6094	.0186	76	.6203	.8969	170	.6343	12.18	264	.6528	77.30	356	.6776	297.1
−16	.6096	.0206	78	.6206	.9585	172	.6346	12.75	266	.6532	79.93	358	.6783	304.8
−14	.6098	.0227	80	.6209	1.024	174	.6350	13.34	268	.6537	82.62	360	.6789	312.6
−12	.6100	.0250	82	.6211	1.092	176	.6353	13.96	270	.6541	85.39	362	.6795	320.6
−10	.6102	.0275	84	.6214	1.165	178	.6357	14.60	272	.6546	88.26	364	.6803	328.7
− 8	.6104	.0303	86	.6217	1.242	180	.6360	15.27	274	.6551	91.18	366	.6809	337.0
− 6	.6107	.0332	88	.6219	1.324	182	.6364	15.97	276	.6555	94.18	368	.6816	345.4
− 4	.6109	.0365	90	.6222	1.410	184	.6367	16.68	278	.6560	97.26	370	.6822	354.0
− 2	.6111	.0400	92	.6225	1.501	186	.6371	17.43	280	.6565	100.4	372	.6829	362.8
0	.6113	.0439	94	.6227	1.597	188	.6374	18.20	282	.6570	103.7	374	.6836	371.8
2	.6115	.0481	96	.6230	1.698	190	.6377	19.00	284	.6575	107.0	376	.6843	380.9
4	.6117	.0526	98	.6233	1.805	192	.6381	19.83	286	.6580	110.4	378	.6850	390.2
6	.6120	.0576	100	.6236	1.918	194	.6385	20.69	288	.6584	113.9	380	.6857	399.6
8	.6122	.0630	102	.6238	2.036	196	.6389	21.58	290	.6590	117.5	382	.6865	409.3
10	.6124	.0690	104	.6241	2.161	198	.6393	22.50	292	.6594	121.2	384	.6871	419.1
12	.6126	.0754	106	.6244	2.294	200	.6396	23.46	294	.6600	125.0	386	.6879	429.1
14	.6128	.0824	108	.6247	2.432	202	.6400	24.44	296	.6604	128.8	388	.6886	439.3
16	.6131	.0900	110	.6250	2.578	204	.6404	25.47	298	.6610	132.8	390	.6893	449.6
18	.6133	.0983	112	.6253	2.731	206	.6407	26.53	300	.6615	136.8	392	.6901	460.2
20	.6135	.1074	114	.6256	2.892	208	.6411	27.62	302	.6620	141.0	394	.6908	470.9
22	.6137	.1172	116	.6258	3.061	210	.6415	28.75	304	.6625	145.3	396	.6915	481.9
24	.6140	.1279	118	.6261	3.239	212	.6419	29.92	306	.6631	149.6	398	.6923	493.0
26	.6142	.1396	120	.6264	3.425	214	.6423	31.14	308	.6636	154.1	400	.6931	504.4
28	.6144	.1523	122	.6267	3.621	216	.6426	32.38	310	.6641	158.7	402	.6939	515.9
30	.6147	.1661	124	.6270	3.826	218	.6430	33.67	312	.6647	163.3	404	.6947	527.6
32	.6149	.1811	126	.6273	4.042	220	.6434	35.01	314	.6652	168.1	406	.6955	539.5
34	.6151	.1960	128	.6276	4.267	222	.6438	36.38	316	.6658	173.0	408	.6962	551.6
36	.6154	.2120	130	.6279	4.503	224	.6442	37.80	318	.6663	178.0	410	.6970	564.0
38	.6156	.2292	132	.6282	4.750	226	.6446	39.27	320	.6669	183.1	412	.6979	576.5
40	.6158	.2476	134	.6285	5.008	228	.6451	40.78	322	.6674	188.3	414	.6987	589.3
42	.6161	.2673	136	.6288	5.280	230	.6455	42.34	324	.6680	193.7	416	.6995	602.2
44	.6163	.2883	138	.6291	5.563	232	.6458	43.95	326	.6686	199.2	418	.7003	615.4
46	.6166	.3109	140	.6294	5.859	234	.6463	45.61	328	.6691	204.8	420	.7012	628.8
48	.6168	.3350	142	.6298	6.167	236	.6467	47.32	330	.6697	210.5	422	.7021	642.5
50	.6170	.3608	144	.6301	6.490	238	.6471	49.08	332	.6703	216.4	424	.7029	656.3
52	.6173	.3883	146	.6304	6.827	240	.6475	50.89	334	.6709	222.4	426	.7037	670.4
54	.6175	.4176	148	.6307	7.178	242	.6479	52.77	336	.6715	228.5	428	.7046	684.7
56	.6178	.4490	150	.6310	7.545	244	.6484	54.69	338	.6721	234.7	430	.7055	699.2
58	.6180	.4824	152	.6313	7.929	246	.6488	56.67	340	.6727	241.1	432	.7064	713.9
60	.6183	.5180	154	.6317	8.328	248	.6492	58.71	342	.6733	247.6	434	.7073	728.9
62	.6185	.5559	156	.6320	8.744	250	.6496	60.81						

Application of the Formulas and Tables. Example 1.—What must be the relative humidity when atmospheric pressure is 14.7 lb. per sq. in. and outside temperature is 60° F., if no moisture is to be deposited in any part of a compressed air system carrying a constant gage pressure of 90 lb. per sq. in.? *Ans.*—The maximum weight of moisture that 1 lb. of pure air can contain at 90 lb. gage, (= 104.7 lb. absolute pressure) and 60° F., is

$$W = \frac{KH}{2.036P - H} = \frac{0.6183 \times 0.5180}{} = 0.001506 \text{ lb.}$$

The maximum weight of moisture that 1 lb. of air can contain at 60° F. and 14.7 lb. (absolute pressure) is

$$W \text{ (at 14.7)} = \frac{0.6183 \times 0.5180}{2.036 \times 14.7 - 0.5180} = 0.01089 \text{ lb.}$$

In order that no moisture may be deposited, the relative humidity must not be above

$$(0.001506 \div 0.01089) \times 100 = 13.83\%.$$

Example 2.—When compressing air into a reservoir carrying a constant gage pressure of 75 lb., from a saturated atmosphere of 14.7 lb. absolute pressure and 70° F., to what temperature must

the air be cooled after compression to cause the deposition of moisture to begin? *Ans.*—First find the maximum weight of moisture contained in 1 lb. of pure air at 14.7 lb. pressure and 70° F.

$$W = \frac{KH}{2.036\,P - H} = \frac{0.6196 \times 0.7332}{2.036 \times 14.7 - 0.7332} = 0.01556 \text{ lb.}$$

The temperature to which the air must be cooled to cause the deposition of moisture may be found by placing 0.01556, together with P equal to 75 + 14.7 in the equation thus:

$$0.01556 = \frac{KH}{2.036 \times 89.7 - H} = \frac{KH}{182.63 - H},$$

or $H = 2.842/(0.01556 + K)$, and the temperature which satisfies this equation is found by aid of the table (by trial and error) to be approximately 129° F.

Example 3.—When the outside temperature is 82° F., and the pressure of the atmosphere is 14.6963 lb. per sq. in., the relative humidity being 100%, how many cu. ft. of free air must be compressed and delivered into a reservoir at 100 lb. gage to cause 1 lb. of water to be deposited when the air is cooled to 82° F.? *Ans.*—Weight of moisture mixed with 1 lb. of air at 82° F., and atmospheric pressure = 0.023526 lb. For 100 lb. gage pressure,

$$W = \frac{KH}{2.036\,P - H} = \frac{0.6211 \times 1.092}{2.036 \times 114.6963 - 1.092} = 0.002918 \text{ lb.}$$

Weight of moisture deposited by each pound of compressed air is equal to 0.023526 − 0.002918 = 0.020608 lb. Each cubic foot of the moist atmosphere contains 0.070595 lb. of pure air. Therefore the number of cubic feet of air that must be delivered to cause 1 lb. of water to be deposited is

$$(1/0.070595) \times (1/0.020608) = 687.37 \text{ cu. ft.}$$

Example 4.—Under the same conditions as in Example 3, what is the loss in volumetric efficiency of the plant when the excess moisture is properly trapped in the main reservoirs? *Ans.*—Before compression, each pound of air is mixed with 0.023526 lb. of water vapor and the weight of 1 cu. ft. of the mixture is 0.072256 lb. Hence the volume of the mixture is

$$1.023526 \div 0.072256 = 14.165 \text{ cu. ft.}$$

For 100 lb. gage pressure and 82° F. as in Example 3, 1 lb. of air can hold 0.002918 lb. of in suspension, having deposited 0.020608 lb. in the reservoir. The weight of 1 cu. ft. of

Table 12.—Weights of Pure Dry Air, Water Vapor and Saturated Mixtures of Air , Water Vapor, and Pressure of Air and Vapor Present in Saturated Mixtures

Air at atmospheric pressure = 14.6963 lb. per sq. in. = 29.921 in. of mercury, and at various temperatures.

(Copyright, 1908, by H. M. Prevost Murphy.)

Temperature, deg. F.	Weight of 1 cu. ft. of Pure Dry Air, lb.	Saturated Mixtures of Air and Water Vapor					
		Elastic Force of the Vapor, in. of Mercury	Elastic Force of the Air Alone, when Saturated, in. of Mercury	Weight of Vapor in 1 cu. ft. of Mixture, or Wt. of 1 cu. ft. of Saturated Steam	Weight of the Air in 1 cu. ft. of the Mixture	Total Weight of 1 cu. ft. of the Mixture	Weight of Water Vapor Mixed with 1 lb. of Air
0	0.086354	0.0439	29.877	0.000077	0.086226	0.086303	0.000898
12	.084154	.0754	29.846	.000130	.083943	.084073	.001548
22	.082405	.1172	29.804	.000198	.082083	.082281	.002413
32	.080728	.1811	29.740	.000300	.080239	.080539	.003744
42	.079117	.2673	29.654	.000435	.078411	.078846	.005554
52	.077569	.3883	29.533	.000621	.076563	.077184	.008116
62	.076081	.5559	29.365	.000874	.074667	.075541	.011709
72	.074649	.7846	29.136	.001213	.072690	.073903	.016691
82	.073270	1.092	28.829	.001661	.070595	.072256	.023526
92	.071940	1.501	28.420	.002247	.068331	.070578	.032877
102	.070658	2.036	27.885	.002999	.065850	.068849	.045546
112	.069421	2.731	27.190	.003962	.063085	.067047	.062806
122	.068227	3.621	26.300	.005175	.059970	.065145	.086285
132	.067073	4.750	25.171	.006689	.056425	.063114	.118548
142	.065957	6.167	23.754	.008562	.052363	.060925	.163508
152	.064878	7.929	21.992	.010854	.047686	.058540	.227609
162	.063834	10.097	19.824	.013636	.042293	.055929	.322407
172	.062822	12.749	17.172	.016987	.036055	.053042	.471146
182	.061843	15.965	13.956	.021000	.028845	.049845	.728012
192	.060893	19.826	10.095	.025746	.020545	.046291	1.25319
202	.059972	24.442	5.479	.031354	.010982	.042336	2.85507
212	.059079	29.921	0.000	.037922	.000000	.037922	Infinite

vapor at 82° is 0.001661 lb.; consequently by Dalton's law the volume of the mixture of 1 lb. of air and 0.002918 lb. of water vapor at 100 lb. gage pressure is the same as that of the vapor or saturated steam alone; that is, $0.002918 \div 0.001661 = 1.757$ cu. ft.

By Mariotte's law, the volume of the 1.757 cu. ft. of mixed gas at 114.6963 lb., absolute, when expanded to atmospheric pressure will be

$$(114.6963 \div 14.6963) \times 1.757 = 13.712 \text{ cu. ft.};$$

hence the decrease of volume, or the loss of volumetric efficiency, is

$$14.165 - 13.712 = 0.453 \text{ cu. ft., or } (0.453 \div 14.165) \times 100 = 3.2\%.$$

This shows that in warm, moist climates, there is an appreciable loss in the efficiency of compressors, due to the condensation of water vapor.

SPECIFIC HEAT OF AIR AT CONSTANT VOLUME AND AT CONSTANT PRESSURE.—Volume of 1 lb. of air at 32° F. and pressure of 14.7 lb. per sq. in. = 12.387 cu. ft. = a column 1 sq. ft. area × 12.387 ft. high. Raising the temperature 1° F. while maintaining atmospheric pressure expands it $^1/_{492}$, or to 12.4122 ft. high, a rise of 0.02522 ft.

Work done = 2116 lb. per sq. ft. × 0.02522 = 53.37 ft.-lb., or $53.37 \div 778 = 0.0686$ heat units.

The specific heat of air at constant pressure C_p, according to Regnault, is 0.2375 B.t.u., but this includes the work of expansion, or 0.0686 B.t.u.; hence the specific heat at constant volume $C_v = 0.2375 - 0.0686 = 0.1689$ B.t.u.

Ratio of specific heat at constant pressure to specific heat at constant volume = $0.2375 \div 0.1689 = 1.406$. (See Specific Heat, p. 3–19.)

2. FLOW OF AIR

FLOW OF AIR THROUGH ORIFICES.—The theoretical velocity of flow in feet per second of any fluid, liquid, or gas through an orifice is $v = \sqrt{2gh} = 8.02\sqrt{h}$, in which h = the "head" or height of the fluid in feet required to produce the pressure of the fluid at the level of the orifice. (For gases the formula holds good only for small differences of pressure on the two sides of the orifice.) The quantity of flow in cubic feet per second is equal to the product of this velocity by the area of the orifice, in square feet, multiplied by a "coefficient of flow," which takes into account the contraction of the vein or flowing stream, the friction of the orifice, etc.

For air flowing through an orifice or short tube, from a reservoir of the pressure p_1 into a reservoir of the pressure p_2, Weisbach gives the following values for the coefficient of flow obtained from his experiments:

Flow of Air Through an Orifice

Coefficient c in formula $v = c\sqrt{2gh}$.

Diam. 1 cm. = 0.394 in.:						
Ratio of pressures	1.05	1.09	1.43	1.65	1.89	2.15
Coefficient	0.555	0.589	0.692	0.724	0.754	0.788
Diam. 2.14 cm. = 0.843 in.:						
Ratio of pressures	1.05	1.09	1.36	1.67	2.01	.....
Coefficient	0.558	0.573	0.634	0.678	0.723	.....

Flow of Air Through a Short Tube

Diam. 1 cm. = 0.394 in., length 3 cm. = 1.181 in.:						
Ratio of pressures $p_1 \div p_2$	1.05	1.10	1.30	.....	.....	.....
Coefficient	0.730	0.771	0.830	.....	.....	.....
Diam. 1.414 cm. = 0.557 in., length 4.242 cm. = 1.670 in.:						
Ratio of pressures	1.41	1.69	.....	.....	.....	.....
Coefficient	0.813	0.822	.....	.....	.....	.....
Diam. 1 cm. = 0.394 in., length 1.6 cm. = 0.630 in. Orifice rounded:						
Ratio of pressures	1.24	1.38	1.59	1.85	2.14	.....
Coefficient	0.979	0.986	0.965	0.971	0.978	.....

Clark (Rules, Tables, and Data, p. 891) gives, for the velocity of flow of air through an orifice due to small differences of pressure.

$$\frac{t-32}{493}$$

or simplified $$V = 352C\sqrt{\left[1 + 0.00203(t - 32)\right]\frac{h}{p}}, \quad \text{[4]}$$

in which V = velocity, ft. per sec.; $2g = 64.4$; h = height of the column of water, in.,

measuring the difference of pressure; t = the temperature, deg. F., and p = pressure, in. of mercury. 773.2 is the volume of air at 32° under a pressure of 29.92 in. of mercury when that of an equal weight of water is taken as 1.

For 62° F., the formula becomes $V = 363C\sqrt{h \div p}$, and if p = 29.92 in., $V = 66.35C\sqrt{h}$.

The coefficient of efflux C, according to Weisbach, is:

For conoidal mouthpiece, of form of the contracted vein, with pressures of from 0.23 to 1.1 atmospheres	C = 0.97 to 0.99
Circular orifices in thin plates	C = 0.56 to 0.79
Short cylindrical mouthpieces	C = 0.81 to 0.84
The same rounded at the inner end	C = 0.92 to 0.93
Conical converging mouthpieces	C = 0.90 to 0.99

The consideration of the adiabatic flow of a perfect gas through a frictionless orifice leads to the equation (R. J. Durley, *Trans.* A. S. M. E., xxvii, 193, 1906),

$$\ldots \, {}^{(\gamma+1)/\gamma} \Big] \quad [5]$$

where W = weight of gas discharged per second, lb.; A = area of cross-section of jet, sq. ft.; P_1 = pressure inside orifice, lb. per sq. ft.; P_2 = pressure outside orifice; V_1 = specific volume of gas inside orifice, cu. ft. per lb.; γ = ratio of specific heat at constant pressure to that at constant volume.

For air, where $\gamma = 1.404$, we have for a circular orifice of diameter d inches, the initial temperature of the air being 60° F. (or 521° abs.),

$$W = 0.000491d^2P_1 \qquad [6]$$

In practice the flow is neither frictionless, nor perfectly adiabatic, and the amount of heat entering or leaving the gas is not known. Hence the weight actually discharged is to be found from the formulas by introducing a coefficient of discharge (generally less than unity) depending on the conditions of the experiment and on the form of orifice employed.

Neglecting the changes of density and temperature occurring as the air passes through the orifice, a simpler, though approximate, formula for the ideal discharge is obtained:

$$W = 0.01369\, d^2 \sqrt{iP/T}. \qquad [7]$$

in which d = diameter, in., i = difference of pressures, in. of water, P = mean absolute

Table 13.—Mean Discharge of Orifice in Pounds per Square Foot per Second
(As found from experiments)

Diameter of Orifice, in.	1-inch Head Discharge per sq. ft.	2-inch Head Discharge per sq. ft.	3-inch Head Discharge per sq. ft.	4-inch Head Discharge per sq. ft.	5-inch Head Discharge per sq. ft.
0.3125	3.060	4.336	5.395	6.188	7.024
0.5005	3.012	4.297	5.242	6.129	6.821
1.002	3.058	4.341	5.348	6.214	6.838
1.505	3.050	4.257	5.222	6.071	6.775
2.002	2.983	4.286	5.284	6.107	6.788
2.502	3.041	4.303	5.224	5.991	6.762
3.001	3.078	4.297	5.219	6.033	6.802
3.497	3.051	4.258	5.202	5.966	6.814
4.002	3.046	4.325	5.264	5.951	6.774
4.506	3.075	4.383	5.508	6.260	7.028

Table 14.—Coefficients of Discharge for Various Heads and Diameters of Orifice

Diameter of Orifice, in.	1-inch Head	2-inch Head	3-inch Head	4-inch Head	5-inch
	0.603	0.606	0.610	0.613	
	.602	.605	.608	.610	.613
1	.601	.603	.605	.606	.607
1 1/2	.601	.601	.602	.603	.603
2	.600	.600	.600	.600	.600
2 1/2	.599	.599	.599	.598	.598
3	.599	.598	.597	.596	.596
3 1/2	.599	.597	.596	.595	.594
4	.598	.597	.595	.594	.593
	.598	.596	.594	.593	

pressure in lb. per sq. ft., and T = absolute temperature, = degrees F. + 461. In the usual case, in which the discharge takes place into the atmosphere, P is approximately 2117 lb. per sq. ft., and

$$W = 0.6299\, d^2 \sqrt{i/T} \quad . \quad . \quad . \quad . \quad . \quad . \quad . \quad . \quad . \quad [8]$$

To obtain the actual discharge the values found by the formula are to be multiplied by an experimental coefficient C, values of which are given in Table 14.

Up to a pressure of about 20 in. of water (or 0.722 lb. per sq. in.) above the atmospheric pressure, the results of formulas [6] and [8] agree closely. At higher differences of pressure, divergence becomes noticeable. The formulas hold good only for orifices of the particular form experimented with, and bored in plates of the same thickness, viz.: iron plates 0.057 in. thick.

The experiments and curves plotted from them indicate that:—

(1) The coefficient for small orifices increases as the head increases, but at a lesser rate for the larger orifices, till for the 2-in. orifice it is almost constant. For orifices larger than 2 in. it decreases as the head increases, and at a greater rate the larger the orifice.

(2) The coefficient decreases as the diameter of the orifice increases, and at a greater rate the higher the head.

(3) The coefficient does not change appreciably with temperature (between 40° and 100° F.).

(4) The coefficient (at heads under 6 in.) is not appreciably affected by the size of the box in which the orifice is placed if the ratio of the areas of the box and orifice is at least 20 : 1.

FLIEGNER'S EQUATION FOR FLOW OF AIR THROUGH AN ORIFICE.— (Peabody's Thermodynamics, also *Trans.* A. S. M. E., xxvii, 194.)

$$W = 0.53A(P/\sqrt{T}). \quad . \quad . \quad . \quad . \quad . \quad . \quad . \quad . \quad . \quad . \quad [9]$$

W = flow, lb. per second; A = area of the orifice (or sum of the areas of all the orifices), sq. in.; P = absolute pressure in the orifice chamber, lb. per sq. in.; T = absolute temperature, deg. F., of the air in the chamber. The formula applies only when the absolute pressure in the reservoir is greater than twice the atmospheric pressure, and for orifices properly made. The orifices are in hardened steel plates $^3/_8$ in. to $^1/_2$ in. thick, accurately ground, with the inside orifice rounded to a radius $^1/_{16}$ in. less than the thickness of the plate, leaving $^1/_{16}$ in. of the hole straight.

Table 15.—Corrected Actual Discharge for Circular Orifices in Plate 0.057 in. Thick
(Pounds per second at 60° F. and 14.7 lb. Barometric Pressure)

Head, in. of Water	Diameter of Orifice, inches										
	0.3125	0.500	1.000	1.500	2.000	2.500	3.000	3.500	4.000	4.500	5.000
$^1/_2$	0.00114	0.00293	0.0117	0.0263	0.0468	0.0732	0.105	0.143	0.187	0.237	0.292
1	.00162	.00416	.0166	.0373	.0663	.103	.149	.202	.264	.334	.413
1 $^1/_2$	.00199	.00510	.0203	.0457	.0811	.127	.182	.248	.323	.409	.505
2	.00231	.00590	.0235	.0528	.0937	.146	.210	.285	.373	.471	.582
2 $^1/_2$	.00259	.00662	.0263	.0591	.105	.163	.235	.319	.416	.526	.649
3	.00285	.00726	.0289	.0648	.115	.179	.257	.349	.455	.575	.710
3 $^1/_2$	.00308	.00786	.0312	.0700	.124	.193	.277	.377	.491	.621	.766
4	.00330	.00842	.0334	.0749	.133	.206	.296	.402	.525	.663	.817
4 $^1/_2$	.00351	.00895	.0355	.0794	.141	.219	.314	.426	.556	.702	.865
5	.00371	.00945	.0375	.0838	.148	.231	.331	.449	.586	.739	.912
5 $^1/_2$	.00390	.00993	.0393	.0879	.155	.242	.347	.471	.613	.774	.953
6	.00408	.01049	.0411	.0918	.162	.252	.362	.492	.640	.808	.995

MEASUREMENT OF AIR BY LOW-PRESSURE NOZZLE TEST. (Compressed Air Society.)—Air from the compressor first passes into the regular receiver, where the pressure is maintained at the rated point by bleeding the air through a globe valve to a second receiver provided with a nozzle. This nozzle is an orifice to atmosphere, with curved inner edges. Its size is such that the pressure in the second receiver will not exceed 1 lb. per sq. in. above the atmosphere, when passing all the air through its opening. The pressure is shown by a water column. A thermometer in the wall of the second receiver, about 2 ft. back of the nozzle, measures the temperature of the flowing air. From temperature, pressure and barometric readings, the volume of air flowing per minute is determined.

Usually, the air at the nozzle is at from 150° to 250° F. The volumetric efficiency of the compressor is obtained by correcting the cubic feet of hot nozzle air to the equivalent volume at the temperature of the compressor intake.

Fig. 2 shows the apparatus. The diameter of receiver B must be at least 2 $^1/_2$ times

the smallest diameter of nozzle throat, preferably larger, and should be 10 to 15 ft. long. The revolutions of the compressor are recorded by a continuously operating counter, readings being taken to the second, every 3 or 4 minutes. When the nozzle is of such size that the water column reading is between 3 and 18 in., the quantity of air is calculated from the formula $V = \sqrt{2gh}$, where V = velocity, ft. per sec., g = acceleration due to

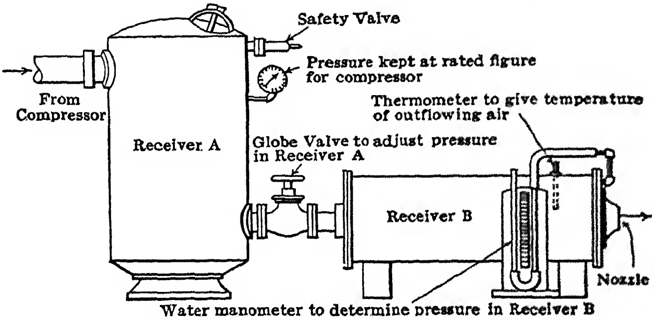


FIG. 2. Apparatus for Low-pressure Nozzle Test

gravity (32.2 usually) ft. per sec. per sec., and h = height, ft., of a column of air of uniform density, corresponding to the observed pressure causing the flow through the nozzle.

Observing the relations of pressure, volume and temperature and the specific weights of air and water, the following formula is derived:

$$Q = 3.64\ Kd^2 \sqrt{(HT \div P_m)} \qquad [10]$$

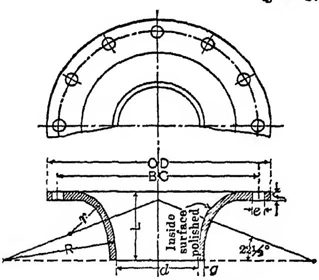


FIG. 3. Standard Nozzle

where Q = cu. ft. of air flowing per min. at the observed temperature of the *upstream side* of nozzle, and the absolute pressure of the *downstream* side (the latter is usually atmospheric pressure); d = diameter of smallest part of nozzle throat, in.; H = observed water column, in.; T = absolute temperature of air entering nozzle = observed temperature, deg. F., plus 460°; P_m = absolute mean pressure between entering and leaving sides of nozzle, lb. per sq. in.; K = coefficient of the nozzle, its value depending upon its shape. With accurately rounded edges, K is between 0.98 and 0.99. Table 16 gives the dimensions of standard nozzles, the inner surfaces of which must be polished and of the form shown by Fig. 3. Another, and more recent, form of nozzle, designed to measure the flow of air in fan and exhauster tests is shown on p. 16–67.

Instead of the above type of nozzle, H. B. Reynolds (*Trans.* A. S. M. E., xxxviii, 1916)

Table 16.—Dimensions of Standard Nozzles

See Fig. 3. Dimensions in inches. Coefficient = 0.99. Capacities calculated for a maximum of 20 in. and a minimum of 3 in. of water column.

d	L	R	r	e	OD	BC	No. of Bolts	Drilled for St'd Flange Size, in.	Capacity, cu. ft Free Air per min. Maximum	Minimum	f	
2	1.70	2.8	1.0	7/8	15.0	13 1/4	12	9	340	140	5/8	
3	2.55	4.2	1.5	7/8	15.0	13 1/4	12	9	755	290	5/8	1/2
4	3.40	5.6	2.0	7/8	15.0	13 1/4	12	9	1330	520	5/8	1/2
5	4.25	7.0	2.5	7/8	15.0	13 1/4	12	9	2065	810	5/8	1/2
7	5.95	9.8	3.5	1	19.0	17	12	12	3990	1560	3/4	1/2
9	7.65	12.6	4.5	1 1/8	29 1/2	21 1/4	16	16	6600	2575	3/4	1/2
11	9.35	15.4	5.5	1 1/4	29 1/2	27 1/4	20	22	9860	3850	3/4	

has proposed the use of plain square orifices in a thin plate, as being easier to make than holes with mathematically rounded and polished edges. The following formula is then applied.

$$Q = \{405A(P_1^2 - P_2^2)^{0.48}\}/\sqrt{T} \quad . \quad . \quad . \quad . \quad . \quad . \quad . \quad . \quad [11]$$

where Q = cu. ft. of air discharged per min., at 32° F. and 14.7 lb. per sq. in., absolute; A = area of orifice, sq. in.; P_1 and P_2 = initial and final pressures, respectively, before and after this orifice, lb. per sq. in., absolute; T = absolute temperature of air entering the orifice, deg. F.

This formula, which gives quite satisfactory results, indicates that, for a stated flow, the area of orifice as computed by formula [9] for round orifices is about 20% too small. Fliegner's formula has sometimes been incorrectly used for square orifices, but is not applicable thereto.

Table 17.—Discharge of Air Through an Orifice

Flowing from a Receiver into the Atmosphere, through a Round Hole with Rounded Inner Edges
(Copyright, 1906, by Ingersoll-Rand Co.)

Receiver Gage Pressure, lb. per sq. in.	Diameter of Orifice, inches													
	1/64	1/32	1/16	1/8	1/4	3/8	1/2	5/8	3/4	7/8	1	1 1/4	1 1/2	2
	Discharge of Free Air per Minute, cu. ft.													
2	0.038	0.153	0.647	2.435	9.74	21.95	39.	61.	87.60	119.50	156.	242	350	625
5	.0597	.242	.965	3.86	15.40	34.60	61.60	96.50	133.	189.	247.	384	550	985
10	.0842	.342	1.36	5.45	21.8	49.	87.	136.	196.	267.	350.	543	780	....
15	.103	.418	1.67	6.65	26.70	60.	107.	167.	240.	326.	427.	665	960	....
20	.119	.485	1.93	7.7	30.8	69.	123.	193.	277.	378.	494.	770	....	....
25	.133	.54	2.16	8.6	34.5	77.	138.	216.	310.	422.	550.	860	....	....
30	.156	.632	2.52	10.	40.	90.	161.	252.	362.	493.	645.	1000	....	....
35	.173	.71	2.80	11.2	44.7	100.	179.	280.	400.	550.	715.	....	....	....
40	.19	.77	3.07	12.27	49.09	110.45	196.35	306.80	441.79	601.32	785.40	....	....	....
45	.208	.843	3.36	13.4	53.8	121.	215.	336.	482.	658.	860.	....	....	....
50	.225	.914	3.64	14.50	58.2	130.	232.	364.	522.	710.	930.	....	....	....
60	.26	1.05	4.2	16.8	67.	151.	268.	420.	604.	622.	......	....	....	....
70	.295	1.19	4.76	19.	76.	171.	304.	476.	685.	930.	......	....	....	....
80	.33	1.33	5.32	21.2	85.	191.	340.	532.	765.	1004.	......	....	....	....
90	.364	1.47	5.87	23.50	94.	211.	376.	587.	843.	....	......	....	....	....
100	.40	1.61	6.45	25.8	103.	231.	412.	645.	925.	....	......	....	....	....
125	.486	1.97	7.85	31.4	125.	282.	502.	785.	.	....	......	....	....	....

3. FLOW OF AIR IN PIPES

GENERAL FORMULAS.—In the flow of a liquid or gas, without friction, according to Bernoulli's theorem, $(V^2 \div 2g) + (p \div w) + z$ = a constant, where V = velocity, ft. per sec.; $2g$ = 64.35; p = absolute pressure, lb. per sq. ft.; w = density, lb. per cu. ft.; and z = height of the section of pipe above a given datum level. When the pipe is level, its axis is taken as datum, and z = 0.

When "fluid friction," or "skin friction" is taken into account there is a "loss of head" or "friction head" between any two selected points, such as the two ends of the pipe; that is, $H = fLv^2 \div R\,2g$; or $H = 4f\dfrac{L}{D}\dfrac{v^2}{2g}$; where H is the loss of head, or head causing the flow, measured in feet, of the fluid; f is a coefficient of friction, and R the mean hydraulic radius, which in circular pipes = 1/4 D. L is the length of the pipe and D the diameter, both in feet. By transposition the velocity in feet per second is

$$[illegible] \sqrt{\frac{[illegible]}{fL}} \quad . \quad . \quad . \quad . \quad . \quad . \quad . \quad . \quad [12]$$

The value of f in formula [12] varies considerably with the roughness of the pipe, with its diameter, and probably to some extent with the velocity of flow. For air and other gases f may be taken approximately as 0.005.

For convenience in calculation, H may be expressed in terms of difference in pressure in lb. per sq. in., or $H = 144\ (p_1 - p_2) \div W$, and the diameter d in inches. Therefore,

$$v = 4.0103\sqrt{\frac{144(p_1 - [illegible]}{fwL}}\ [illegible] = 13.892\sqrt{[illegible]} \qquad [13]$$

The quantity of flow in cubic feet per minute is $Q = 60AV$, where A = the area in sq. ft. = $60 \times 0.7854 \times d^2 \div 144$, whence

$$Q = 4.546 \sqrt{\frac{\;}{\;}} \quad . \quad [14]$$

which is the common formula for flow of any liquid or gas when Q is in cubic feet per minute measured at the density w corresponding to the higher pressure p_1. To reduce this to the equivalent volume of air at atmospheric pressure, $Q_a = Q \times (p_1/14.7)$.

The weight flowing per minute is

$$Qw = W = c\sqrt{\frac{u}{\;}}$$

Values of c corresponding to different values of f are as follows:

f.........	0.003	.0035	.004	.0045	.005	.0055	.006	.0065	.007	.0075
c.........	83.0	76.9	71.9	67.8	64.3	61.3	58.7	56.4	54.7	52.4

The experimental data from which the values of c and f for air and gas may be determined are few in number and of doubtful accuracy. Probably the most reliable are those obtained by Stockalper at the St Gothard tunnel. Unwin found from these data that the value of f varied with the diameter and that it might be expressed by the formula $f = 0\,0028(1 + 3\,6/d)$, d being taken in inches.

	1	2	3	4	6	12	24	48 in.
f =	0.013	.0078	.0062	.0053	.0045	.0036	.0032	.0030
c =	40.0	51.3	57.9	62.3	67.9	75.3	80.1	82.8

Unwin's formula may take the form, $Q = K\sqrt{\frac{(p_1 -}{wL(1}}$ [16]

in which $K = 4\,546\sqrt{1 \div 0\,0028} = 85.9$. This is practically the same as Babcock's formula for steam, in which f is taken at 0.0027, giving $K = 87.5$.

FORMULAS FOR FLOW WITH LARGE DROP IN PRESSURE.—The above formulas are based on the assumption that the drop in pressure is small, and that, therefore, the density remains practically constant during the flow. When the drop is large the density decreases with the pressure and the velocity increases. Church (Mechanics of Engineering, p. 791) and Unwin (*Ency. Brit.*, 11th ed., vol. xiv, p 67), develop formulas for compressible fluids with large drop of pressure and increasing velocity. The temperature is assumed to be constant, the heat generated by friction balancing the cooling due to the work done during expansion.

Church's formula: [17]

Unwin's formula: $\sqrt{\frac{gRTd\,(}{4fl}}\quad p_1^2$ . . . . [18]

In these formulas V = velocity, ft. per sec.; Q = volume, cu. ft. per sec at the pressure p_1; $g = 32.2$; R = the constant in the formula $PV = RT$ (see Thermodynamics) = 53 32 for air; d = diam., and L = length, ft.; p_1, p_2 = absolute pressures in lb. per sq. ft.; w = density, lb. per cu. ft.; T = temperature, deg. F + 459.6. The value of f is given by Church as from 0.004 to 0.005. Unwin makes it vary with the diameter as stated above.

These two formulas give identical results when the value of f is taken the same in both, for $RT/p_1^2 = 1 \div wp_1$.

J. E. Johnson, Jr. (*Am. Mach.*, July 27, 1899) gives formula [17] in a simpler form as follows.

where p_1 and p_2 are the initial and final pressures in lb. per sq. in.; Q = volume of free air (that is the volume reduced to atmospheric pressure) in cubic feet per minute; d = the diameter of the pipe, in ; L = the length, ft.; and K = a numerical coefficient which from the Mt. Cenis and St. Gothard experiments has a value of about 0 0006. E. A. Rix, in a paper on the Compression and Transmission of Illuminating Gas, read before the Pacific Coast Gas Assoc., 1905, uses formula [19] with a coefficient of 0.0005, which he considers more nearly correct than 0.0006. For gas the velocity varies inversely as the square root of the density, and for gas of a density G, relative to air as 1, Rix gives the formula

$$p_1^2 - p_2^2 = 0.0005\sqrt{G} \times Q^2L/d^5 \quad . \quad . \quad . \quad [20]$$

If formula [17] is translated into the same form as [19], taking $f = 0.005$, $w = 0.07608$ for air at 62° F., and atmospheric pressure, 14.7 lb. per sq. in., the value of K is 0.00054. A more convenient form is D'Arcy's:

$$Q_a = C_1\sqrt{(p_1^2 - p_2^2)d^5/L} \quad . \quad . \quad . \quad [21]$$

in which $C_1 = \sqrt{1/K}$. With K in formula [19] taken at 0.0006, $C_1 = 40.8$. With f in formula [17] taken at 0.005, $C_1 = 43.0$.

Note that formula [17] gives Q in cubic feet per second, measured at the pressure p_1, while in formula [19] Q_a is in cubic feet per minute reduced to atmospheric pressure. Both Church and Johnson assume that the flow varies as $\sqrt{d^5}$, the coefficients f and K being independent of the diameter. In this respect their formulas are faulty, for, as Unwin shows, the coefficient of friction is a function of the diameter.

The relation between the results given by these formulas and those given by the common formulas is the relation between $\sqrt{p_1^2 - p_2^2}$ and $\sqrt{p_1 - p_2}$. Taking p_1 (in any unit) as 100, and different drops in pressure, the relative results are as follows:

Pressure drop	1	10	20	40	60	80
Values of p_2	99	90	80	60	40	20
$\sqrt{p_1^2 - p_2^2} \div \sqrt{p_1 - p_2}$	14.1	13.8	13.4	12.2	11.8	10.8
Ratio, 14.1 = 100	100	97.6	95.0	86.5	83.7	76.6

It thus appears that the calculated result by formula [19] is not more than 5% less than that calculated by the common formula, when the same value of f is used, if the drop in pressure is not greater than 20% of p_1.

COMPARISON OF DIFFERENT FORMULAS may be made by applying them to the data of the St. Gothard experiments, as in Table 18. The value of Q is given as reduced to atmospheric pressure, 14.7 lb per sq. in. and 62° F. The length of the 7.87-in. pipe was 15,092 ft., and that of the 5.91-in. pipe, 1712.6 ft. The mean temperature of the air in the large pipe was 70° F. and in the small pipe 80° F.

In Table 18, the figures in the columns headed (14), (16) and (21) refer respectively to:

The common formula, $$c\sqrt{\frac{(p_1 -}{wL}} \qquad [14]$$

Unwin's formula, $$K\sqrt{\frac{(p_1 - p_2)d^5}{wL(1 + 3.6/d)}}. \qquad [16]$$

D'Arcy's formula, [21]

where Q_1 = cu. ft. per min. at pressure p_1; Q_a = cu. ft. per min. reduced to atmospheric pressure = $Qp_1 \div 14.7$.

Table 18.—Comparison of Different Formulas as Applied to St. Gothard Experiments

Diameter, in.	Mean Velocity, ft. per sec.	Cu. ft. per min. Q	Lb. per sec.	Absolute Pressures, lb. per sq. in.		Coefficient in Formula			Ratio of Coefficient to Average Value		
				p_1	p_2	(14) c	(16) K	(21) C_1	(14)	(16)	(21)
7.87	19.3	2105	2.669	82.32	77.03	76.0	89.6	51.3	1.06	1.03	1.09
7.87	16.3	1401	1.776	63.95	60.71	73.5	86.5	49.3	1.02	0.99	1.05
7.87	15.6	1169	1.483	56.45	53.66	70.2	82.8	46.0	0.98	0.95	0.98
5.91	37.1	2105	2.669	77.03	73.50	74.8	94.9	44.5	1.04	1.09	0.95
5.91	29.3	1169	1.483	53.66	52.04	65.5	83.1	43.6	0.91	0.95	0.93
Average						72.0	87.4	46.9			

This comparison shows that no one of the three formulas fits the St. Gothard experiments better than any other; each one when applied with the average value of its coefficient may give a result that differs as much as 9% from the observed result.

ARSON'S EXPERIMENTS.—Unwin quotes some experiments by A. Arson on the flow of air through cast-iron pipes which showed the coefficient of friction to vary with velocity. For a velocity of 100 ft. per sec., and without much error for higher velocities, Unwin finds that values of f agree fairly with the formula $f = 0.005\ (1 + 3.6/d)$. His figures for values of f, translated into values of c for use in the common formula, are:

Diameter of pipe, inches	1.97	3.19	4.06	10	12.8	19.7
Values of c: V = 10 ft. per sec.	35.7	39.4	39.8	49.2	52.8	64.7
50 " "	38.6	42.5	45.0	51.5	55.7	65.2
100 " "	41.3	45.5	46.0	53.6	56.4	65.4

The values of c for the same diameter with $f = 0.0028\ (1 + 3.6/d)$, as deduced by Unwin from Stockalper's experiments are: 51.4, 57.9, 62.3, 73.7, 75.9, 79.1. Unwin states that Stockalper's pipes were probably less rough than Arson's. The values of c from Stockalper's experiments range from 21 to 37% higher than those calculated from the formula derived from Arson's experiments.

USE OF THE FORMULAS.—It is evident from the above comparisons that any formula for the flow of air or gas must be considered as giving only a rough approximation, and that an observed result may differ as much as 40% from that calculated by a formula. This difference is due to variations in the roughness of pipes, to error in measurements

of the actual flow, and to the fact that the coefficients of the several formulas are based on too few experiments. Unwin's formula for moderate drop,

$$Q = 87 \cdot \sqrt{\frac{(p_1 - p_2)d^5}{(1 + 3.6/d)}} \qquad [22]$$

is probably the best one to use for all cases where the drop in pressure does not exceed 20% of the absolute initial pressure, and Johnson's formula,

for cases where the drop is larger and the pipes are not less than 12 in. diameter. For smaller pipes it is best to use the term $(1 + 3.6/d)$ after L in the denominator. These formulas, with the coefficients given, apply only to straight pipes with a fairly smooth interior surface. For crooked or rough pipes it may be well to use the common formula with the coefficients derived from Arson's experiments, given above.

Another comparison of the three formulas [14], [16] and [21] may be made by applying them to some extreme cases, as follows: The initial pressure is taken at 100 lb. absolute per sq. in., the corresponding density is 0.5176 lb. per cu. ft.; diameters are assumed at 1 in. and 48 in., the drop in pressure 1 lb. and 40 lb. and the length 100 ft. and 40,000 ft., making eight cases in all. A ninth case is taken with intermediate values; diameter, 10 in.; length, 1,000 ft.; and drop, 1 lb. The results are given in Table 19. Those obtained from formula [21] have been reduced by dividing them by the ratio (100/14.7) to obtain Q. In formula [14], c is taken at 72, the average figure from the St. Gothard experiments.

Table 19.—Comparison of Formulas for Flow of Air by Application to Extreme Cases

Diam., in.	1				48				10
(p_1-p_2), lb.	1		40		1		40		1
L, ft.	100	40,000	100	40,000	100	40,000	100	40,000	1,000
Formula	Cubic Feet of Air per Minute at the Pressure p_1								
[14]	10.08	0.50	63.3	3.16	159,800	7,990	1,010,000	50,500	1,008
[16]	5.64	0.28	35.7	1.78	186,200	9,310	1,178,000	58,900	1,037
[21]	9.75	0.49	55.3	2.76	155,600	7,778	882,300	44,110	974
	Ratio of Results to Unwin's = 1								
[14]	1.79	1.79	1.78	1.78	0.86	0.86	0.86	0.86	0.96
[21]	1.73	1.75	1.55	1.55	0.84	0.84	0.75	0.75	0.94

These figures show that while the three formulas agree fairly well for the 10-in. pipe with 1-lb. drop in 1000 ft., they show wide disagreements when a great range of diameters, lengths, and drops in pressure are taken. For the 1-in. pipe Unwin's figures (formula 16) are from 35 to 45% lower than those given by formulas [14] and [21], but they are not, therefore, certainly too low. A check on them is supplied by Culley and Sabine's experiments on 2 1/4-in. lead pipes, 2000 to nearly 6000 ft. long, quoted by Unwin, which gave a value of $f = 0.07$. Unwin's formula, $f = 0.0028\ (1 + 3.6/d)$, gives $f = 0.0073$. The corresponding values of c in the common formula are 54.7 and 53.2.

FORMULA FOR FLOW OF AIR AT LOW PRESSURES.—For ventilating and similar purposes, air is usually carried at pressures only slightly above that of the atmosphere. Pressures are measured in inches of water column or in ounces per square inch above atmospheric pressure. For smooth and straight circular pipes, probably the best formula to use is Unwin's,

$$Q = 87 \quad \frac{(p_1 -}{wL(1 + 3.6/d)},$$

the coefficient 87 being derived from the St. Gothard experiments on compressed air. To put the formula into a more convenient form for low pressures, let h = head or difference in pressures measured, in. of water column = 27.712 $(p_1 - p_2)$, and take w = 0.07493 = density of air, lb. per cu. ft. at 70° F. and atmospheric pressure. Then

$$Q = 87 \times \sqrt{\quad h \qquad 1 \quad}$$

or $$Q = C\sqrt{hd^5/L}. \qquad [25]$$

in which C is a coefficient varying with the diameter, values for different diameters being given in Table 20. For other temperatures and pressures, the flow varying inversely as the square root of the density, the figure 0.07493 in the above equation should be replaced by $0.07493 \times \frac{p}{14.7} \times \frac{530}{460 + T}$, in which p = absolute pressure, lb. per sq. in.; T = deg. F.; Q = quantity of air, measured at given pressure and temperature, cu. ft. per min.

In Table 20, Q = cu. ft. per min. $= C\sqrt{hd^5/L}$, h = drop in pressure, in. of water column, d = diam., in., L = length of pipe, ft., C = a coefficient varying with the diameter. The values of C in the table are based on air at atmospheric pressure and 70° F., and the values of Q are calculated for the same pressure and temperature and for a drop of 1 in. of water column in 100 ft.

Table 20.—Values of C and Q, for Different Values of d

d	C	Q	d	C	Q	d	C	Q	d	C	Q
4	43.9	140	10	51.8	1,637	22	56.0	12,700	42	57.9	66,240
5	46.1	257	12	53.0	2,642	24	56.3	15,880	48	58.2	92,930
6	47.7	421	14	53.9	3,950	26	56.6	19,500	54	58.4	125,200
7	49.1	636	16	54.6	5,585	28	56.8	23,580	60	58.6	163,500
8	50.1	908	18	55.1	7,579	30	57.1	28,130	66	58.8	208,000
9	51.0	1,240	20	55.6	9,946	36	57.6	44,760	72	58.9	259,200

For any other pressure drop than 1 in. of water column per 100 ft., multiply Q by the square root of the drop, or by the factor given below:

Drop, h	0.5	2	3	4	6	8	10	12	14	16	18	20
Factor	0.71	1.41	1.73	2	2.45	2.83	3.16	3.46	3.74	4	4.24	4.47

For drop in oz. per sq. in. (1 oz. = 1.732 in. of water) the factors are:

Drop, oz.	0.5	1	2	3	4	5	6	7	8	9	10	12
Factor	0.93	1.32	1.86	2.28	2.63	2.94	3.22	3.48	3.72	3.95	4.16	4.56

B. F. Sturtevant Co. gives the following formula for loss of pressure, oz. per sq. in.

$$p_1 = \frac{Lv^2}{25,000\,d}; \quad v = \sqrt{\frac{25,000\,dp_1}{L}}; \quad d = \sqrt{\frac{0.0000025\,Lv^2}{p}}$$

where p_1 = loss of pressure, oz. per sq. in.; v = velocity, ft. per sec.; d = diam., in.; L = length, ft. From the value of v we obtain the flow in cu. ft. per min.

$$Q = 60\,av = 60 \times 0.7854\,d^2 \div 144 \times \sqrt{(25,000\,dp_1/L)} = 51.74\,(p_1\,d^5/L) \quad [26]$$

If drop is in. of water column, h, then $Q = 39.24\sqrt{hd^5/L}$. . . . . [27]

The value of Q from this formula is 9% less than that given in the above table for a 4-in. pipe, and 33% less for a 72-in. pipe.

Table 21.—Flow of Compressed Air in Pipes of Standard Lap-welded Sizes

Cubic feet per minute. For a drop in pressure of 1 lb. per 1000 ft. length

Nominal, Size, in.	Actual Internal Diameter, in.	Gage Pressure				
		60	70	80	90	100
1/2	0.622	0.5183	0.4868	0.4603	0.4378	0.4183
3/4	0.824	1.176	1.104	1.044	0.9929	0.9487
1	1.049	2.367	2.223	2.103	2.000	1.910
1 1/4	1.380	5.211	4.894	4.628	4.402	4.205
1 1/2	1.610	8.096	7.604	7.191	6.838	6.534
2	2.067	16.40	15.40	14.57	13.85	12.94
2 1/2	2.469	24.10	22.63	21.40	20.35	19.45
3	3.068	49.47	46.46	43.94	41.79	39.92
3 1/2	3.548	73.92	69.43	65.66	62.44	59.66
4	4.026	104.5	98.18	92.85	88.30	84.36
4 1/2	4.506	142.0	133.4	126.2	120.0	114.6
5	5.047	193.5	181.8	171.9	163.5	156.2
6	6.065	316.5	297.3	281.1	267.3	255.4
7	7.023	470.0	441.4	417.4	397.0	379.3
8	7.981	660.8	620.6	586.9	558.1	533.3
9	8.941	892.3	838.1	792.5	753.7	720.1
10	10.02	1204	1131	1070	1017	971.9
11	11.00	1541	1447	1369	1302	1244
12	12.00	1936	1818	1719	1635	1299
13	13.25	2506	2353	2226	2117	2022
14	14.25	3029	2845	2690	2559	2445
15	15.25	3612	3392	3208	3051	2915
17 O.D.	16.214	4237	3979	3763	3579	3419
18 O.D.	17.182	4923	4624	4372	4158	3973
20 O.D.	19.182	6540	6143	5809	5524	5278
22 O.D.	21.25	8512	7994	7560	7189	6869
24 O.D.	23.25	10824	10072	9643	9170	8762

FLOW IN RECTANGULAR PIPES.—To economize space, air pipes for ventilating purposes are commonly made of rectangular instead of circular section. No records of experiments on the flow of air in such pipes are available, but a fair estimate of their capacity as compared with that of circular pipes of the same area may be made on the assumption that they follow the law expressed by Chezy's formula for flow of water, viz.: that the flow is proportional to the square root of the mean hydraulic radius r, which is defined as the quotient of the area divided by the perimeter of the wetted surface. For a circular pipe $r = 1/4 \times$ diameter in feet, and for a square pipe of the same area, $r = 0.222\,d$. For rectangles of the same area, r decreases as the ratio of the longer to the shorter side increases. For different proportions of sides, the values of r and the ratio of $\sqrt{r}$ to the value of $\sqrt{r_1}$ (hydraulic radius of a circular pipe of the same area) are as below.

Ratio of sides....	(circle)	1 (sq.)	1.5	2	3	4	5	6
r = ...	0.25	0.222	0.217	0.209	0.192	0.177	0.165	0.155
Ratio $\sqrt{r} \div \sqrt{r_1}$..	1	0.942	0.932	0.914	0.875	0.842	0.813	0.787

That is, a square pipe has 94% of the carrying capacity of a circular pipe of the same area, and a rectangular pipe whose sides are in the ratio of 6 : 1 has only 79% of the capacity of a circular pipe of the same area.

Formula $Q = c$. Q = cu. ft. per min. measured at pressure p_1 and 62° F. $p_1 - p_2$ = lb. per sq. in.; w = density, lb. per cu. ft.; d = diam. in.; L = length, ft.

Values of $c = 87\sqrt{\dfrac{1}{1 + 3.6/d}}$ are as follows:

Diam., in........	1/2	3/4	1	1 1/4	1 1/2	2	2 1/2	3	3 1/2
c = ...	33.4	37.5	41.3	45.8	48.4	52.5	55.5	59.0	61.3
Diam., in........	4	4 1/2	5	6	7	8	9	10	11
c = ...	63.2	64.8	66.5	68.7	70.7	72.2	73.4	74.5	75.5
Diam., in........	12	13	14	15	17 O.D.	18 O.D.	20 O.D.	22 O.D.	24 O.D.
c = ...	76.3	77.1	77.7	78.2	78.7	79.1	79.8	80.4	80.9

For any other drop than 1 lb. per sq. in. in 1000 ft. length, multiply the figures in the table by the square root of the drop.

Table 22.—Volume of Air Transmitted in Cubic Feet per Minute in Pipes of Various Diameters

Formula $Q = (0.7854/144)\, d^2 v \times 60$

Velocity of Flow, feet per sec.	Actual Diameter of Pipe, in.										
	1	2	3	4	5	6	8	10	12	16	20	24
1	0.327	1.31	2.95	5.24	8.18	11.78	20.94	32.73	47.12	83.77	130.9	188.5
2	0.655	2.62	5.89	10.47	16.36	23.56	41.89	65.45	94.25	167.5	261.8	377.0
3	0.982	3.93	8.84	15.7	24.5	35.3	62.8	98.2	141.4	251.3	392.7	565.5
4	1.31	5.24	11.78	20.9	32.7	47.1	83.8	131	188	335	523	754
5	1.64	6.54	14.7	26.2	41.0	59.0	104	163	235	419	654	942
6	1.96	7.85	17.7	31.4	49.1	70.7	125	196	283	502	785	1131
7	2.29	9.16	20.6	36.6	57.2	82.4	146	229	330	586	916	1319
8	2.62	10.5	23.5	41.9	65.4	94	167	262	377	670	1047	1508
9	2.95	11.78	26.5	47	73	106	188	294	424	754	1178	1696
10	3.27	13.1	29.4	52	82	118	209	327	471	838	1309	1885
12	3.93	15.7	35.3	63	98	141	251	393	565	1005	1571	2262
15	4.91	19.6	44.2	78	122	177	314	491	707	1256	1963	2827
18	5.89	23.5	53	94	147	212	377	589	848	1508	2356	3393
20	6.54	26.2	59	105	164	235	419	654	942	1675	2618	3770
24	7.85	31.4	71	125	196	283	502	785	1131	2010	3141	4524
25	8.18	32.7	73	131	204	294	523	818	1178	2094	3272	4712
28	9.16	36.6	82	146	229	330	586	916	1319	2346	3665	5278
30	9.8	39.3	88	157	245	353	628	982	1414	2513	3927	5655

EFFECT OF BENDS IN PIPES.—The Norwalk Iron Works Co., South Conn., gives the effect of bends in pipes as follows, D being the diameter of the and L the length of straight pipe diameters whose effect is equivalent to the bend:

Radius of elbow.	$5D$	$3D$	$2D$	$1\,1/2\,D$	$1\,1/4\,D$			
L..............	7.85	8.24	9.03	10.36	12.72	17.51	35.09	121.2

E. A. Rix and A. E. Chodzko, in their treatise on compressed air (1896) give the following as the loss in pressure through a 90-deg. bend, V being the velocity of air at

entrance, ft. per sec., R the radius of bend, in., and D the internal diameter of pipe, in:

		2	3	4	5
Loss in lb. per sq. in..........	$0.005V^2$	$0.0022V^2$	$0.0016V^2$	$0.0013V^2$	$0.0012V^2$

REDUCTION OF PRESSURE BY GLOBE VALVES, TEES AND ELBOWS.—The reduction of pressure produced by globe valves is the same as that caused by the addition of lengths of straight pipe, as calculated by the formula

$$\text{Additional length of pipe, inches} = 114\,d \div \{1 + (3.6 \div d)\} \quad . \; . \; . \; . \quad [28]$$

where d is the pipe diameter, in. The reduction of pressure produced by elbows and tees is equal to 2/3 of that caused by globe valves. These additional lengths of pipe for globe valves, elbows and tees must be added in each case to the actual length of straight pipe. Thus a 6-in. pipe 500 ft. long with 1 globe valve, 2 elbows and 3 tees would be equivalent to a straight pipe $500 + 36 + (2 \times 24) + (3 \times 24) = 656$ ft. long. See Table 23.

Table 23.—Pipe Lengths Equivalent to Valves and Fittings

Diam. of Pipe, in.	Additional Length, ft.		Diam. of Pipe, in.	Additional Length, ft.		Diam. of Pipe, in.	Additional Length, ft.	
	Globe Valves	Elbows and Tees		Globe Valves	Elbows and Tees		Globe Valves	Elbows and Tees
1	2	2	4	20	13	12	88	59
1 1/2	4	3	5	28	19	15	115	77
2	7	5	6	36	24	18	143	96
2 1/2	10	7	7	44	30	20	162	108
3	13	9	8	53	35	22	181	120
3 1/2	16	11	10	70	47	24	200	134

MEASUREMENT OF AIR VELOCITY BY ANEMOMETER.—Tests by B. Donkin, Jr. (Inst. Civil Engrs., 1892) on pipes 8 to 20 in. diam., with air velocities of 140 to 690 ft. per min., showed the anemometer to have an error ranging from 14.5% fast to 10% slow,

8 ft. square

1712	1795	1859	1329
1622	1685	1782	1091
1477	1344	1524	1049
1202	1356	1203	1333

Average = 1469

5 × 8 ft.

1170	1209	1288
948	1104	1177
1134	1049	1106

Average = 1132

the percentage of error not being constant. In a 24-in. pipe, air velocity 73 ft. per min., as measured by the descent of a gas holder of 1622 cu. ft. capacity, showed as 44 to 63 ft. per min. on the anemometer, or 13.6% to 39.6% slow. See *Eng. News*, Dec. 22, 1892. The impossibility of measuring the true quantity of air by an anemometer held in one position is shown by the accompanying diagram (Wm. Daniel, *Proc.* Inst. M. E., 1875), of the velocities of air at different points in two airways in a mine.

The anemometer is most frequently used to determine the velocity of air leaving a register or air vent, where the velocity is low. It should be frequently calibrated, and readings made over a series of squares covering the area over which velocity is measured, the average being used in calculations. (See Fan Engineering, Buffalo Forge Co., p. 74).

Tests by L. E. Davies at Armour Inst. of Tech. (*Jour.* A. S. H. V. E., Jan., 1930, April, 1931), developed the formula $Q = 0.5\,CV(A + a) = 0.5\,CVA(1 + p)$ *for supply registers*, with the anemometer in direct contact with the register face and the dial facing the operator. *For exhaust systems*, best results were given by $Q = KVA$, the anemometer being held against the grille, with the dial facing it. In both formulas, Q = cu. ft. of air per min.; V = average of corrected velocity readings, ft. per min.; A = gross area of register or grille, sq. ft.; a = net free area of register, sq. ft.; $p = a/A$ expressed as a decimal; C and K = factors varying with velocity of air through register or grille. Values of C and K are as follows:

V (average)......	150	200	300	400	500	600	700	800
C (supply grilles).	0.952	.957	.967	.977	.985	.992	.998	1.000
K (exhaust grilles)	0.762	.772	.789	.806	.820	.828	.832	.....

EQUALIZATION OF PIPES.—It is frequently desired to know what number of pipes of a given size are equal in carrying capacity to one pipe of a larger size. At the

same velocity of flow the volume delivered by two pipes of different sizes is proportional to the squares of their diameters; thus, one 4-in. pipe will deliver the same volume as four 2-in. pipes. With the same head, however, the velocity is less in the smaller pipe, and the volume delivered varies about as the square root of the fifth power (*i.e.*, as the 2.5 power).

Table 24.—Loss of Pressure in Compressed Air Transmission

(Copyright 1906, by Ingersoll-Rand Company)

Delivery, cu. ft. of Compressed Air per min.	Equiv. Delivery, cu. ft. of Free Air per min.	Size of Pipe, in.															
		1	1 1/4	1 1/2	2	2 1/2	3	3 1/2	4	4 1/2	5	6	8	10	12	14	16
		Loss of Pressure, in Pounds, by Friction in Pipe 1000 ft. Long															
		At 60 Pounds, Gage															
9.84	50	18.24	5.06	1.95	0.42	0.13	0.05										
14.73	75		11.34	4.33	.95	.29	.11	0.05									
19.64	100		20.16	7.79	1.69	.52	.19	.08	0.04								
24.60	125			12.23	2.65	.81	.30	.13	.07	0.03							
29.45	150			17.53	3.80	1.16	.44	.19	.09	.05	0.03						
34.44	175				5.17	1.58	.59	.26	.13	.07	.04	0.01					
39.35	200				6.77	2.09	.78	.36	.17	.09	.06	.02					
49.20	250				10.61	3.24	1.22	.55	.27	.15	.08	.03					
58.90	300				15.20	4.65	1.78	.78	.38	.21	.12	.05	0.01				
68.6	350					6.31	2.37	1.07	.53	.29	.17	.06	.01				
78.6	400					8.28	3.11	1.40	.69	.39	.22	.08	.01				
88.4	450					10.47	3.94	1.77	.88	.48	.28	.11	.02				
98.4	500						4.88	2.20	1.08	.60	.34	.14	.03				
118.1	600						7.03	3.17	1.56	.87	.49	.19	.04	0.01			
137.5	700						9.52	4.29	2.12	1.17	.67	.27	.06	.02			
156.6	800							5.57	2.75	1.52	.87	.34	.08	.03	0.01		
176.5	900							7.08	3.49	1.94	1.17	.43	.09	.03	.01		
196.4	1000							8.77	4.33	2.40	1.37	.54	.12	.04	.02	0.01	
294.5	1500								9.73	5.39	3.08	1.20	.27	.09	.03	.01	
393.7	2000									9.65	5.51	2.16	.41	.16	.06	.03	0.01
492	2500										8.61	3.36	.77	.25	.09	.04	.02
589	3000											4.82	1.11	.36	.14	.06	.03
686	3500											6.54	1.50	.48	.19	.09	.04
786	4000												1.98	.63	.25	.11	.05
884	4500												2.51	.79	.32	.15	.07
984	5000												3.10	.99	.39	.18	.09
		At 80 Pounds, Gage															
7.74	50	14.31	3.96	1.53	0.33	0.10	0.03	0.01									
11.3	75		8.46	3.26	.71	.21	.08	.03	0.01								
15.2	100		15.31	5.92	1.28	.39	.14	.06	.03	0.02	0.01						
19.4	125			9.64	2.09	.64	.24	.11	.05	.03	.01						
23.2	150			13.79	2.99	.91	.34	.15	.07	.04	.02	0.01					
27.2	175				4.09	1.25	.47	.21	.10	.06	.03	.01					
31.0	200				5.34	1.63	.61	.27	.13	.07	.04	.01					
38.7	250				8.32	2.54	.96	.43	.21	.12	.07	.02					
46.5	300				12.01	3.67	1.38	.62	.30	.17	.09	.03					
54.2	350					4.99	1.88	.84	.41	.23	.13	.05	0.01				
62.0	400					6.53	2.45	1.11	.54	.30	.17	.06	.01				
69.7	450					8.25	3.13	1.40	.69	.38	.22	.08	.01				
77.4	500					10.81	3.83	1.73	.85	.47	.27	.10	.02				
92.9	600						5.61	2.46	1.22	.68	.39	.15	.03	0.01			
108.2	700						7.46	3.37	1.66	.92	.53	.20	.04	.01			
124.0	800						9.86	4.42	2.18	1.19	.69	.27	.06	.02	0.01		
139.5	900							5.61	2.77	1.54	.88	.34	.08	.02	.01		
152	1000							6.64	3.29	1.82	1.04	.40	.09	.03	.01		
232	1500							15.41	7.62	4.24	2.43	.95	.22	.06	.02	0.01	
310	2000								13.62	7.58	4.32	1.69	.39	.12	.04	.02	0.01
387	2500									11.79	6.88	2.64	.60	.19	.07	.03	.01
465	3000										9.72	3.79	.87	.28	.11	.05	.02
542	3500										13.25	5.27	1.19	.37	.15	.06	.03
620	4000											6.78	1.55	.49	.19	.09	.04
697	4500											8.54	1.97	.66	.25	.11	.05
774	5000											10.55	2.46	.77	.30	.14	.07

(Table continued on following page).

Table 25 has been calculated on this basis. The figures opposite the intersection of any two sizes is the number of the smaller-size pipes required to equal one of the larger. Thus one 4-in. pipe is equal to 5.7 two-in. pipes.

Table 24.—Loss of Pressure in Compressed Air Transmission—(*Cont.*)

Delivery, cu. ft. of Compressed Air per min.	Equiv. Delivery, cu. ft. of Free Air per min.	Size of Pipe, in. 1	$1\frac{1}{4}$	$1\frac{1}{2}$	2	$2\frac{1}{2}$	3	$3\frac{1}{2}$	4	$4\frac{1}{2}$	5	6	8	10	12	14	16
		Loss of Pressure, in Pounds, by Friction, in Pipe 1000 ft. Long															
		At 100 Pounds, Gage															
6.41	50	11.89	3.29	1.28	0.27	0.08	0.03	0.01	....	....	....	....	....	....	....	....	....
9.61	75	....	7.42	2.87	.62	.19	.07	.03	0.01	....	....	....	....	....	....	....	....
12.81	100	....	13.20	5.11	1.15	.34	.12	.05	.02	0.01	....	....	....	....	....	....	....
15.81	125	....	....	7.75	1.68	.52	.19	.08	.04	.02	0.01	....	....	....	....	....	....
19.22	150	....	....	11.42	2.48	.76	.29	.13	.06	.03	.02	0.01	....	....	....	....	....
22.39	175	....	....	....	3.36	1.03	.39	.17	.09	.04	.03	.01	....	....	....	....	....
25.62	200	....	....	....	4.43	1.36	.51	.23	.12	.06	.04	.02	....	....	....	....	....
31.62	250	....	....	....	6.72	2.06	.77	.35	.17	.09	.05	.02	....	....	....	....	....
38.44	300	....	....	....	9.95	3.04	1.14	.51	.25	.14	.08	.03	....	....	....	....	....
44.78	350	....	....	....	13.41	4.11	1.54	.69	.34	.19	.11	.04	0.01	....	....	....	....
51.24	400	....	....	....	....	5.40	2.06	.92	.45	.25	.15	.05	.01	....	....	....	....
57.65	450	....	....	....	....	6.85	2.57	1.16	.57	.32	.18	.07	.02	....	....	....	....
63.24	500	....	....	....	....	8.21	3.08	1.39	.68	.38	.22	.08	.02	....	....	....	....
76.88	600	....	....	....	....	12.21	4.58	2.14	1.03	.57	.33	.12	.03	0.01	....	....	....
89.56	700	....	....	....	....	....	6.19	2.79	1.38	.77	.44	.17	.04	.01	....	....	....
102.5	800	....	....	....	....	....	8.13	3.67	1.81	1.00	.57	.22	.05	.02	0.01	....	....
115.3	900	....	....	....	....	....	10.23	4.64	2.29	1.27	.76	.28	.06	.02	.01	....	....
126.5	1000	....	....	....	....	....	12.39	5.00	2.76	1.23	.88	.34	.08	.03	.01	....	....
192.2	1500	....	....	....	....	....	....	12.81	6.68	3.51	2.03	.78	.18	.05	.02	0.01	....
256.2	2000	....	....	....	....	....	....	....	11.35	6.61	3.62	1.41	.33	.10	.04	.02	0.01
316.2	2500	....	....	....	....	....	....	....	....	9.50	5.51	2.14	.49	.16	.06	.03	.01
384.4	3000	....	....	....	....	....	....	....	....	14.04	8.11	3.16	.76	.23	.09	.04	.02
447.8	3500	....	....	....	....	....	....	....	....	....	10.95	4.26	.98	.31	.12	.05	.03
512.4	4000	....	....	....	....	....	....	....	....	....	14.48	5.59	1.30	.41	.16	.07	.04
576.5	4500	....	....	....	....	....	....	....	....	....	....	7.04	1.84	.52	.21	.09	.05
632.4	5000	....	....	....	....	....	....	....	....	....	....	8.51	1.93	.63	.25	.11	.08
		At 125 Pounds, Gage															
5.	50	9.88	2.70	1.05	0.23		0.03	0.01									
7.	75	22.20	6.07	2.37	.51	.16	.06	.03	0.01								
10.	100	39.50	10.82	4.22	.91	.28	.10	.05	.02								
13.	125		16.88	6.58	1.42	.43	.16	.07	.04		0.01						
15.	150		24.33	9.47	2.04	.63	.23	.11	.05		.02	0 01					
18.	175		33.05	12.90	2.78	.85	.32	.14	.07		.02	01					
21.	200			16.84	3.63	1.11	.42	.19	.09		.03	01					
26.	250			26.30	5.68	1.73	.65	.29	.15		.05	02					
31.	300			37.90	8.18	2.51	.94	.42	.21		.07	03					
36.	350				11.08	3.39	1.27	.58	.28		.09	04	0.01				
42.	400				14.51	4.44	1.67	.75	.37		.12	05					
47.	450				18.38	5.61	2.11	.95	.47		.15	06					
52.	500				22.68	6.95	2.61	1.18	.58		.18		.02				
63.	600						3.76	1.69	.84		.27		.02	0.01			
73.	700						5.11	2.31	1.14		.36		.03	.01			
84.20	800						6.68	3.01	1.49		.47		.04	.01	0		
94.	900						8.45	3.81	1.88		.60		.05	.02			
105.	1000						10.42	4.71	2.32		.74		.07	.02			
157.	1500						23.48	10.5	5.2		1.65			.0		0.01	....
210.	2000							18.81	9.30		2.94					.02	0.01
263.	2500							40	14.52		4.60	1.80				.02	.01
315.	3000								20.90	11	6.63	2.59				.03	.02
368.	3500								28.51	15.78	9.01	3.53				.04	.02
422.	4000										1.80	4.61	1.06			.06	.03
473.	4500	....									14.90	5.83	1.34			.08	.04
526.	5000	....								20	18.45	7.20	1.65			.10	.05

For longer or shorter pipes the friction loss is proportional to the length, *i.e.*, for 500 ft., $1/2$ of the above; for 4000 ft., four times the above, etc.

Table 25.—Equalization of Pipes

Diam., in.	1	2	3	4	5	6	7	8	9	10	12	14	16	18	20	24
2	5.7	1	.....	.....	.....	.....	.....	.....	.....	.....	.....	.....	.....	.....	.....	...
3	15.6	2.8	1	.....	.....	.....	.....	.....	.....	.....	.....	.....	.....	.....	.....	...
4	32.0	5.7	2.1	1	.....	.....	.....	.....	.....	.....	.....	.....	.....	.....	.....	...
5	55.9	9.9	3.6	1.7	1	.....	.....	.....	.....	.....	.....	.....	.....	.....	.....	...
6	88.2	15.6	5.7	2.8	1.6	1	.....	.....	.....	.....	.....	.....	.....	.....	.....	...
7	130	22.9	8.3	4.1	2.3	1.5	1	.....	.....	.....	.....	.....	.....	.....	.....	...
8	181	32.0	11.7	5.7	3.2	2.1	1.4	1	.....	.....	.....	.....	.....	.....	.....	...
9	243	43.0	15.6	7.6	4.3	2.8	1.9	1.3	1	.....	.....	.....	.....	.....	.....	...
10	316	55.9	20.3	9.9	5.7	3.6	2.4	1.7	1.3	1	.....	.....	.....	.....	.....	...
11	401	70.9	25.7	12.5	7.2	4.6	3.1	2.2	1.7	1.3	.....	.....	.....	.....	.....	..
12	499	88.2	32.0	15.6	8.9	5.7	3.8	2.8	2.1	1.6	1	.....	.....	.....	.....	...
13	609	108	39.1	19.0	10.9	7.1	4.7	3.4	2.5	1.9	1.2	.....	.....	.....	.....	...
14	733	130	47.0	22.9	13.1	8.3	5.7	4.1	3.0	2.3	1.5	1	.....	.....	.....	...
15	871	154	55.9	27.2	15.6	9.9	6.7	4.8	3.6	2.8	1.7	1.2	.....	.....	.....	...
16	.....	181	65.7	32.0	18.3	11.7	7.9	5.7	4.2	3.2	2.1	1.4	1	.....	.....	...
17	.....	211	76.4	37.2	21.3	13.5	9.2	6.6	4.9	3.8	2.4	1.6	1.2	.....	.....	...
18	.....	243	88.2	43.0	24.6	15.6	10.6	7.6	5.7	4.3	2.8	1.9	1.3	1	.....	...
19	.....	278	101	49.1	28.1	17.8	12.1	8.7	6.5	5.0	3.2	2.1	1.5	1.1	.....	...
20	.....	316	115	55.9	32.0	20.3	13.8	9.9	7.4	5.7	3.6	2.4	1.7	1.3	1	...
22	.....	401	146	70.9	40.6	25.7	17.5	12.5	9.3	7.2	4.6	3.1	2.2	1.7	1.3	...
24	.....	499	181	88.2	50.5	32.0	21.8	15.6	11.6	8.9	5.7	3.8	2.8	2.1	1.6	1
26	.....	609	221	108	61.7	39.1	26.6	19.0	14.2	10.9	7.1	4.7	3.4	2.5	1.9	1.2
28	.....	733	266	130	74.2	47.0	32.0	22.9	17.1	13.1	8.3	5.7	4.1	3.0	2.3	1.5
30	.....	871	316	154	88.2	55.9	38.0	27.2	20.3	15.6	9.9	6.7	4.8	3.6	2.8	1.7
36	.....	.....	499	243	130	88.2	60.0	43.0	32.0	24.6	15.6	10.6	7.6	5.7	4.3	2.8
42	.....	.....	733	357	205	130	88.2	63.2	47.0	36.2	19.0	15.6	11.2	8.3	6.4	4.1
48	.....	.....	.....	499	286	181	123	88.2	62.7	50.5	32.0	21.8	15.6	11.6	8.9	5.7
54	.....	.....	.....	670	383	243	165	118	88.2	67.8	43.0	29.2	20.9	15.6	12.0	7.6
60	.....	.....	.....	871	499	316	215	154	115	88.2	55.9	38.0	27.2	20.3	15.6	9.9

4. AIR METERS

DIRECT READING TYPE.—An example of a direct reading meter is the small "Tool-om-eter" made at Plainfield, N. J., which may be used to detect and measure leakage in air lines, valves, hose, cocks, etc., to determine actual compressor capacity as compared with nominal rating or displacement, show where air goes after it is compressed and furnish the facts on disputed questions. The scale of the meter is calibrated to read directly in cubic feet of air per minute, when the gage pressure of the air passing the meter is at 80 lb. per sq. in. For pressures greater or less than 80 lb. per sq. in., a pressure factor is used from a table furnished by the maker. The meters are calibrated in terms of air at 14.7 lb. per sq. in. and 60° F. or 520° absolute temperature. If the air is warmer, the meter will over-register approximately 1% for each 11° F. in excess of 60° F. It ordinarily is not necessary to take account of this for pneumatic tools, etc., as their air consumption is also influenced by temperature. For close measurements where temperature must be considered, multiply the meter readings by the temperature coefficient $K = \sqrt{520 \div (T + 460)}$, where T = temperature, deg. F., of the air passing through the meter.

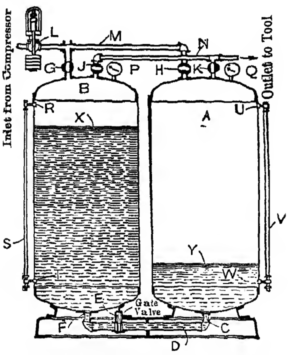


Fig. 4. Water-displacement Meter

WATER DISPLACEMENT TYPE METERS.—(For further information see Peele's Compressed Air Plant, 5th Edition, 1930, Chap. xxv.) There are several forms of this type. The following apparatus will serve to illustrate the principles involved. Fig. 4 is a section through the meter. Fig. 5 gives diagrams ...fferent arrangements of the valves. The meter consists of two tanks, A and B, of the same size and of known volume, mounted side by side. Their upper portions are connected to the source of supply by the inlet pipe M, the opening into tank A being controlled by valve H, and into tank B by valve G. The outlet line also is connected to the top of the tanks by pipe N, the opening into tank A being controlled by valve K, and to tank B by

valve J. The tanks are connected at the bottom by pipe D, in which is a valve E, preferably a quick-opening gate valve. The tanks have gage-glasses V and S and pressure gages Q and P. The inlet line has a pressure-reducing valve L which, while not essential, is a convenience, as it prevents fluctuations in inlet pressure from affecting the readings of the meter, and allows the meter to be used on pressures lower than that of the source of supply. The reducing valve L is set for the desired pressure, connecting the machine whose capacity is to be measured to the outlet line. The valves are set as shown in diagram 1 of Fig. 5, the top part of each tank being open to both inlet and outlet line. Valve E being closed, the water is held stationary in each tank. Air from the inlet line will then pass through the tank to the outlet line and through the machine connected thereto. The meter thus acts as a receiver and has no effect on the air passing through it.

To measure the volume of air used, the throttle valve on the machine is first closed. When the tank pressures equalize, as indicated by the gages, the valves are changed to the position shown in diagram 2 of Fig. 5. Tank B is now in communication with the inlet line, but cut off from the outlet. Tank A is in communication with the outlet line, but closed to the inlet. The level of the water in the gage-glass S is noted, and the valve E opened. The valves are now in the position shown in Fig. 4 and the meter is ready to measure the volume of air passing through the machine. On opening the throttle of the machine, the air in tank A passes through the outlet to the machine. This causes a reduction of pressure in tank A. The air in tank B now causes the water to flow from B to A, equalizing the pressures in both, the pressure in B being maintained by the air

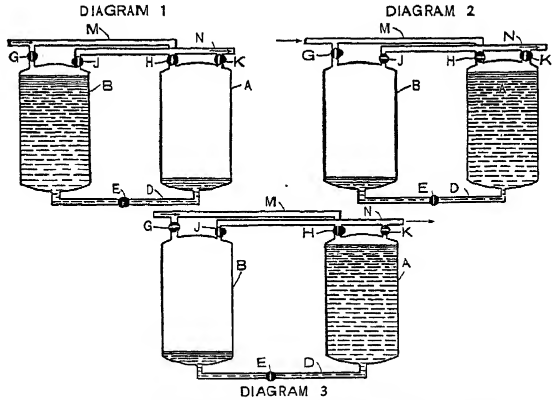


FIG. 5. Valve Operation of Water-displacement Meter

from the inlet line N. This flow of water causes a drop in the water level in gage-glass S. At the end of the interval during which it is desired to measure the volume of air used, valve E is quickly closed, preventing further flow of water. The drop of level in gage-glass S is now measured, from which the volume of air used is calculated.

When the water level rises to the top of tank A, the valves in inlet and outlet lines are thrown to the position shown in diagram 3 of Fig. 5. The meter is now ready for further measurements, after the pressure in the tanks has been equalized and the valve E opened. The conditions are now reversed and the water will flow from tank A to tank B, readings being taken on gage-glass V. In practice the valves can be manipulated with such rapidity that it is unnecessary to stop the machine being tested, and any number of readings can be taken.

The meter readings are taken by using a graduated board placed behind the gage-glasses, one graduation being used for each pressure. On beginning a test, the height of water in each gage-glass is marked, and at the end the drop is measured on the graduated board, thus giving the volume of air used.

Other forms of water-displacement meter have been devised. Among them is a simple single-tank meter, that has been satisfactorily used by the Sullivan Machinery Co. (For details of operation and computation of results, see Peele's Compressed Air Plant, 5th Ed., pp. 503–5.)

LOSS OF ENERGY IN THE PRODUCTION AND USE OF COMPRESSED AIR.—The practical results of Charles's law are (Zahner Transmission of Power by Compressed Air): 1. As the heat of compression increases the volume of the air, the pressure must be carried higher in the compressor cylinder to produce a given volume at a given pressure, after it has returned to normal atmospheric temperature. The work expended in producing this excess pressure is work lost. 2. The heat generated reacts upon the air under compression, and increases the pressure due merely to the reduction of volume. As the compressed air, before it is used, usually has time to cool to the temperature of the surrounding atmosphere, the mechanical equivalent of the dissipated heat is work lost.

Two other statements are deduced from what precedes: 1. Under constant pressure the air varies directly as the absolute temperature, and, at constant volume the absolute p directly as the absolute temperature. These relations are expressed by $(p_1 v_1)/t = p_2$ a constant. This constant is found as follows: Since the density of a given weight of air is inversely proportional to its volume, $v_1 = (1/0.08073)$, 0.08073 being the weight in pounds of 1 cu. ft. of dry air, at sea-level pressure (14.7 lb.) and 32° F. The normal atmospheric pressure per sq. ft. = 14.7 $\times$ 144 = 2116.8 lb. Hence, if, by applying heat, 1 cu. ft. be expanded to 2 cu. ft., the work done against atmospheric pressure, per lb. of air, will be $(2116.8 \times 1)/0.08073 = 26{,}220$ ft.-lb. To double the volume (according to Boyle's law) would require the expenditure of 491.4° F. of heat. Hence, in raising the temperature 1° F., the external work done by expansion is

$$(p_1 v_1/t) = (26{,}220/491.4) = 53.35 = R.$$

The heat generated during compression to different pressures is shown in Table 26, the volume at normal atmospheric pressure being 1, at a temperature of 60° F.

Table 26.—Heat Generated During Compression of Air to Different Pressures

Pressure in Atmospheres	Absolute Pressure, lb. per sq. in.	Vol., cu. ft., Adiabatic Compression	Final Temp., deg. F.	Corresponding Increase of Temp., deg. F.	Pressure in Atmospheres	Abolute Pressure, lb. per sq. in.	Vol., cu. ft., Adiabatic Compression	Final Temp., deg. F.	Corresponding Increase of Temp., deg. F.
1.00	14.70	1.000	60.0	00.0	5.00	73.50	0.319	369.4	309.4
1.25	18.37	.854	94.8	34.8	6.00	88.20	.281	414.5	354.5
1.50	22.05	.750	124.9	64.9	7.00	102.90	.252	454.5	394.5
2.00	29.40	.612	175.8	115.8	8.00	117.60	.229	490.6	430.6
2.50	36.70	.522	218.3	158.3	9.00	132.30	.211	523.7	463.4
3.00	44.10	.459	255.1	195.1	10.00	147.00	.195	554.0	494.0
3.50	51.40	.411	287.8	227.8	15.00	220.50	.147	681.0	621.0
4.00	58.80	.374	317.4	257.4					

ISOTHERMAL AND ADIABATIC COMPRESSION.—If the air during c be kept at constant temperature, the heat being abstracted as fast as it is produced, the compression curve of an indicator diagram from the cylinder is an isothermal curve, following Boyle's law, $(p_2/p_1) = (v_1/v_2)$ = constant.

If the temperature be allowed to rise unchecked during compression, without transference of heat by either radiation or cooling devices, the pressure rises faster than the volume diminishes, and p_2/p_1 is greater than v_1/v_2. The curve of the diagram is then an adiabatic curve, the equation of which is $(p_2/p_1) = (v_1/v_2)^k$.

The value of k for air is found as follows: The heat required to increase the temperature of 1 lb. of air 1° F. at constant pressure, is its *specific heat at constant pressure*. It includes both the external work of expansion and the internal work of raising its temperature. Its value as given by Regnault is 0.2375 B.t.u. per lb. Therefore, the heat required to raise the temperature only, without external work, is 0.2375 − 0.0686 (work of expansion) = 0.1689 B.t.u. per lb. This is the *specific heat of air at constant volume*. In the expression for adiabatic compression, the exponent k is therefore the ratio of these specific heats, or 0.2375/0.1689 = 1.406.

FORMULAS FOR ADIABATIC COMPRESSION OR EXPANSION OF AIR (OR OTHER SENSIBLY PERFECT GAS).—Let air at an absolute temperature T_1, absolute pressure p_1, and volume v_1 be compressed to an absolute pressure p_2 and corresponding volume v_2 and absolute temperature T_2; or let compressed air of an initial pressure, volume, and temperature p_2, v_2, and T_2 be expanded to p_1, v_1, and T_1, there being no transmission of heat from or into the air during the operation. Then the express the relations between pressure, volume, and

$$\left(\frac{p_2}{p_1}\right)^{0.71}; \quad \frac{p_2}{p_1} = \left(\frac{v_1}{v_2}\right)^{1.41}; \quad \frac{v_1}{v_2} \qquad T_1$$

$$\frac{T_2}{T_1} = \left(\frac{v_1}{v_2}\right)^{0.41}; \quad \frac{T_2}{T_1} = \left(\frac{p_2}{p_1}\right)^{0.29}; \quad \frac{p_2}{p_1} = \left(\frac{T_2}{T_1}\right)^{3.46}$$

The exponents are derived from the ratio $c_p \div c_v = k$ of the specific heats of air at

constant pressure and constant volume. Taking $k = 1.406$, $1 \div k = 0.711$; $k - 1 = 0.406$; $1 \div (k - 1) = 2.463$; $k \div (k - 1) = 3.463$; $(k - 1) \div k = 0.289$.

WORK OF ADIABATIC COMPRESSION OF AIR.—If air is compressed in a cylinder without clearance, from a volume v_1 and pressure p_1 to a smaller volume v_2 and higher pressure p_2, work equal to p_1v_1 is done by the external air on the piston while the air is drawn into the cylinder. Work is then done by the piston on the air, first, in compressing it to the pressure p_2 and volume v_2, and then in expelling the volume v_2 from the cylinder against the pressure p_2. If the compression is adiabatic, $p_1v_1^k = p_2v_2^k =$ constant. $k = 1.406$. The work of compression of a given quantity of air is, in foot-pounds,

$$\left(\frac{v_1}{\ }\right)^{k-1} \quad \Big] \qquad p_1 v_1 \left\{ \left(\frac{p_2}{\ _1}\right)^{\frac{k-1}{k}} - 1 \right\} \qquad [3]$$

$$2.463 p_1 v_1 \left\{ \left(\frac{v_1}{v_2}\right)^{0.41} - 1 \right\} = 2.463 p_1 v_1 \quad \left(\frac{p_2}{p_1}\right)^{0.29} - 1 \Big\}. \qquad [4]$$

The work of expulsion is $W_e\ p_2 v_2 = p_1 v_1 \left(\frac{p_2}{p_1}\right)^{0.29}$.

The total work is the sum of the work of compression and expulsion less the work done on the piston during admission, and is

$$\left(\frac{p_2}{p_1}\right)^{\frac{k-1}{k}} - 1 \Big\} = 3. \qquad [5]$$

The mean effective pressure during the stroke is

$$k-1 \qquad 3.463 p_1 \quad \left(\frac{p_2}{p_1}\right)^{0.29} - 1 \Big\} \qquad [6]$$

p_1 and p_2 are absolute pressures above a vacuum, lb. per sq. ft.

Example.—Required the work done in compressing 1 cu. ft. of air per second from 1 to 6 atmospheres, including the work of expulsion from the cylinder.

$p_2 \div p_1 = 6$; $6^{0.29} - 1 = 0.681$; $3.463 \times 0.681 = 2.358$ atmospheres $\times 14.7 = 34.66$ lb. per sq. in. mean effective pressure, $\times 144 = 4991$ lb. per sq. ft., $\times 1$ ft. stroke $= 4991$ ft.-lb., $\div 550$ ft.-lb. per second $= 9.08$ Hp.

If R = ratio of pressures $= p_2 \div p_1$, and if $v_1 = 1$ cu. ft., the work done in compressing 1 cu. ft. from p_1 to p_2 is $3.463\, p_1\, (R^{0.29} - 1)$ ft.-lb., p_1 being taken in lb. per sq. ft. For compression at the sea level p_1 may be taken at 14 lb. per sq. in. = 2016 lb. per sq. ft., as there is some loss of pressure due to friction of valves and passages.

Table 27.—Loss of Work Due to Heat in Compressing Air from Atmospheric Pressure by Simple and Stage Compression

(E. F. Schaefer)

Initial Temperature of Air = 60° F.

Gage Pressure, lb.	One Stage		Two Stage		Three Stage		Four Stage	
	Percentage of Work Lost in Terms of							
	Isothermal Comp.	Adiabatic Comp.	Isothermal Comp.	Adiabatic Comp.	Isothermal Comp.	Adiabatic Comp.	Isothermal Comp.	Adiabatic Comp.
60	29.9	23.0	13.4	11.8	8.6	7.9	4.7	4.5
70	30.6	23.4	14.1	12.4	8.7	8.0	6.1	5.7
80	32.7	24.6	14.7	12.8	9.7	8.9	6.4	6.0
90	34.7	25.8	16.1	13.8	10.5	9.5	7.3	6.8
100	36.7	26.8	16.9	14.5	10.9	9.8	7.8	7.3
125	41.1	29.2	18.5	15.6	11.6	10.4	8.8	8.1
150	44.8	30.9	20.1	16.7	12.3	10.9	9.1	8.4
200	51.2	33.9	22.2	18.1	14.0	12.3	10.5	9.5
300	61.2	37.9	25.7	20.5	16.6	14.2	12.0	10.7
400	68.7	40.7	28.9	22.4	18.2	15.4	13.1	11.5
500	70.6	41.4	31.2	23.8	19.3	16.2	14.1	12.3
600	80.4	44.5	32.8	24.7	20.4	16.9	14.9	13.0
700	85.0	46.0	34.6	25.7	21.3	17.6	16.1	13.8
800	89.5	47.2	35.7	26.3	22.0	18.1	16.2	13.9
900	93.0	48.2	37.1	27.0	22.6	18.5	16.6	14.4
1000	96.1	49.0	37.9	27.5	23.2	18.8	16.9	14.5
1200	102.8	50.7	40.3	28.8	24.8	19.9	17.7	15.0
1400	108.6	52.0						15.7
1600	113.4	53.1						16.1
1800	117.5	54.0						16.4

LOSS OF ENERGY IN THE PRODUCTION AND USE OF COMPRESSED AIR.—he practical results of Charles's law are (Zahner Transmission of Power by Compressed ir): 1. As the heat of compression increases the volume of the air, the pressure must carried higher in the compressor cylinder to produce a given volume at a given pressure, : it has returned to normal atmospheric temperature. The work expended in producing this excess pressure is work lost. 2. The heat generated reacts upon the air under compression, and increases the pressure due merely to the reduction of volume. As the compressed air, before it is used, usually has time to cool to the temperature of the surrounding atmosphere, the mechanical equivalent of the dissipated heat is work lost.

Two other statements are deduced from what precedes: 1. Under constant pressure the volume of air varies directly as the absolute temperature, and, at constant volume the absolute pressure varies directly as the absolute temperature. These relations are expressed by $(p_1 v_1)/t = (p_2 v_2)/t' = R$, a constant. This constant is found as follows: Since the density of a given weight of air is inversely proportional to its volume, $v_1 = (1/0.08073)$, 0.08073 being the weight in pounds of 1 cu. ft. of dry air, at sea-level pressure (14.7 lb.) and 32° F. The normal atmospheric pressure per sq. ft. = 14.7 × 144 = 2116.8 lb. Hence, if, by applying heat, 1 cu. ft. be expanded to 2 cu. ft., the work done against atmospheric pressure, per lb. of air, will be (2116.8 × 1)/0.08073 = 26,220 ft.-lb. To double the volume (according to Boyle's law) would require the expenditure of 491.4° F. of heat. Hence, in raising the temperature 1° F., the external work done by expansio_

$$(p_1 v_1/t) = (26{,}220/491.4) = 53.35 = R.$$

The heat generated during compression to different pressures is shown in 26, at normal atmospheric pressure being 1, at a temperature of

Table 26.—Heat Generated During Compression of Air to Different P

Pressure in Atmospheres	Absolute Pressure, lb. per sq. in.	Vol., cu. ft., Adiabatic Compression	Final Temp., deg. F.	Corresponding Increase of Temp., deg. F.	Pressure in Atmospheres	Abolute Pressure, lb. per sq. in.	Vol., cu. ft., Adiabatic Compression	Final Temp., deg. F.	Corresponding Increase of Temp., deg. F.
1.00	14.70	1.000	60.0	00.0	5.00	73.50	0.319	369.4	309.4
1.25	18.37	.854	94.8	34.8	6.00	88.20	.281	414.5	354.5
1.50	22.05	.750	124.9	64.9	7.00	102.90	.252	454.5	394.5
2.00	29.40	.612	175.8	115.8	8.00	117.60	.229	490.6	430.6
2.50	36.70	.522	218.3	158.3	9.00	132.30	.211	523.7	463.4
3.00	44.10	.459	255.1	195.1	10.00	147.00	.195	554.0	494.0
3.50	51.40	.411	287.8	227.8	15.00	220.50	.147	681.0	621.0
4.00	58.80	.374	317.4	257.4					

ISOTHERMAL AND ADIABATIC COMPRESSION.—If the air du be kept at constant temperature, the heat being abstracted as fast as it compression curve of an indicator diagram from the cylinder is an isoth lowing Boyle's law, $(p_2/p_1) = (v_1/v_2) =$ constant.

If the temperature be allowed to rise unchecked during compression, without transference of heat by either radiation or cooling devices, the pressure rises faster than the volume diminishes, and p_2/p_1 is greater than v_1/v_2. The curve of the diagram is then an adiabatic curve, the equation of which is $(p_2/p_1) = (v_1/v_2)^k$.

The value of k for air is found as follows: The heat required to increase the temperature of 1 lb. of air 1° F. at constant pressure, is its *specific heat at constant pressure*. It includes both the external work of expansion and the internal work of raising its temperature. Its value as given by Regnault is 0.2375 B.t.u. per lb. Therefore, the heat required to raise the temperature only, without external work, is 0.2375 − 0.0686 (work of expansion) = 0.1689 B.t.u. per lb. This is the *specific heat of air at constant volume*. In the expression for adiabatic compression, the exponent k is therefore the ratio of these specific heats, or 0.2375/0.1689 = 1.406.

FORMULAS FOR ADIABATIC COMPRESSION OR EXPANSION OF AIR (OR OTHER SENSIBLY PERFECT GAS).—Let air at an absolute temperature T_1, absolute pressure p_1, and volume v_1 be compressed to an absolute pressure p_2 and corresponding volume v_2 and absolute temperature T_2; or let compressed air of an initial pressure, volume, and temperature p_2, v_2, and T_2 be expanded to p_1, v_1, and T_1, there being no transmission of heat from or into the air during the operation. Then the following express the relations between pressure, volume, and temperature:

$$\frac{v_1}{v_2} = \left(\frac{p_2}{p_1}\right)^{0.71};$$

T_1 — $v_2)$, T_1 — $p_1)$ p_1

The exponents are derived from the ratio $c_p \div c_v = k$ of the specific heats of

constant pressure and constant volume. Taking $k = 1.406$, $1 \div k = 0.711$; $k - 1 = 0.406$; $1 \div (k - 1) = 2.463$; $k \div (k - 1) = 3.463$; $(k - 1) \div k = 0.289$.

WORK OF ADIABATIC COMPRESSION OF AIR.—If air is compressed in a cylinder without clearance, from a volume v_1 and pressure p_1 to a smaller volume v_2 and higher pressure p_2, work equal to p_1v_1 is done by the external air on the piston while the air is drawn into the cylinder. Work is then done by the piston on the air, first, in compressing it to the pressure p_2 and volume v_2, and then in expelling the volume v_2 from the cylinder against the pressure p_2. If the compression is adiabatic, $p_1v_1^k = p_2v_2^k$ = constant. $k = 1.406$. The work of compression of a given quantity of air is, in foot-pounds,

$$W_c \quad k - 1 \qquad \frac{}{k-1}\left\{\left(\frac{p_2}{p_1}\right)^{\frac{k-1}{k}} \right. \qquad [3]$$

$$\left.\left(\frac{p_2}{p_1}\right)^{0.29} - 1\right\}. \qquad [4]$$

The work of expulsion is $W_e\, p_2 v_2 = p_1 v_1 \left(\frac{p_2}{p_1}\right.$

The total work is the sum of the work of compression and expulsion less the work done on the piston during admission, and is

$$W = p_1 v_1 \, k - \qquad [5]$$

The mean effective pressure during the stroke is

$$k-1 \qquad [6]$$

p_1 and p_2 are absolute pressures above a vacuum, lb. per sq. ft.

EXAMPLE.—Required the work done in compressing 1 cu. ft. of air per second from 1 to 6 atmospheres, including the work of expulsion from the cylinder.

$p_2 \div p_1 = 6$; $6^{0.29} - 1 = 0.681$; $3.463 \times 0.681 = 2.358$ atmospheres $\times 14.7 = 34.66$ lb. per sq. in. mean effective pressure, $\times 144 = 4991$ lb. per sq. ft., $\times 1$ ft. stroke $= 4991$ ft.-lb., $\div 550$ ft.-lb. per second $= 9.08$ Hp.

If R = ratio of pressures $= p_2 \div p_1$, and if $v_1 = 1$ cu. ft., the work done in compressing 1 cu. ft. from p_1 to p_2 is $3.463\, p_1\, (R^{0.29} - 1)$ ft.-lb., p_1 being taken in lb. per sq. ft. For compression at the sea level p_1 may be taken at 14 lb. per sq. in. = 2016 lb. per sq. ft., as there is some loss of pressure due to friction of valves and passages.

Table 27.—Loss of Work Due to Heat in Compressing Air from Atmospheric Pressure by Simple and Stage Compression

(E. F. Schaefer)

Initial Temperature of Air = 60° F.

Gage Pressure, lb.	One Stage		Two Stage		Three Stage		Four Stage	
	Percentage of Work Lost in Terms of							
	Iso-thermal Comp.	Adia-batic Comp.	Iso-thermal Comp.	Adia-batic Comp.	Iso-thermal Comp.	Adia-batic Comp.	Iso-thermal Comp.	Adia-batic Comp.
60	29.9	23.0	13.4	11.8	8.6	7.9	4.7	4.5
70	30.6	23.4	14.1	12.4	8.7	8.0	6.1	5.7
80	32.7	24.6	14.7	12.8	9.7	8.9	6.4	6.0
90	34.7	25.8	16.1	13.8	10.5	9.5	7.3	6.8
100	36.7	26.8	16.9	14.5	10.9	9.8	7.8	7.3
125	41.1	29.2	18.5	15.6	11.6	10.4	8.8	8.1
150	44.8	30.9	20.1	16.7	12.3	10.9	9.1	8.4
200	51.2	33.9	22.2	18.1	14.0	12.3	10.5	9.5
300	61.2	37.9	25.7	20.5	16.6	14.2	12.0	10.7
400	68.7	40.7	28.9	22.4	18.2	15.4	13.1	11.5
500	70.6	41.4	31.2	23.8	19.3	16.2	14.1	12.3
600	80.4	44.5	32.8	24.7	20.4	16.9	14.9	13.0
700	85.0	46.0	34.6	25.7	21.3	17.6	16.1	13.8
800	89.5	47.2	35.7	26.3	22.0	18.1	16.2	13.9
900	93.0	48.2	37.1	27.0	22.6	18.5	16.6	14.4
1000	96.1	49.0	37.9	27.5	23.2	18.8	16.9	14.5
1200	102.8	50.7	40.3	28.8	24.8	19.9	17.7	15.0
1400	108.6	52.0						15.7
1600	113.4	53.1						16.1
1800	117.5	54.0						16.4

Theoretical horsepower required to compress adiabatically and deliver 100 cu. ft. of free air per min. = $1.511\,P_1(R^{0.29} - 1)$; P_1 = pressure of free air, lb. per sq. in., abs.

Indicator-cards from compressors in good condition and under working-speeds usually follow the adiabatic line closely. A low curve indicates piston leakage. Such cooling as there may be from the cylinder-jacket and the re-expansion of the air in clearance-spaces tends to reduce the mean effective pressure, while the "camel-backs" in the expulsion-line, due to resistance to opening of the discharge-valve, tend to increase it.

Work of One Stroke of a Compressor, with adiabatic compression, in foot-pounds,

$$W = 3.463P_1V_1(R^{0.29} - 1) \quad . \quad . \quad . \quad . \quad . \quad . \quad . \quad . \quad [7]$$

in which P_1 = initial absolute pressure, lb. per sq. ft. and V_1 = volume traversed by piston, cu. ft.

The work done during adiabatic compression (or expansion) of 1 lb. of air from a volume v_1 and pressure p_1 to another volume v_2 and pressure p_2 is equal to the mechanical equivalent of the heating (or cooling). If t_1 is the higher and t_2 the lower temperature, deg. F., the work done is $c_vJ(t_1 - t_2)$ foot-pounds; c_v = specific heat of air at constant volume = 0.1689, and J = 778, c_vJ = 131.4.

The Work During Compression is

$$v_1\left[\left(\frac{p_2}{p_1}\right)^{0.29} - 1\right], \quad . \quad . \quad . \quad . \quad [8]$$

R_a being the value of pv ÷ absolute temperature for 1 lb. of air = 53.315.

The Work During Expansion is

$$W_e \quad \left[1 - \left(\frac{p_2}{p_1}\right)^{0.29}\right] \qquad \left[\left(\frac{p_1}{p_2}\right)^{0.29} - 1\right], .$$

in which p_1v_1 are the initial, and p_2v_2 the final, pressures and volumes.

Table 28.—Horsepower to Compress 1 Cu. Ft. of Air per Minute from Atmospheric Pressure (14.696 lb.), to Various Gage Pressures—Single-stage Compression

Initial Temperature of Air 60° F.—Jacket cooling neglected

Pressure			Isothermal Compression		Adiabatic Compression: Theoretical		Adiabatic Compression: Theoretical + 15% Friction	
Gage, lb. per sq. in.	Absolute, lb. per sq. in.	Atmospheres	Mean Effective Pressure, lb. per sq. in.	Horsepower	Mean Effective Pressure, lb. per sq. in.	Horsepower	Mean Effective Pressure, lb. per sq. in.	Horsepower
5	19.696	1.34	4.30	0.019	4.51	0.020	5.186	0.023
10	24.696	1.68	7.62	.033	8.27	.036	9.507	.042
15	29.696	2.02	10.33	.045	11.52	.050	13.242	.058
20	34.696	2.36	12.62	.055	14.39	.063	16.550	.072
25	39.696	2.70	14.59	.064	16.99	.074	19.541	.085
30	44.696	3.04	16.33	.071	19.37	.085	22.275	.097
35	49.696	3.38	17.90	.078	21.56	.094	24.798	.108
40	54.696	3.72	19.31	.084	23.61	.103	27.146	.118
45	59.696	4.06	20.59	.090	25.52	.111	29.347	.128
50	64.696	4.40	21.77	.095	27.33	.119	31.425	.137
55	69.696	4.74	22.87	.100	29.03	.127	33.387	.146
60	74.696	5.08	23.89	.104	30.65	.134	35.248	.154
65	79.696	5.42	24.86	.109	32.20	.141	37.028	.162
70	84.696	5.76	25.73	.112	33.68	.147	38.731	.169
75	89.696	6.10	26.57	.116	35.10	.153	40.365	.176
80	94.696	6.44	27.37	.119	36.46	.159	41.928	.183
85	99.696	6.78	28.13	.123	37.77	.164	43.438	.190
90	104.696	7.12	28.85	.126	39.04	.170	44.896	.196
95	109.696	7.46	29.53	.129	40.27	.176	46.307	.202
100	114.696	7.80	30.19	.132	41.45	.181	47.671	.208
110	124.696	8.48	31.43	.137	43.72	.191	50.282	.219
120	134.696	9.16	32.55	.142	45.86	.200	52.741	.230
130	144.696	9.84	33.60	.147	47.89	.209	55.076	.240
140	154.696	10.52	34.59	.151	49.82	.217	57.295	.250
150	164.696	11.20	35.51	.155	51.67	.226	59.420	.259
160	174.696	11.88	36.38	.159	53.44	.233	61.452	.268
170	184.696	12.56	37.20	.162	55.14	.240	63.407	.276
180	194.696	13.24	37.97	.166	56.77	.248	65.286	.285
190	204.696	13.92	38.71	.169	58.34	.255	67.095	.293
200	214.696	14.60	39.41	.172	59.87	.261	68.845	.300

Values of Exponent k.—In the above equations, k is taken at its adiabatic value = 1.406. In practice, this value is closely approached for ordinary single-stage dry compressors, but for large, slow-speed, single-stage compressors in good order, k may become 1.3 to 1.35, and for large, multi-stage compressors it may become as low as 1.2 to 1.25.

STAGE COMPRESSION.—Work (foot-pounds) in the low-pressure or intake cylinder is W_1, and in the high-pressure cylinder, W_2. The compression in each cylinder is assumed to be adiabatic, but an intercooler between them is supposed to reduce to normal temperature the partly compressed air entering the high-pressure cylinder.

$k-1$

First stage . . [10]

. . . [11]

For a given total ratio of pressures, R, the work will be a minimum when it is equally divided between the two cylinders. Assuming perfect intercooling, $p_1 v_1 = p_2 v_3$; whence, by substitution and reduction, and making $k = 1.406$, the work for two-stage compression i

$+ W_2 = 2 \times 3.46$; = 6.9 [12]

Dividing this expression by v_1 gives the mean effective pressure reduced to the low-pressure cylinder, $P_m = 6.92 p_1 (R^{0.145} - 1)$.

For three-stage compression,

$$W_1 + W_2 + W_3 = 10.389\, p_1 v_1 (R^{0.0962} - 1). \quad \ldots \quad [13]$$

Table 29.—Horsepower to Compress 1 Cu. Ft. of Air per Minute from Atmospheric Pressure (14.696 lb.) to Various Gage Pressures—2-stage Compression

Initial Temperature of Air 60° F.—Jacket cooling neglected

Pressures			Correct Ratio of Cylinder Volumes	Intercooler Gage Pressure, lb. per sq. in.	Isothermal Compression		Adiabatic Compression: Theoretical		Adiabatic Compression: Theoretical + 15% Friction		Saving Over Single-stage Compression, per cent
Gage, lb. per sq. in.	Absolute, lb. per sq. in.	Atmospheres			Mean Effective Pressure, lb. per sq. in.	Horsepower	Mean Effective Pressure, lb. per sq. in.	Horsepower	Mean Effective Pressure, lb. per sq. in.	Horsepower	
50	64.696	4.402	2.098	16.14	21.77	0.095	24.38	0.106	28.03	0.122	10.95
60	74.696	5.083	2.255	18.44	23.89	.104	27.04	.118	31.10	.136	11.69
70	84.696	5.763	2.400	20.59	25.73	.112	29.40	.128	33.81	.148	12.43
80	94.696	6.444	2.539	22.26	27.37	.119	31.52	.138	36.24	.158	13.66
90	104.696	7.124	2.650	24.25	28.85	.126	33.50	.146	38.52	.168	14.29
100	114.696	7.805	2.794	26.07	30.19	.132	35.30	.154	40.59	.177	14.90
110	124.696	8.485	2.913	28.11	31.43	.137	36.97	.161	42.51	.186	15.07
120	134.696	9.166	3.027	29.79	32.55	.142	38.52	.168	44.30	.193	16.09
130	144.696	9.846	3.138	31.42	33.60	.147	39.99	.175	45.98	.201	16.25
140	154.696	10.526	3.246	33.01	34.59	.151	41.37	.181	47.58	.208	16.80
150	164.696	11.217	3.349	34.52	35.51	.155	42.68	.186	49.08	.214	17.37
160	174.696	11.887	3.448	35.98	36.38	.159	43.91	.192	50.50	.221	17.53
170	184.696	12.568	3.545	37.40	37.20	.162	45.09	.197	51.86	.226	18.12
180	194.696	13.248	3.640	38.80	37.97	.166	46.22	.202	53.15	.232	18.60
190	204.696	13.929	3.732	40.15	38.71	.169	47.30	.207	54.39	.238	18.77
200	214.696	14.609	3.822	41.47	39.41	.172	48.34	.211	55.59	.243	19.00
210	224.696	15.290	3.910	42.77	40.08	.175	49.32	.215	56.72	.248	....
220	234.696	15.970	3.996	44.03	40.72	.178	50.28	.220	57.82	.253	....
230	244.696	16.651	4.080	45.26	41.33	.180	51.20	.224	58.88	.257	....
240	254.696	17.331	4.163	46.48	41.92	.183	52.10	.228	59.91	.262	....
250	264.696	18.012	4.244	47.67	42.49	.185	52.95	.231	60.90	.266	....
260	274.696	18.692	4.323	48.84	43.03	.188	53.79	.235	61.86	.270	....
270	284.696	19.372	4.401	49.98	43.56	.190	54.60	.238	62.79	.274	....
280	294.696	20.053	4.478	51.11	44.06	.192	55.38	.242	63.69	.278	....
290	304.696	20.733	4.553	52.22	44.55	.194	56.15	.245	64.57	.282	....
300	314.696	21.414	4.627	53.30	45.03	.196	56.89	.248	65.42	.286	....
350	364.696	24.816	4.982	58.51	47.20	.206	60.32	.263	69.36	.303	....
400	414.696	28.218	5.311	63.35	49.09	.214	63.37	.277	72.87	.318	....
450	464.696	31.621	5.622	67.92	50.76	.221	66.10	.289	76.02	.332	....
500	514.696	35.023	5.918	72.28	52.26	.228	68.61	.300	78.91	.345	....

Table 30.—Horsepower to Compress 1 Cu. Ft. of Air per Minute from Atmospheric Pressure (14.696 lb.) to Various Gage Pressures—3-stage Compression

Initial Temperature of Air 60° F.—Jacket cooling neglected

Pressures			Correct Ratio of Cylinder Volumes	Intercooler Gage Pressures, lb. per sq. in.		Isothermal Compression		Adiabatic Compression: Theoretical		Adiabatic Compression: Theoretical + 15% Friction		Saving Over 2-stage Compression, per cent
Gage, lb. per sq. in.	Absolute, lb. per sq. in.	Atmospheres		First Stage	Second Stage	Mean Effective Pressure, lb. per sq. in.	Horsepower	Mean Effective Pressure, lb. per sq. in.	Horsepower	Mean Effective Pressure, lb. per sq. in.	Horsepower	
100	114.696	7.805	1.983	14.45	43.09	30.19	0.132	33.38	0.146	38.38	0.167	5.65
150	164.696	11.207	2.238	18.19	58.91	35.51	.155	39.94	.174	45.93	.200	6.54
200	214.696	14.609	2.445	21.12	73.16	39.41	.172	44.93	.196	51.67	.225	7.41
250	264.696	18.012	2.621	23.82	86.26	42.49	.185	48.95	.213	56.29	.245	7.89
300	314.696	21.414	2.777	26.12	98.64	45.03	.196	52.34	.228	60.19	.262	8.39
350	364.696	24.816	2.917	28.17	130.35	47.20	.206	55.27	.241	63.56	.277	8.58
400	414.696	28.218	3.045	30.05	121.57	49.09	.214	57.85	.252	66.53	.290	8.81
450	464.696	31.621	3.162	31.77	132.24	50.76	.221	60.17	.262	69.20	.302	9.04
500	514.696	35.023	3.272	33.39	142.64	52.26	.228	62.28	.272	71.62	.312	9.57
550	564.696	38.425	3.374	34.89	152.60	53.62	.234	64.20	.280	73.83	.322	.....
600	614.696	41.828	3.471	36.31	162.36	54.87	.239	65.97	.288	75.87	.331	.....
650	664.696	45.230	3.563	37.67	171.97	56.02	.244	67.64	.296	77.78	.339	.....
700	714.696	48.632	3.650	38.94	181.09	57.08	.249	69.18	.302	79.55	.347	......
750	764.696	52.035	3.733	40.16	191.10	58.08	.253	70.61	.308	81.21	.354	......
800	814.696	55.437	3.813	41.34	198.97	59.01	.257	71.98	.314	82.79	.361	......
850	864.696	58.839	3.889	42.46	207.57	59.88	.261	73.27	.320	84.26	.367	......
900	914.696	62.214	3.963	43.54	216.11	60.71	.265	74.49	.325	85.66	.374	......
950	964.696	65.644	4.034	44.59	224.46	61.49	.268	75.67	.330	87.02	.379	......
1000	1014.696	69.046	4.102	45.59	232.59	62.23	.272	76.78	.335	88.30	.385	......
1050	1064.696	72.448	4.169	46.57	240.73	62.94	.275	77.85	.340	89.53	.390	......
1100	1114.696	75.851	4.233	47.51	248.63	63.62	.278	78.86	.344	90.69	.395	......
1150	1164.696	79.253	4.295	48.31	256.40	64.26	.280	79.85	.348	91.83	.400	......
1200	1214.696	82.655	4.356	49.32	264.16	64.88	.283	80.78	.352	92.90	.405	......
1250	1264.696	86.058	4.415	50.19	271.76	65.47	.286	81.68	.356	93.93	.410	......
1300	1314.696	89.460	4.472	51.03	279.21	66.04	.288	82.57	.360	94.95	.414	......
1350	1364.696	92.862	4.528	51.85	286.61	66.59	.291	83.41	.364	95.92	.418	......
1400	1414.696	92.264	4.583	52.66	293.98	67.12	.293	84.23	.367	96.87	.422	......
1500	1514.696	103.069	4.689	54.20	308.42	68.12	.297	85.79	.374	98.66	.430	......
1600	1614.696	109.874	4.789	55.68	322.35	69.06	.301	87.32	.381	100.41	.438	......

Table 31.—Mean and Terminal Pressures of Compressed Air Used Expansively—Gage Pressures from 60 to 100 Lb.

(Frank Richards, *Am. Mach.*, April 13, 1893)

Point of Cut-off	Initial Pressure, Gage (Pressures in italic are absolute): 60		70		80		90		100	
	Mean Pressure	Terminal Pressure	Mean Pressure	Terminal Pressure	Mean Pressure	Terminal Pressure	Mean Pressure	Terminal Pressure	Mean Pressure	Terminal Pressure
0.25	23.6	*10.65*	28.74	*12.07*	33.89	*13.49*	39.04	*14.91*	44.19	1.33
0.30	28.9	*13.77*	34.75	0.6	40.61	2.44	46.46	4.27	53.32	6.11
1/3	32.13	0.96	38.41	3.09	44.69	5.22	50.98	7.35	57.26	9.48
0.35	33.66	2.33	40.15	4.38	46.64	6.66	53.13	8.95	59.62	11.23
3/8	35.85	3.85	42.63	6.36	49.41	7.88	56.2	11.39	62.98	13.89
0.40	37.93	5.64	44.99	8.39	52.05	11.14	59.11	13.88	66.16	16.64
0.45	41.75	10.71	49.31	12.61	56.9	15.86	64.45	19.11	72.02	22.36
0.50	45.14	13.26	53.16	17.	61.18	20.81	69.19	24.56	77.21	28.33
0.60	50.75	21.53	59.51	26.4	68.28	31.27	77.05	36.14	85.82	41.01
5/8	51.92	23.69	60.84	28.85	69.76	34.01	78.69	39.16	87.61	44.32
2/3	53.67	27.94	62.83	33.03	71.99	38.68	81.14	44.33	90.32	49.97
0.70	54.93	30.39	64.25	36.44	73.57	42.49	82.9	48.54	92.22	54.59
0.75	56.52	35.01	66.05	41.68	75.59	48.35	85.12	55.02	94.66	61.69
0.80	57.79	39.78	67.5	47.08	77.2	54.38	86.91	61.69	96.61	68.99
7/8	59.15	47.14	69.03	55.43	78.92	63.81	88.81	72.	98.7	80.28
0.90	59.46	49.65	69.38	58.27	79.31	66.89	89.24	75.52	99.17	87.82

Horsepower required to compress and deliver 100 cu. ft. free air per min.:

For two-stage, $\mathrm{Hp.} = 3.022\,p_1(R^{0.145} - 1)$ . . . . [14]

For three-stage, $\mathrm{Hp.} = 4.53\,p_1(R^{0.0962} - 1)$. . . . . . . [15]

EXAMPLE.—To compress 100 cu. ft. free air per min. in two stages, from 1 to 6 atmospheres, $p_1 = 14.7$ and $R = 6$; hence, $3.022 \times 14.7 \times 0.2964 = 13.17$ Hp.

TO FIND THE INDEX OF THE CURVE OF AN AIR DIAGRAM.—If P_1V_1 be pressure and volume at one point on the curve, and PV the pressure and volume at another point, then $\frac{P}{P_1} = \left(\frac{V_1}{V}\right)^x$, in which x is the index to be found. Let $P \div P_1 = R$, and $V_1 \div V = r$; then $R = r_x$; $\log R = x \log r$, whence $x = \log R \div \log r$. (A graphic method is described in *Am. Mach.*, June 21, 1900.)

COMPRESSED-AIR ENGINES, ADIABATIC EXPANSION.—Let the initial pressure and volume taken into the cylinder be p_1 lb. per sq. ft. and v_1 cu. ft.; let expansion take place to p_2 and v_2 according to the adiabatic law $p_1 v_1^{1.41} = p_2 v_2^{1.41}$; then at the end of the stroke let the pressure drop to the back-pressure p_3, at which the air is exhausted. Assuming no clearance, the work done by 1 lb. of air during admission, measured above vacuum, is p_1v_1, the work during expansion is $2.463\ p_1 v_1\{1 - (p_2/p_1)^{0.29}\}$, and the negative or back-pressure work is $-\,p_3 v_2$. The total work is

$$p_1 v_1 + 2.463\ p_1 v_1\{1 - (p_2/p_1)^{0.29}\} - p_3 v_2,$$

and the mean effective pressure is the total work divided by v_2.

If the air is expanded down to the back-pressure p_3 the total work is

$$3.463\ p_1 v_1\{1 - (p_3/p_1)^{0.29}\},$$

or, in terms of the final pressure and volume, $3.463\,p_3 v_2\{(p_1/p_3)^{0.29} - 1\}$, and the mean effective pressure is $3.463\,p_3\{(p_1/p_3)^{0.29} - 1\}$.

The actual work is reduced by clearance. Then the product of the initial pressure p_1 by the clearance volume is to be subtracted from the total work calculated from the initial volume v_1, including clearance. (See p. 7–13 under "Steam-engine.") For further data on air engines see Peele's Compressed Air Plant, 5th ed., 1930, Chap. xvi.

CENTRAL COMPRESSED AIR PLANTS.—Independent plants, selling large volumes of compressed air, are not common. The most extensive in 1932 are those of the Victoria Falls and Transvaal Power Co., furnishing metered air to groups of deep Johannesburg gold mines, South Africa. Turbo-compressors are used, having unit capacities of 12,000 to 60,000 cu. ft. of free air per min., compressed to 135–170 lb. per sq. in. gage; total, about 300,000 Hp. See *Trans.* So. African Inst. of Engrs., Jan., 1925, also p. 1–51, herein.

2. AIR COMPRESSION AT ALTITUDES

EFFECT OF ALTITUDE ON COMPRESSOR CAPACITY.—Other things being equal, the volumetric efficiency of a compressor depends upon cylinder clearance and compression ratio. In a single-stage compressor, since the compression ratio depends on the discharge and intake pressures, it varies with the altitude for given discharge gage pressure. See Fig. 8. The change of volumetric efficiency with compression ratio is due to the re-expansion of some of the air trapped in the clearance space, and has no relation to the relative density of the air at the altitude, nor to any so-called "equivalent free air at sea-level." The volumetric efficiency is expressed as the percentage of "free air" delivered, measured at the temperature and pressure of the compressor intake, ordinarily depending on the local atmospheric conditions.

In the case of a compound compressor, the volumetric efficiency depends upon the

Table 32.—Multipliers to Determine the Volume of Free Air Which, When Compressed, Is Equivalent in Effect to a Given Volume of Free Air at Sea Level

(Ingersoll-Rand Co. Copyright, 1906, by F. M. Hitchcock)

Altitude, ft.	Barometric Pressure		Gage Pressure (lb.)				
	In. of Mercury	Lb. per sq. in.	60	80	100	125	150
1,000	28.88	14.20	1.032	1.033	1.034	1.035	1.036
2,000	27.80	13.67	1.064	1.066	1.068	1.071	1.072
3,000	26.76	13.16	1.097	1.102	1.105	1.107	1.109
4,000	25.76	12.67	1.132	1.139	1.142	1.147	1.149
5,000	24.79	12.20	1.168	1.178	1.182	1.187	1.190
6,000	23.86	11.73	1.206	1.218	1.224	1.231	1.234
7,000	22.97	11.30	1.245	1.258	1.267	1.274	1.278
8,000	22.11	10.87	1.287	1.300	1.310	1.319	1.326
9,000	21.29	10.46	1.329	1.346	1.356	1.366	1.374
10,000	20.49	10.07	1.373	1.394	1.404	1.416	1.424

compression ratio of the first stage, which is determined by the cylinder ratio. Hence for a given cylinder ratio, the volumetric efficiency is practically unaffected by a change of altitude.

EFFECT OF ALTITUDE ON DRILL REQUIREMENTS.—Besides the change of volumetric efficiency with altitude of a compressor, the amount of air required by a drill or other air machine increases with the altitude, due to the greater volume of free air

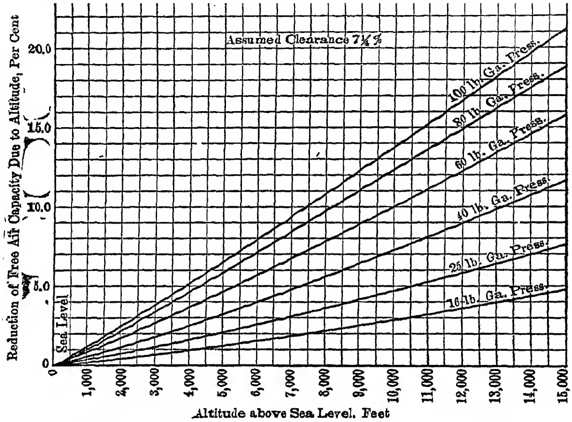


FIG. 8. Effect of Altitude on Compressor Capacity. (S. B. Redfield)

at local atmospheric conditions that will be forced into the drill cylinder under the greater number of compressions at the altitude. Thus there are two distinct effects: The drill requires more, and the single-stage compressor delivers less free air at a higher altitude.

Selection of Compressor for a given number of drills, pneumatic tools, hoisting engines or other class of pneumatic machines at an altitude: Add to the sea-level air consumption of the machines, the percentage increase they require at the altitude. Then select a compressor whose sea-level capacity is such that, when it is reduced by the percentage loss of volumetric efficiency for the altitude, shown by the curves in Fig. 8, its net delivery will at least equal the total air requirements. The diagram figures refer to air as measured by nozzle, not to piston displacement.

EXAMPLE.—Free air required by drills at 80 lb. per sq. in. at sea-level, is 250 cu. ft. per min. What is the compressor capacity for 6000 ft. altitude?

Increase of drill requirements (See Table 54) at 6000 ft. altitude, 20%. Free air required by drills, 250 × 1.20 = 300 cu. ft. per min. A certain single-stage compressor which delivers 340 cu. ft. per min. of free air at sea-level will do the work; the curves, Fig. 8, show a reduction of volumetric efficiency at 6000 ft. altitude and 80 lb. per sq. in. pressure, of about 7%. The net delivery at 6000 ft. altitude would then be 340 × 0.93 = 316 cu. ft. per min. This does not mean that the volumetric efficiency of the compressor is 93% but that its volumetric efficiency at 6000 ft. altitude is 93% of the volumetric efficiency at sea-level, as previously determined by sea-level test. This all depends on actual cylinder clearance. Fig. 8 is based on an average of 7 1/4% clearance.

Table 33.—Efficiency of Air Compression at Different Altitudes

Altitude, ft.	Barometric Pressure		Vol. Efficiency of Compressor, %	Loss of Capacity, %	Decreased Power Required, %	Altitude, ft.	Barometric Pressure		Vol. Efficiency of Compressor, %	Loss of Capacity, %	Decreased Power Required, %
	In. of Mercury	Lb. per sq. in.					In. of Mercury	Lb. per sq. in.			
0	30.00	14.75	100	0	0	8,000	22.11	10.87	76	24	13.1
1,000	28.88	14.20	97	3	1.8	9,000	21.29	10.46	73	27	14.6
2,000	27.80	13.67	93	7	3.5	10,000	20.49	10.07	70	30	16.1
3,000	26.76	13.16	90	10	5.2	11,000	19.72	9.70	68	32	17.6
4,000	25.76	12.67	87	13	6.9	12,000	18.98	9.34	65	35	19.1
5,000	24.79	12.20	84	16	8.5	13,000	18.27	8.98	63	37	20.6
6,000	23.86	11.73	81	19	10.1	14,000	17.59	8.65	60	40	22.1
7,000	22.97	11.30	78	22	11.6	15,000	16.93	8.32	58	42	23.5

Table 34.—Horsepower for Compressing 1 Cu. Ft. of Free Air at Various Altitudes from Atmospheric Pressure. (F. M. Hitchcock)

Initial temperature of the air in each cylinder taken as 60° F.; Jacket-cooling not considered; allowance made for usual losses.

Altitude, ft.	Simple Compression			Two-Stage Compression				
	Gage Pressure, lb.			Gage Pressure, lb.				
	60	80	100	60	80	100	125	150
0	0.1533	0.1824	0.2075	0.1354	0.1580	0.1765	0.1964	0.2138
1,000	.1511	.1795	.2040	.1336	.1553	.1734	.1926	.2093
2,000	.1489	.1766	.2006	.1312	.1524	.1700	.1887	.2048
3,000	.1469	.1739	.1971	.1280	.1493	.1666	.1848	.2003
4,000	.1448	.1712	.1939	.1263	.1464	.1635	.1810	.1963
5,000	.1425	.1685	.1906	.1241	.1438	.1600	.1772	.1921
6,000	.1402	.1656	.1872	.1218	.1409	.1566	.1737	.1879
7,000	.1379	.1628	.1839	.1197	.1383	.1536	.1700	.1838
8,000	.1358	.1600	.1807	.1173	.1358	.1504	.1662	.1797
9,000	.1337	.1572	.1774	.1151	.1329	.1473	.1627	.1758
10,000	.1316	.1547	.1743	.1132	.1303	.1442	.1592	.1717

E.—Required the volume of free air which when compressed to 100 lb. gage at 9000 ft. altitude will be equivalent to 1000 cu. ft. of free air at sea level; also the power developed in compressing this volume to 100 lb. gage in two-stage compression at this altitude. From Table 32, multiplier is 1.356. Equivalent free air = 1000 × 1.356 = 1356 cu. ft. From Table 34, power developed in compressing 1 cu. ft. of free air is 0.1473 Hp.; hence, 1356 × 0.1473 = 199.73 Hp.

3. SIZES AND TYPES OF AIR COMPRESSORS*

CLASSIFICATION OF AIR COMPRESSORS.—Air compressors may be classified according to the mode of drive as follows (Peele's Compressed Air Plant):

STEAM DRIVEN

Straight-line.
- Single-stage, double-acting: Direct-connected to steam engine. For long belting to steam engine. For short belting to steam engine.
- NOTE.—Two-stage, straight-line compressors for steam operation are built for high-pressure work only.
- Single-stage, double-acting, portable: Direct-connected to steam engine.

Duplex, small and medium capacity.
- Single-stage / Two-stage: Direct-connected to steam engine. belting to steam engine.

Duplex, large capacity: Direct-connected to steam engine.

Turbo-compressors: Direct-connected to steam turbine.

POWER DRIVEN

Vertical.
- Single-stage, single-acting, stationary: For long belting to line shaft, gasoline engine, or electric motor. For short belt drive.
- Duplex single-stage, single-acting portable: For short belt drive.

Straight-line.
- Single-stage, double-acting: For long belt drive. For short belt drive. For chain drive.
- NOTE.—Two-stage compressors of this type are built for high-pressure work only.
- Single-stage, double-acting, portable: For short belt drive. For chain drive.

Duplex, small and medium capacity.
- Single-stage / Two-stage: For long belt drive. For short belt drive.

Duplex, large capacity.
- Single-stage / Two-stage: For long belt drive. Direct-connected to electric motor, gasoline, gas, or oil engine, or water-wheel.

* Tables considerably abridged from the originals.

NOTE.—The following tables illustrate the sizes, capacities and horsepowers of standard type forms of compressors built by well-known makers. The compressors listed have been selected as typical of their respective classes, and there is no intention to discriminate against other good makes.

Table 35.—Straight-line Power-driven Compressors

Class ER-1. Air Pressure, 10 to 125 lb. per sq. in.

(Ingersoll-Rand Co., New York)

Cylinder Dimensions, in.		Piston Displacement, cu. ft. per min.	Air Pressure, lb. per sq. in.		Brake Hp., Including Belt Loss		Cylinder, Dimensions, in.		Piston Displacement, cu. ft. per min.	Air Pressure, lb. per sq. in.		Brake Hp. Including, Belt Loss	
Diam.	Stroke		Min.	Max.	Min. Press.	Max. Press.	Diam.	Stroke		Min.	Max.	Min. Press.	Max. Press.
4	4	28	100	150	4	4.3	8	8	113	100	130	19	21
5	4	44	20	100	3.8	6.5	9	8	145	65	100	21	24
6	4	64	.....	20	.....	6.3	10	8	179	30	65	19	25
5	5	44	100	150	6.5	7	12	8	258	20	30	19	24
6	5	64	45	100	7.3	9.3	14	8	354	.....	20	.....	27
7	5	88	.....	45	.....	10	10	10	223	100	120	39	42
6	6	67	100	140	10	10	12	10	324	50	100	43	57
7	6	92	45	100	11	13	14	10	441	25	50	38	56
8	6	120	25	45	13	15	17	10	655	.....	25	.....	56
9	6	154	10	25	10	15	12	12	340	100	115	60	62
12	6	274	.....	10	.....	16	14	12	464	45	100	57	74
							17	12	688	20	45	52	80
							20	12	955	.....	20	.....	77

Stroke, in.	4	5	6	8	10	12
Revolutions per min.	500	400	350	250	250	220
Size belt wheel, in.	20×4 1/2	28×5 1/2	36×5 1/2	45×8 1/2	58×10 1/2	72×14 1/2

Similar steam-driven compressors also are built, together with those of the XPV class, having duplex, single-stage air cylinders and simple or cross-compound steam ends.

Table 36.—Straight-line, Steam-driven Compressors

Class N-SS. Air Pressures, 30 to 125 lb. per sq. in.

(Chicago Pneumatic Tool Co., Chicago)

Cylinder Dimensions, in.			R.p.m.	Displacement, cu. ft. per min.	Max. Air Pressure, lb. per sq. in.	Floor Space			
						Length		Width	
Steam	Air	Stroke				ft.	in.	ft.	in.
6	6	6	350	67	125	7	4	2	6
6	7 1/2	6	350	106	100	7	4	2	6
6	9	6	350	153	50	7	8	2	6
8	8	8	300	136	125	9	3	3	0
8	9	8	300	173	100	9	3	3	0
8	10	8	300	215	70	9	3	3	0
8	12	8	300	311	40	9	3	3	0
10	10	10	275	246	125	10	5	3	5
10	12	10	275	356	100	10	5	3	5
10	15	10	275	558	50	10	7	3	5
10	17	10	275	719	30	11	2	3	5
13 1/2	12	12	250	387	125	12	3	3	9
13 1/2	14	12	250	529	100	12	3	3	9
13 1/2	17	12	250	783	50	12	6	3	9
13 1/2	20	12	250	1085	30	13	3	3	9
15	17	14	235	857	80	13	10	4	7
15	20	14	235	1189	40	14	2	4	7

The steam cylinders of the 6-, 8-, and 10-in. stroke N Class compressors have D-slide-valves; 12-in. stroke, double-port balanced valves. The air cylinders have "simplate" disc-valves. This company also builds duplex, single-stage compressors, Class O, driven by steam, belt or direct-connected electric motor, for pressures of 30 to 70 lb. per sq. in., and displacements from 1022 to 4424 cu. ft. per min.

Table 37.—Straight-line Steam-driven Compressors

Class FR-1. Air Pressures, 10 to 125 lb. per sq. in. Steam, 80 to 175 lb. per sq. in
(Ingersoll-Rand Co., New York)

Cylinder Dimensions, in. Diam. Steam	Air	Stroke	R.p.m.	Displacement, cu. ft. per min.	Air Pressure, lb., Gage	I.Hp. in Steam Cyl.	Cylinder Dimensions, in Diam. Steam	Air	Stroke	R.p.m.	Displacement, cu. ft. per min.	Air Pressure, lb., Gage	I.Hp. in Steam Cyl.
7	6	6	350	67	80–125	9–10	12	10	10	275	245	80–125	39–46
7	7	6	350	92	55–100	11–13	12	12	10	275	355	60–100	51–62
7	8	6	350	120	30– 50	12–14	12	14	10	275	484	30– 55	45–63
7	9	6	350	153	15– 25	11–14	12	17	10	275	718	10– 25	42–63
7	12	6	350	273	10	16							
9	8	8	300	136	80–125	21–25	14	12	12	250	386	80–125	61–70
9	9	8	300	173	65–100	24–29	14	14	12	250	528	45–100	66–88
9	10	8	300	215	35– 60	24–30	14	17	12	250	781	25– 40	75–98
9	12	8	300	311	25– 30	26–31	14	20	12	250	1086	10– 20	72–95
9	14	8	300	424	10– 20	22–31							

Floor space for these compressors ranges from 8 ft. 2 in. × 2 ft. 3 in. to 14 ft. × 5 ft.

Table 38.—Straight-line, Steam-driven Compressors

Class 4A. Air Pressures, 25 to 125 lb. per sq. in.
(Pennsylvania Pump and Compressor Co., Easton, Pa.)

Cylinder Diam., in. Steam 80–120 lb.	Steam 125–175 lb.	Air	Stroke, in.	R.p.m.	Displacement, cu. ft. per min.	Air Pressure, lb. per sq. in.	I.Hp.	Floor Space Length, in.	Floor Space Width, in.
7	6	6	6	350	67	75–125	9 1/2–10	93	29
7	6	7	6	350	92	50–100	12 – 14	95	29
7	6	7 1/2	6	350	106	40– 90	12 – 15 1/2	97	29
9	8	8	8	325	148	75–125	22 – 26	114	37
9	8	9	8	325	188	50–100	23 – 30	116	37
10	9	9	9	300	195	75–125	29 – 36	130	42
10	9	10	9	300	240	50–100	30 – 40	131	42
12	10	10	10	275	244	75–125	37 – 48	143	49
12	10	12	10	275	353	50–100	45 – 61	145	49
12	10	14	10	275	485	25– 60	43 – 66	148	49
14	12	12	12	250	385	75–125	60 – 77	165	60
14	12	14	12	250	525	50–100	65 – 87	167	60
14	12	16	12	250	690	25– 60	50 – 96	167	60

Twelve other sizes are built for pressures of 10 to 50 lb. per sq. in.; also a line of belt-driven compressors, with displacements of 28 to 1085 cu. ft. per min., the larger sizes being for low pressures (25 to 50 lb. per sq. in.).

Table 39.—Belt-driven Compressors

Class WG-6. Air pressures, 50 to 120 lb. per sq. in.
(Sullivan Machinery Co., Chicago)

Cylinders, in. Diam.	Stroke	Displacement, cu. ft. per min.	R.p.m.	Air Pressure, lb. per sq. in.	Hp.	Floor Space Length ft.	Length in.	Width ft.	Width in.
6	6	68	350	120	10.5	4	10 1/2	1	10 1/2
7	6	93	350	100	12.5	5	0	1	10 1/2
8	6	122	350	50	14	5	5 1/2	1	10 1/2
8	8	139	300	120	22.5	6	3	2	4 1/2
9	8	176	300	100	25	6	4	2	4 1/2
10	8	218	300	50	25.5	6	4	2	4 1/2
10	10	250	275	120	38.4	7	5	3	0
11	10	302	275	100	48.3	7	6	3	0
12	10	360	275	100	58.2	7	7	3	0
14	10	490	275	50	60.8	7	7	3	0
12	12	377	240	120	68.5	8	10	3	11
14	12	513	240	100	90.3	8	10	3	11
16	12	670	240	50	86.5	8	11	3	11

This company also makes similar straight-line, steam-driven compressors, Class WA–6. Five other sizes are built for air pressures of 15 to 30 lb. per sq. in.

Table 40.—Duplex Belt-driven Compressors

Type TB—SD. Air pressures, 60 to 125 lb. per sq. in.
(Norwalk Co., South Norwalk, Conn.)

Cylinders, in.		R.p.m.	Displacement, cu. ft. per min.	Air Pressure, lb. per sq. in.	Brake Hp.	Floor space, in.	
Diam.	Stroke					Length	Width
6	6	350	134	125	28	74	54
7	6	350	184	100	33	79	54
8	8	350	274	125	57	94	67
9	8	300	348	100	62	94	67
10	8	300	432	70	65	95	67
10	10	275	492	100	88	109	74 1/4
12	10	275	712	100	127	109	74 1/4
14	10	275	969	60	131	111	74 1/4
12	12	250	774	100	138	124	83
14	12	250	1060	100	189	126	83
16	12	250	1390	60	188	126	83

These compressors are driven by a short belt. Five other sizes are built, for pressures of 30 to 50 lb. per sq. in.; also, 16 sizes of single-cylinder, belt-driven compressors (Type TB-S), with cylinders of the same dimensions, the displacements being one-half the above. This company also builds: A duplex, single-stage direct-motor driven compressor (Type SB-SD), with 18 × 14-in. cylinders; displacement, 1830 cu. ft. per min.; delivery pressure, 70 lb. Five sizes of the same type, with 20 × 14 to 28 × 16-in. cylinders; displacements, 2265 to 4520 cu. ft. per min.; delivery pressures, 50, 40 and 30 lb. per sq. in.; also several sizes of straight-line, single-stage, steam-driven compressors (Type STB-S).

Table 41.—Imperial Compressors with Duplex Steam and Two-stage Air Cylinders

Air pressures 15 to 100 lb. per sq. in.; steam pressures, 80 to 120 lb. per sq. in.
(Ingersoll-Rand Co., New York)

For Sea-level				For 5,000 ft. Altitude				For 10,000 ft. Altitude				For All Altitudes	
Cylinder Diam., in.			Piston Displacement, cu. ft. per min.	Cylinder Diam., in.			Piston Displacement, cu. ft. per min.	Cylinder Diam., in.			Piston Displacement, cu. ft. per min.	Stroke, in.	R.p.m.
Steam	Low-pressure Air	High-pressure Air		Steam	Low-pressure Air	High-pressure Air		Steam	Low-pressure Air	High-pressure Air			
7	11	7	246	7	12	7	293	7	13	7	344	10	225
8	13	8	412	8	14	8	479	8	15	8	550	12	225
9	15	9 1/4	549	9	16	9 1/4	625	9	17	9 1/4	706	12	225
10	17	10 1/2	823	10	18	10 1/2	923	10	19	10 1/2	1029	14	225
12	20	12 1/2	1156	12	21	12 1/2	1275	12	22	12 1/2	1400	16	200
14	22	14	1400	14	23	14	1531	14	24	14	1667	16	200
15	23	14	1723	15	24	14	1877	15	25	14	2037	20	180
16	26	16	2200	16	27	16	2373	16	28	16	2553	20	180
19	30	19	3026	19	32	19	3444	19	34	19	3890	24	155
22	35	22	4150	22	37	22	4640	22	39	22	5157	30	125

The Ingersoll-Rand XPV class are built in four forms, all duplex; single- or two-stage air, with simple steam cylinders; single- or two-stage air, with cross-compound steam cylinders. Compressors of these general types are also built for belt and direct-connected motor drive.

Table 42.—Two-stage, Fuel Oil-driven Compressors, with Differential, Single-acting Air Cylinder

Class POC-2.
(Ingersoll-Rand Co., New York)

Displacement, cu. ft. per min.	Delivery Pressure, lb. per sq. in.*		Floor space				Height		Approx. Shipping Weight, lb.
			Length		Width				
	Normal	Max.	ft.	in.	ft.	in.	ft.	in.	
354	100	125	10	6	4	4†	9	4	15,750
603	100	125	18	2	6	0	7	6 1/2	35,000
435	300	500	18	0	6	0	6	9	31,800
895	100	125	22	6	8	3	7	11	58,000

* Single-stage compressors of this type for lower air pressures are also made.
† Air cylinder is set vertical; in the other sizes it is horizontal.

Table 43.—Straight-line, Two-stage, Belt-driven Compressors

Type TB-S2T. For 110 lb. air pressure

(Norwalk Co., South Norwalk, Conn.)

Cylinder diam., in.		Stroke, in.	R.p.m.	Displacement, cu. ft. per min.	Brake Hp.	Floor space	
Low-pressure	High-pressure					Length, in.	Width, in.
10	6	8	300	219	40	133	37
12	7.5	8	300	315	58	135	37
12	7.5	10	275	355	67	156	44
14	8	10	275	490	90	156	44
15	9	12	250	600	111	175	52
16	10	12	250	700	129	175	52
18	10	12	250	885	164	177	52

The Norwalk Co. also builds: Type SB–S2X, duplex, driven by direct-connected motor, displacements, 920 to 2280 cu. ft. per min.; types TB–S2TSA (belt-driven) and STB–S2TSA (steam-driven), for pressures of 200 to 500 lb. per sq. in.; 3-stage compressors, suitable for charging compressed air locomotives, for pressures of 500 to 2000 lb. per sq. in.; and 4-stage compressors, for pressures of 2000 to 3000 lb. per sq. in.

l.—Angle-compound Compressors

Class WJ-3, belt-driven; WN-3, direct-connected to motor

(Sullivan Machinery Co., Chicago)

Cylinder Diam., in.		Stroke, in.	Net displacement, cu. ft. per min.	Hp. for 80 to 120 lb.	Floor Space			
					Length		Width	
Low-pressure	High-pressure				ft.	in.	ft.	in.
14	8 3/4	10	455– 620	72–100	8	10	4	0
16	9 3/4	10	545– 752	86–120	8	11 1/4	4	2 3/8
17	10 1/4	12	705– 941	113–153	9	11 1/2	5	11 1/2
18	11	14	869–1133	142–188	11	2 1/2	7	6 1/2
20	12	14	1011–1300	165–214	11	4 1/2	7	10 1/2
22	13	14	1224–1573	200–260	11	7	8	1
24	14 1/2	16	1482–1852	243–307	12	2	10	0

In these compressors, the high-pressure cylinder is vertical, the low-pressure cylinder horizontal. They are also made in twin form (belt-driven or direct-connected), the capacity being then doubled. The Sullivan Machinery Co. also builds nine sizes of angle-compound compressors, with displacements of 313–2553 cu. ft. per min., direct-driven by Diesel engines.

Table 45.—Horizontal, Two-stage, Corliss Cross-compound, Condensing Compressors

Steam and air pressures, 100 lb. per sq. in.

(Walker Bros., Wigan, England)

Cylinder diam., in.				Stroke, in.	R.p.m.	Capacity, cu. ft. of Free Air per min.	Floor Space			
Steam		Air					Length		Width	
High-pressure	Low-pressure	Low-pressure	High-pressure				ft.	in.	ft.	in.
13	24	20	12	30	100	1000	25	3	11	0
15	26	22	13	30	100	1250	26	10	12	2
16	28	24	15	36	90	1660	30	3	13	8
17	31	26	16	36	90	1880	30	3	13	8
18	33	28	17	39	85	2200	34	5	15	0
20	36	30	18 1/2	42	80	2600	34	6	15	6
21	38	32	20	42	80	3000	34	6	15	6
22	40	34	21	42	80	3350	37	4	16	0
23	42	36	22	42	80	3750	38	2	16	0
24	44	38	23	48	70	4200	41	4	17	6
26	47	40	24	48	70	4650	41	4	17	6
27	49	42	26	54	65	5300	44	0	17	7
29	52	44	27	54	65	5850	44	0	17	7
32	57	48	29	60	60	7100	50	5	20	3
34	61	52	32	60	60	8400	52	0	22	0

Walker Bros. also build several lines of similar single- and 2-stage compressors, condensing or non-condensing, up to displacements of 9400 cu. ft.; 2-stage, power-driven; and large 2-stage high-speed, vertical compressors.

Table 46.—Cross-compound, Two-stage Compressors

Initial steam pressure, 110–150 lb. per sq. in.; air pressure, 100 lb. per sq. in.
(Worthington Pump & Machinery Corp., Harrison, N. J.)

Cylinder Diam., in.				Stroke, in.	R.p.m.	Displacement, cu. ft. per min.	Floor Space			
Steam		Air					Length		Width	
High-pressure	Low-pressure	Low-pressure	High-pressure				ft.	in.	ft.	in.
9	15	13	8	10	265	402	12	11	7	6
10	17	15	9 1/2	12	225	549	13	10	7	9
12	20	17	10 1/2	14	225	823	15	3	8	3
14	22	20	12	14	225	1140	16	0	9	0
14	23	22	13	16	200	1400	17	3	9	6
16	25	23	14	16	200	1531	18	0	10	3
16	25	24	14	18	185	1735	18	9	11	0
17	29	26 1/2	16	18	180	2040	20	9	11	0

DUPLEX TWO-STAGE, WITH SIMPLE STEAM END

Steam pressure, 125–150 lb. per sq. in.; air pressure, 100–125 lb. per sq. in.

Steam	Air, Low-pressure	Air, High-pressure	Stroke, in.	R.p.m.	Displacement, cu. ft. per min.	Length, ft.	Length, in.	Width, ft.	Width, in.
12	16	9 1/2	14	245	790	17	0	8	6
13	18	11	16	220	1025	18	9	9	3
14	20	12	16	220	1270	19	0	9	3
15 1/2	22	13	18	200	1574	21	6	10	0
16 1/2	23	14	18	200	1721	21	6	10	3
18 1/2	26 1/2	16	21	170	2265	23	0	11	0
20	29	18	24	165	3010	26	0	12	0

The Worthington Corp. also builds duplex, two-stage compressors, belt-driven or direct-connected to synchronous motors, with displacements from 350 to 3865 cu. ft. per min.

EFFICIENCIES OF STEAM- AND ELECTRICALLY-DRIVEN COMPRESSORS.—The measurements necessary for computing efficiencies are given in Figs. 9 and 10. Under the diagrams are the ratios of these measurements, resulting in the respective efficiency and performance measurement of a steam-driven and electrically-driven air compressor. (S. B. Redfield in Compressed Air Data, 1920.)

Notation.

E = Mechanical Efficiency.
E_M = Motor Efficiency.
E_V = Indicated Volumetric Efficiency.
E_A = Actual Volumetric Efficiency.
E_S = Slippage Efficiency.
E_I = Isothermal Compression Efficiency.

For Steam-Driven Compressors:

$E =$

$I.Hp._S$ per 1000 cu. ft. per min.

per ... ft. of air deliv'd $= \frac{\text{Lb. Steam per min.} \times 1000}{C}$

For Electrically-Driven Compressors:

$B.Hp._S$

$E.Hp._M$

p. per 100 cu. ft. of air delivered per min.

For Steam- and Electrically-driven Compressors:

E_V

$I.Hp._A$ per 100 cu. ft. per min. $= E_I$.

$I.Hp._A \times 100$

$I.Hp._A$ per 100 cu. ft. of air per minute.

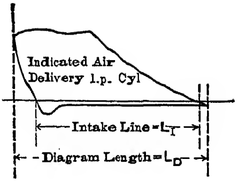
The following formulas (S. B. Redfield) exhibit in other forms the relations between the factors entering into the general expressions for compressor efficiencies. They are useful in dealing with different phases of air-compression problems.

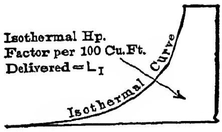
$$\frac{\text{Indicated Hp. of Air Cylinders}}{\text{Indicated Hp. of Steam Cylinders}} = \text{Mechanical efficiency, which is ordinarily } 90\%$$

to 95%. For a power-driven compressor, the denominator of this expression is the brake horsepower. The difference between the two indicated horsepowers is the power consumed in overcoming frictional resistance of the compressor.

$$\frac{\text{Length of Intake Line, Low-pressure Cyl.}}{\text{Total Length of Indicator Diagram}} = \text{Indicated volumetric efficiency.}$$

$$\frac{\text{Cu. Ft. of Free Air Delivered}}{\text{Cu. Ft. of Piston Displacement}} = \text{Actual volumetric efficiency, which usually is 65 to}$$





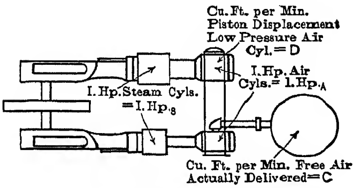


FIG. 9. Steam-driven Compressor

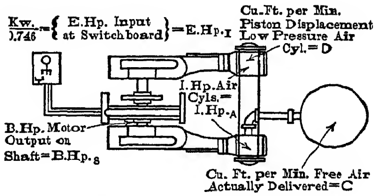


FIG. 10. Electrically-driven Compressor

This is useful for determining the actual output of air available for operating rock drills or other air-driven machines, but it does not necessarily connote economical performance of the compressor; that is, a compressor having high volumetric efficiency may have low total (overall) efficiency and *vice versa*.

$$\frac{\text{Actual Volumetric Efficiency}}{\text{Indicated Volumetric Efficiency}} = \text{Slippage Efficiency.}$$ This is the ratio of the volume of measured air delivered to the apparent volume shown by the indicator diagram, and is commonly 91 to 92%.

$$\frac{\text{Average Actual Work Done}}{\text{Maximum Capacity for Work}} = \text{Load Factor of Compressor,}$$ which otherwise may be defined as the ratio that the average power consumed bears to the maximum power during a stated period of time, as a day, month, or year.

Compression efficiency is the ratio that the theoretical power required to compress and deliver a given volume of air bears to the actual indicated horsepower developed in the air cylinder. According to the mode of compression, this is designated as Isothermal Compression Efficiency (usually 72 to 75%), or Adiabatic Compression Efficiency (say 80 to 85%). The terms Isothermal Factor and Adiabatic Factor are the theoretical horsepowers required to compress, isothermally or adiabatically, 100 cu. ft. of free air per minute to the designated pressure. For computations involving differences in barometric

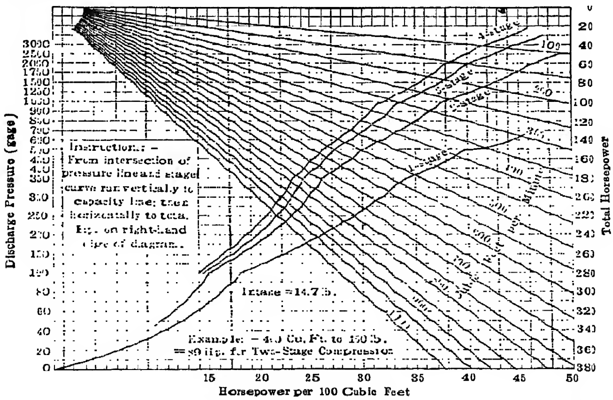


FIG. 11. Horsepower Chart for Adiabatic Compression, Perfect Intercooling. (S. B. Redfield)

pressure at altitudes it is customary to use the lower base, represented by the Isothermal Factor.

On the other hand, the Adiabatic Factor is most convenient for ordinary calculations of horsepower (see Tables 28, 29, and 30). Fig. 11 shows these relations graphically.

$$\frac{\text{Adiabatic Factor}}{\text{I.Hp. of steam cyl. per 100 cu. ft. free air per min.}} = \text{Overall or total adiabatic efficiency,}$$

which may also be expressed as the Adiabatic Compression Efficiency (see above) multiplied by the Mechanical Efficiency, and is roughly 75%. This is the most important, from the standpoint of the user of a compressor, because it is the ratio of the power represented by the volume of compressed air delivered to the power consumed in producing it. For a steam-driven compressor, the overall efficiency is as stated above; for a power-driven compressor, it is the Adiabatic Compression Efficiency × Mechanical Efficiency × Overall Efficiency of the prime mover (including belting).

STEAM CONSUMPTION IN AIR COMPRESSORS.—(O. S. Shantz, *Power*, Feb. 4, 1908).—Tables 47 and 48 show the steam required to compress 100 cu. ft. of free air to different gage pressures, in single- or in two-stage compression. They are based on the steam consumptions per I. Hp.-hr. at the head of each column. The computations are based on adiabatic compression in all the cylinders, with intercooling to atmospheric temperature in the case of two-stage compression, and a mechanical efficiency of 90%.

The tables are used as follows: A steam consumption per I. Hp. is assumed, according to the type of steam end of the compressor, steam pressure, cut-off, vacuum if condensing, etc. Under this assumed figure, on the line opposite the required air pressure are the pounds of steam per 100 cu. ft. of free air compressed. The accuracy of these tables in practice depends upon the correct assumption of the steam consumption per I. Hp. Where this is not exactly known the tables are, at best, only approximate. They are, however, useful for making comparisons of the fuel consumed by different types of compressors, thus showing the expected yearly saving by the use of, for instance, a compound as compared with a simple compressor. Thus, a straight-line compressor, with a steam consumption of 30 lb. per I.Hp., single-stage compression to 100 lb. per sq. in. gage, required 9.9 lb. of steam per 100 cu. ft. of free air. A duplex compressor, at the same steam consumption, but with two-stage air cylinders, requires 8.42 lb. of steam. A stage compressor with compound steam cylinders, non-condensing, with 26 lb. steam rating requires 7.3 lb. of steam, while a high class Corliss two-stage compound condensing compressor, using steam at high pressure, having a water rate of 17 lb. (including condenser), requires only 4.77 lb. of steam.

Table 47.—Steam Consumption of Air Compressors—Single-stage Compression

Air Gage Pressure	Steam per I.Hp.-hr., lb.												
	12	14	16	18	20	22	24	26	28	30	32	36	40
20	1.36	1.58	1.82	2.04	2.26	2.49	2.72	2.94	3.17	3.40	3.61	4.08	4.54
30	1.84	2.14	2.45	2.76	3.06	3.37	3.68	3.98	4.29	4.60	4.90	5.51	6.12
40	2.26	2.64	3.02	3.39	3.77	4.15	4.52	4.90	5.26	5.65	6.03	6.78	7.50
50	2.62	3.06	3.50	3.93	4.36	4.80	5.25	5.68	6.10	6.55	7.00	8.86	8.71
60	2.92	3.41	3.90	4.38	4.80	5.36	5.85	6.32	6.80	7.30	7.80	8.76	9.71
70	3.22	3.76	4.30	4.83	5.36	5.90	6.45	6.97	7.50	8.05	8.60	9.66	10.70
80	3.50	4.08	4.67	5.25	5.84	6.42	7.00	7.59	8.15	8.75	9.34	10.50	11.61
90	3.72	4.34	4.96	5.58	6.20	6.82	7.45	8.05	8.66	9.30	9.94	11.15	12.35
100	3.96	4.61	5.29	5.95	6.60	7.25	7.92	8.58	9.22	9.90	10.56	11.88	13.15
110	4.18	4.87	5.58	6.26	6.96	7.66	8.36	9.05	9.75	10.45	11.15	12.52	13.90
120	4.38	5.11	5.85	6.57	7.30	8.04	8.76	9.50	10.20	10.95	11.66	13.13	14.55

Table 48.—Steam Consumption of Air Compressors—Two-stage Compression

Air Gage Pressure	Steam per I.Hp.-hr., lb.												
	12	14	16	18	20	22	24	26	28	30	32	36	40
70	2.82	3.25	3.76	4.23	4.69	5.16	5.63	6.10	6.56	7.04	7.50	8.45	9.35
80	3.01	3.51	4.03	4.52	5.02	5.53	6.03	6.53	7.03	7.53	8.03	9.05	10.01
90	3.19	3.72	4.26	4.79	5.32	5.85	6.38	6.91	7.44	7.98	8.50	9.57	10.60
100	3.37	3.93	4.50	5.05	5.61	6.19	6.74	7.30	7.85	8.42	8.99	10.10	11.20
110	3.54	4.14	4.74	5.32	5.91	6.51	7.10	7.70	8.27	8.86	9.46	10.64	11.80
120	3.69	4.30	4.93	5.54	6.15	6.78	7.38	8.00	8.61	9.24	9.85	11.05	12.27
130	3.83	4.46	5.11	5.75	6.38	7.03	7.66	8.30	8.92	9.57	10.20	11.48	12.72
140	3.96	4.62	5.29	5.94	6.60	7.26	7.92	8.60	9.23	9.90	10.56	11.88	13.15
150	4.10	4.76	5.46	6 14	6.81	7.50	6.74	8.86	9.55	10.20	10.90	12.26	13.60

COST OF COMPRESSING AIR BY DIFFERENT DRIVES. (H. V. Conrad.)—In considering the purchase of a small compressor, say 50 to 100 Hp., it is desirable to consider the different sources of power and cost of fuel. Data in Tables 49, 50 and 51 give comparisons where the compressor is driven by: *a*. Gasoline engine, direct-connected or belt-driven; *b*. Steam engine, direct-connected or belt-driven; *c*. Electric motor, direct-connected or belt-driven.

In using the tables, figures suitable for local conditions must be assumed, after first

Table 49.—Cost of Gasoline, in Cents, to Compress and Deliver 100 Cu. Ft. of Free Air to 90 lb. per Sq. In., Gage

Gasoline consumption taken as one pint per brake Hp.-hour

Brake Hp. to Deliver 100 cu. ft. Free Air per min.	Price of Gasoline per Gallon, Cents							
	8	9	10	11	12	13	14	15
16	0.266	0.300	0.333	0.366	0.400	0.433	0.466	0.500
18	.300	.337	.375	.413	.450	.487	.525	.562
20	.333	.375	.416	.458	.500	.542	.584	.626
22	.366	.412	.458	.504	.550	.595	.642	.687
24	.400	.450	.500	.550	.600	.650	.700	.750
26	.433	.488	.542	.596	.650	.705	.759	.813
28	.466	.525	.583	.642	.700	.758	.816	.875
30	.500	.563	.625	.688	.750	.813	.875	.938

EXAMPLE.—If it requires 22 B.Hp. to deliver 100 cu. ft., free air per min. at 90 lb. pressure with gasoline at 22 cts. per gal., the gasoline cost per 100 cu. ft. of free air is 0.504 × 2 = 1.08 cts.

Table 50.—Cost of Coal to Compress and Deliver 100 Cu. Ft. Free Air.

Air compressed to 90 lb. per sq. in., gage. Cost of coal taken at $1.00 per ton of 2000 lb.

	Steam Required by Engine, lb. per Hp.-hr									
	24	26	28	30	32	34	36	38	40	42
	Coal per lb. per Hp.-hr. when Boiler Evaporates 7 lb. Water per lb. of Coal									
	3.43	3.714	4.00	4.286	4.571	4.757	5.143	5.428	5.714	6.00
Brake Hp. to Deliver 100 cu. ft. Free Air per min.	Cost of Coal per Hp.-hr. at $1.00 per Ton, Fractions of Cent									
	0.1715	0.1857	0.200	0.2143	0.2285	0.2378	0.2571	0.2714	0.2857	0.300
	Cost of Coal per 100 cu. ft. Free Air, Fractions of Cent									
16	0.0457	0.0495	0.0533	0.0572	0.0610	0.0634	0.0686	0.0724	0.0762	0.08
18	.0514	.0557	.06	.0643	.0686	.0714	.0772	.0814	.0857	.09
20	.057	.0619	.0667	.0714	.0762	.0793	.0857	.0904	.0952	.10
22	.0627	.0681	.0733	.0785	.0838	.0872	.0943	.0995	.1047	.11
24	.0684	.0743	.08	.0857	.0914	.0951	.1029	.1085	.1142	.12
26	.0741	.0805	.0866	.0929	.099	.103	.1114	.1175	.1238	.13
28	.0798	.0867	.0933	.10	.1066	.1109	.12	.1266	.1334	.14
30	.0855	.0929	.10	.1071	.1142	.1189	.1285	.1357	.1429	.15

Example.—If it requires 22 B.Hp. to deliver 100 cu. ft. free air per min. at 90 lb., and the engine requires 34 lb. of steam per Hp.-hr., and the boiler evaporates 7 lb. of water per lb. of coal, with coal at $6 per ton, the coal cost per 100 cu. ft. of free air is 0.087 × 6 = 0.5232 cent.

Table 51.—Cost of Electric Current, in Cents, to Compress and Deliver 100 Cu. Ft. of Free Air to 90 Lb. per Sq. In. Gage

Brake Hp. to Deliver 100 cu. ft Free Air per min.	Price of Electric Current per kw.-hr., Cents								
	1 0	1.5	2.0	2.5	3.0	3.5	4 0	4.5	5.0
	Price per Hp.-hr., Cents								
	0.75	1.125	1.5	1 875	2.25	2 625	3.0	3.375	3.75
16	0.2	0.3	0.4	0.5	0.6	0.7	0.8	0.9	1.0
18	.225	.337	.45	.562	.675	.787	.9	1.012	1.125
20	.25	.375	.5	.625	.75	.875	1.0	1.125	1.25
22	.275	.412	.55	.687	.825	.962	1.1	1.237	1.375
24	.3	.45	.6	.75	.9	1.05	1.2	1.35	1.5
26	.325	.487	.65	.812	.975	1.137	1.3	1.46	1.625
28	.35	.525	.7	.875	1.05	1.225	1.4	1.575	1.75
30	.375	.562	.75	.937	1.125	1 312	1.5	1.687	1.875

EXAMPLE.—If it requires 22 B.Hp. to deliver 100 cu. ft. of free air per min. at 90 lb. pressure, with electric current at 4 cts. per Kw.-hr. the cost of electric current per 100 cu. ft. free air is 1.1 cts

determining the horsepower required to compress and deliver 100 cu. ft. of free air per minute, and assuming that the mechanical efficiency of all the machines is about the same.

In Table 50 the steam consumption must be approximately determined. In the example, it is taken at 34 lb. per Hp.-hr. The boiler evaporation is given as 7 lb. of water per pound of coal, and any variation from this performance should appear in the calculations.

COST OF COMPRESSORS.—Previous to requesting bids from compressor builders, preliminary estimates often are made by engineers or managers, on the basis of ascertained local conditions. In this connection, Table 52 will be useful. The costs cited

Table 52.—Cost of Compressors per Cubic Feet of Free Air per Min., Compressed to 100 Pounds per Square Inch.

(Including accessories for standard equipment, Jan. 1, 1930.)

Type of Compressor	Capacity Range, cu. ft. of Free Air per min.	Cost per cu. ft. of Free Air
Straight-line, steam-driven	67– 386	$15.90–$
Straight-line, belt-driven	44– 528	6.25– 2.85
Vertical, single-cylinder, belt-driven	7.3– 44	7.85– 4.50
Vertical duplex, direct electric-driven	58– 316	11.20– 6.80
Cross-compound, 2-stage air, duplex steam, Meyer valve	246–1723	11.10– 5.95
Cross-compound, 2-stage air, duplex steam, piston valve	246–3026	11.10– 5.75
Cross-compound, 2-stage, direct electric-driven	340–3864	9.50– 4.90
Cross-compound, 2-stage, belt-driven	348–1575	5.00– 3.55
Horizontal, single-acting, 2-stage, driven by horizontal oil engine	603– 895	15.10–14.80
Vertical, single-acting, 2-stage, driven by horizontal oil engine	354	15.50
Horizontal, double-acting, duplex or 2-stage, driven by vertical, duplex, single-acting, 4-cycle gas engine *	143–1938	34.00–31.00
Portable, truck-mounted, vertical, duplex gasoline-driven	66– 310	15.90–11.30
Mine-car type, vertical, duplex, electric-driven	70– 250	18.00–12.00

* The high cost of compressors driven by gas engines is offset in part by the cheapness of fuel gas.

therein are as of Jan. 1, 1930, apply to types of compressors in general use and of the capacities (sizes) stated for each type, and include all accessories generally considered

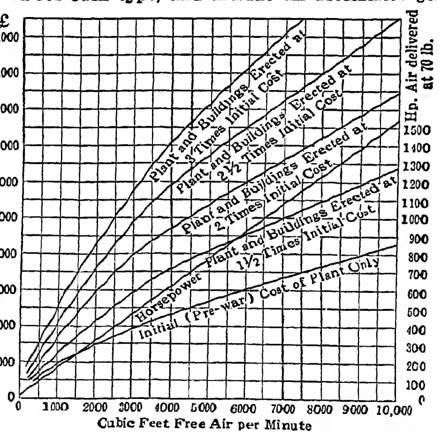


Fig. 12. Approximate Cost of Compressed Air Installations; 2-stage, with Compound Non-condensing Steam Cylinders; Air Pressure, 70 lb. (Daw, *Comp. Air Power*)

necessary for standard equipment, except such items as special governors, regulators, automatic water valves, etc.

Fig. 12 will assist in estimating approximate costs of completely installed plants (pre-war figures, 1914). Further aid generally is obtainable for cost figures existing in the great mass of engineering literature. On finding an installation similar to that desired, the unit costs applying to it can be adjusted to accord roughly with present-day

costs, by making the proper percentage increase or decrease. The results may be checked by referring to Table 52.

In asking bids from makers, the following points must be covered: Kind of service (as for drills, pumps, hoists, etc.), working pressure and volume of air required per minute, altitude above sea-level at which compressor will be used, general type of plant desired, whether steam- or belt-driven, kind and cost of fuel, electric current or water power available, and local transport facilities.

4. USES OF COMPRESSED AIR

CONSUMPTION OF AIR BY ROCK DRILLS depends on the size of drill and its physical condition, air pressure, character of rock, and the proportion of the total time actually occupied in drilling.

In hard rock, the drilling speed is slower than in soft, so that the machine makes longer continuous runs; hence, less total time is consumed in shifting and setting up for a succession of holes in hard rock, and the air consumption per unit of time is greater.

Table 53.—Cubic Feet of Free Air per Minute Consumed by One Drill at Sea Level

Gage Pressure, lb. per sq in.	Cylinder Diameter of Drills, in.												
	2	2 1/4	2 1/2	2 3/4	3	3 1/8	3 3/16	3 1/4	3 1/2	3 5/8	4 1/4	5	5 1/2
60	50	60	68	82	90	95	97	100	108	113	130	150	164
70	56	68	77	93	102	108	110	113	124	129	147	170	181
80	63	76	86	104	114	120	123	127	131	143	164	190	207
90	70	84	95	115	126	133	136	141	152	159	182	210	230
100	77	92	104	126	138	146	149	154	166	174	199	240	252

The compressor capacity required for one drill is greater than the average for a number of drills, because the delays to which each drill is subject, for setting up and shifting, changing bits, and for stoppages, make it improbable that all will be in simultaneous operation. See Peele's Compressed Air Plant, 5th Ed., pp. 384–392.

Table 54.—Multipliers to Determine Compressor Capacity for Operating from 1 to 70 Rock Drills at Different Altitudes

Altitude Above Sea Level, ft.	Number of Drills									
	1	2	3	4	5	6	7	8	9	10
	Multipliers									
0	1.0	1.80	2.70	3.40	4.10	4.80	5.40	6.00	6.50	7.10
1,000	1.03	1.85	2.78	3.50	4.22	4.94	5.56	6.18	6.69	7.30
2,000	1.07	1.92	2.89	3.64	4.39	5.14	5.78	6.42	6.95	7.60
3,000	1.10	1.98	2.97	3.74	4.51	5.28	5.94	6.60	7.15	7.81
4,000	1.14	2.05	3.08	3.88	4.67	5.47	6.15	6.84	7.41	8.09
5,000	1.17	2.10	3.16	3.98	4.80	5.62	6.32	7.02	7.61	8.31
6,000	1.20	2.16	3.24	4.08	4.90	5.76	6.48	7.20	7.80	8.52
7,000	1.23	2.21	3.32	4.18	5.04	5.90	6.64	7.38	7.99	8.73
8,000	1.26	2.27	3.40	4.28	5.17	6.05	6.80	7.56	8.19	8.95
9,000	1.29	2.32	3.48	4.39	5.29	6.19	6.96	7.74	8.38	9.16
10,000	1.32	2.38	3.56	4.49	5.41	6.34	7.13	7.92	8.58	9.37
12,000	1.37	2.47	3.70	4.66	5.62	6.57	7.40	8.22	8.90	9.73
15,000	1.43	2.57	3.86	4.86	5.86	6.86	7.72	8.58	9.30	10.15

Altitude Above Sea Level, ft.	Number of Drills								
	12	15	20	25	30	40	50	60	70
	Multipliers								
0	8.10	9.50	11.70	13.70	15.80	21.40	25.50	29.40	33.20
1,000	8.34	9.78	12.05	14.10	16.30	22.00	26.26	30.30	34.20
2,000	8.67	10.17	12.52	14.66	16.90	22.90	27.28	31.46	35.52
3,000	8.91	10.45	12.87	15.07	17.38	23.54	28.05	32.34	36.52
4,000	9.23	10.83	13.34	15.62	18.01	24.40	29.07	33.52	37.80
5,000	9.48	11.12	13.69	16.03	18.49	25.04	29.84	34.40	38.84
6,000	9.72	11.40	14.04	16.44	18.96	25.68	30.60	35.40	39.84
7,000	9.96	11.68	14.39	16.85	19.43	26.32	31.36	36.16	40.84
8,000	10.21	11.97	14.74	17.26	19.90	26.96	32.13	37.04	41.83
9,000	10.45	12.26	15.09	17.67	20.38	27.60	32.90	37.92	42.83
10,000	10.69	12.54	15.44	18.08	20.86	28.25	33.66	38.80	43.82
12,000	11.10	13.02	16.03	18.77	21.64	29.32	34.94	40.28	45.48
15,000	11.58	13.58	16.73	19.59	22.59	30.60	36.46	42.04	47.47

In soft rock, though the drilling speed is greater, there are apt to be more delays, due to "rifling" of the hole and sticking ("fitchering") of the bit; also, longer delays for piston than for hammer drills.

The figures in Table 53 are for average conditions, in moderately hard rock. For very hard rock ground and high air pressure, it is often advisable to provide, say, 20 to 25% additional compressor capacity. No allowance is made in the table for transmission losses due to pipe friction and leakage.

Example.—Required the amount of free air for operating 30 5-in. drills at 9000 ft. altitude, and gage pressure of 80 lb. per sq. in.

From Table 53, one 5-in. drill requires 190 cu. ft. of free air per min. From Table 54, the factor for 30 drills at 9000 ft. altitude is 20.38: $20.38 \times 190 = 3872$ cu. ft. free air per min. = displacement of a compressor for the above outfit under average conditions, to which must be added pipe line losses.

The size of compressor in cu. ft., required for any number of drills of all types may be found from the formula $S = M \times A \times P$, where S = size of compressor required (cu. ft.); M = a multiplier obtained from the curve, Fig. 13; A = ($5 \times$ number of piston drills $+ 4 \times$ number of leyners $+ 3 \times$ number of jackhamers and stopers); P = gage pressure at compressor, lb. per sq. in.

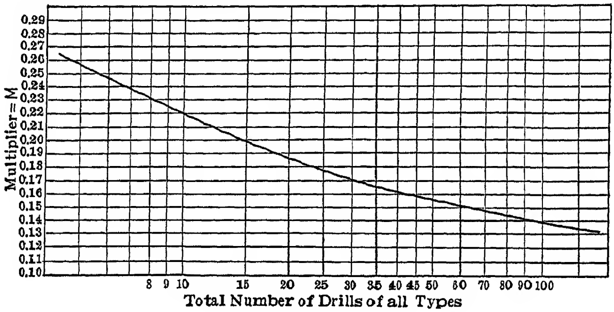


Fig. 13. Values of M

Example.—What size of compressor is required to operate one 3-in. piston drill, 3 leyners, 10 stopers and 6 jackhamers at 80 lb. per sq. in. pressure? Total number of drills = 20; from Fig. 13, for 20 drills, $M = 1.86$.

1 piston drill $\times 5 = 5$; 3 leyners $\times 4 = 12$; 10 stopers $\times 3 = 30$; 6 jackhamers $\times 3 = 18$; Total $= 65 = A$. Then $M \times A \times P = 0.186 \times 65 \times 80 = 970$, or, say 1000 cu. ft. compressor capacity.

AIR REQUIRED FOR HOISTING.—Table 55 includes the power required to hoist the skip and rope and to overcome friction. It assumes that 20% of the power is consumed in friction, that the skip weighs one-half as much as its contents, and that the traction on a horizontal mine track is 30 lb. per ton. It also provides for 500 ft. of wire rope. The air consumption is based on the assumption that a plain slide-valve engine is used, loaded to nearly full capacity. In estimating the consumption of air per hour, allowance should be made for the time the engine stands idle. If the compressor is large, doing other work also, its capacity should be based on the hourly consumption, in which case allowance is made for the time the hoist stands idle. If, however, the compressor drives the hoist alone, the hoist should be considered as running continuously. For a balanced hoist use only the unbalanced load in applying this rule. If the engine is not loaded to its full capacity, the air being throttled, it requires more air for the same work. Decreasing hoisting speed will diminish the power developed and the air consumed, but as the hoisting time is thereby lengthened, the air consumed per hoist remains unchanged. Increased speed will give a corresponding result.

COMPRESSED-AIR TABLE FOR HOISTING-ENGINES.—(Ingersoll-Rand Co. 1924.)—Table 56 gives the approximate volume of free air required for operating hoisting-engines, the air pressure at the engine being 60 lb. gage. There are so many variable conditions in the operation of hoisting-engines that accurate computations can be made only when fixed data are given. In the table, the engine is assumed to run one-half of the time for hoisting, while the compressor runs continuously. If the engine runs less than one-half the time, the volume of air required is proportionately less. The table is computed for maximum loads, which vary widely. Due to the intermittent work of a hoisting-engine, the parts resume their normal temperature between hoists, and there is little probability of freezing the exhaust-passages.

Table 55.—Air Required to Hoist One Ton (2240 Lb.) at Various Angles of Slope
(Rope Speed, 350 ft. per min.)
(F. A. Halsey)

Angle of Slope to Horizontal, deg.	Cu. Ft. of Free Air Consumed per min. of Actual Hoisting	Cu. Ft. Free Air for Each 100 ft. of Hoist (Measured on Slope)	Angle of Slope to Horizontal, deg.	Cu. Ft. of Free Air Consumed per min. of Actual Hoisting	Cu. Ft. Free Air for Each 100 ft. of Hoist (Measured on Slope)
0	11	3.14	35	520	148
5	90	25.7	40	590	168
10	100	45.6	45	650	186
15	230	65.7	50	700	200
20	320	91.2	60	790	225
25	400	114.0	75	880	250
30	470	134.0	90	900	257

Table 56.—Volume of Free Air Required for Operating Hoisting-engines
(Ingersoll-Rand Co., New York)

SINGLE-CYLINDER HOISTING-ENGINE. Air Pressure, 60 lb. per sq. in.

Diam. of Cylinder,	Stroke, in.	R.p.m.	Nominal Hp.	Actual Hp.	Wt. Lifted, Single Rope, lb.	Cu. Ft. Free Air Required
5		200	3	5.9	600	75
5		160	4	6.3	1000	80
6 1/4	8	160	6	9.9	1500	125
7	10	125	10	12.1	2000	151
	10	125	15	16.8	3000	170
	12	110	20	18.9	5000	238
10	12	110	25	26.2	6000	330

DOUBLE-CYLINDER HOISTING-ENGINE. Air Pressure, 60 lb. per sq. in.

Diam. of Cylinder,	Stroke, in.	R.p.m.	Nominal Hp.	Actual Hp.	Wt. Lifted, Single Rope, lb.	Cu. Ft. Free Air Required
5	6	200	6	11.8	1,000	150
5	8	160	8	12.6	1,650	160
6 1/4	8	160	12	19.8	2,500	250
7	10	125	20	24.2	3,500	302
8 1/4	10	125	30	33.6	6,000	340
8 1/2	12	110	40	37.8	8,000	476
10	12	110	50	52.4	10,000	660
12 1/4	15	100	75	89.2	....	1125
14	18	90	100	125	....	1587

OPERATION OF MINE PUMPS BY COMPRESSED AIR.—The advantages of compressed air over steam for mine pumps are: Absence of condensation and radiation losses in pipe lines; high efficiency of compressed-air transmission; ease of disposal of exhaust; absence of danger from broken pipes. The disadvantage is that, at a given initial pressure without reheating, a cylinder full of air develops less power than steam. The power end of the pump should be designed for the use of air, with small clearances and proper proportions of air and water ends, for the head under which the pump is to operate. Wm. Cox (*Comp. Air Mag.*, Feb., 1899) states the relations of simple or single-cylinder pumps to be $A/W = {}^1/_2 h/p$, where A = area of air cylinder, sq. in.; W = area of water cylinder, sq. in.; h = head, ft.; and p = air pressure, lb. per sq. in. The volume V of free air in cu. ft. per min. to operate a single-cylinder pump, working without cut-off is

$$V = 0.093W_2 /(hG \div P),$$

where W_2 = volume of 1 cu. ft. of free air corresponding to 1 cu. ft. of air at pressure P; G = gal. of water per min.; P = gage pressure of air, lb. per sq. in.; and h = head, ft. This formula is based on a piston speed of 100 ft. per min., and 15% is added to the volume of air to cover losses. The useful work and efficiency of a pump using air at full pressure is greater at low air pressures than at high. Hence, as high-pressure air is required for drills, etc., and as the air for pumps is drawn from the same main, the air must either be wire-drawn into the pumps, or a reducing valve be inserted between the pump and main. Wire-drawing causes a low efficiency in the pump. If a reducing valve is used, the increase of volume will be accompanied by a drop in temperature, so that the full value of the increase is not realized. Part of the lost heat may be regained by friction, and from external sources. Efficiency may be increased by underground receivers for the expanded air before it passes to the pump. If the receiver be of ample size, the air will regain nearly its normal temperature, the entrained moisture will be deposited and freezing troubles avoided. By compounding the pumps, the efficiency may be increased to about

25%. In simple pumps it ranges from 7 to 16%. For further information on this subject see Peele's Compressed-Air Plant, 5th ed., 1923.

Table 57.—Cubic Feet of Free Air Required for Sinking Pumps (Sea Level)
Pumps working continuously at listed capacity, 10% allowed for slippage

Size, in.	Capacity, gal. per min.	Piston Speed, ft. per min.	Air Pressure at Pump, lb. per sq. in.				
			50	60	70	80	100
6 × 3 × 7	27	75	71	81	93	104	126
6 × 6 & 4 × 7	48	67	63	73	83	92	112
8 × 4 × 12	83	128	213	246	278	312	378
10 × 5 × 13	130	128	333	384	435	486	590
12 × 5 × 13	130	128	480	554	628	702	850
12 × 7 × 13	206	103	385	444	503	563	682
14 × 7 × 13	206	103	525	605	686	768	930
18 × 9 × 16	247	75	635	732	830	930	1120
16 × 10 1/2 × 16	427	95	635	732	830	930	1120
18 × 10 1/2 × 16	427	95	800	922	1050	1170	1420

MULTIPLIERS FOR ALTITUDE

Altitude, ft.	1000	2000	3000	4000	5000	6000
Multiplier	1.04	1.08	1.12	1.165	1.21	1.255

Table 58.—Compressed-air Table for Reciprocating Pumps
Reasonable allowances have been made for loss due to clearance in pump and friction in pipe.

Height in Feet to which the Water is to be Pumped	Ratio of Diameter of Air Cylinder (the Steam Cylinder) to Water Cylinder											
	1 to 1		1 1/2 to 1		1 3/4 to 1		2 to 1		2 1/4 to 1		2 1/2 to 1	
	Air Pressure at Pump, lb. per sq. in.	Cu. Ft. Free Air per gal. Water	Air Pressure at Pump, lb. per sq. in.	Cu. Ft. Free Air per gal. Water	Air Pressure at Pump, lb. per sq. in.	Cu. Ft. Free Air per gal. Water	Air Pressure at Pump, lb. per sq. in.	Cu. Ft. Free Air per gal. Water	Air Pressure at Pump, lb. per sq. in.	Cu. Ft. Free Air per gal. Water	Air Pressure at Pump, lb. per sq. in.	Cu. Ft. Free Air per gal. Water
25	13.75	0.21	......	......	......	......	......	......	......	......	......	......
50	27.5	0.45	12.22	0.65	......	......	......	......	......	......	......	......
75	41.25	0.60	18.33	0.80	13.75	0.94	......	......	......	......	......	......
100	55.0	0.75	24.44	0.95	19.8	1.14	13.75	1.23	......	......	......	......
125	68.25	0.89	30.33	1.09	22.8	1.24	17.19	1.37	13.75	1.533	......	......
150	82.5	1.04	36.66	1.24	27.3	1.30	20.63	1.52	16.5	1.68	13.2	1.79
175	96.25	1.20	42.76	1.39	32.1	1.54	24.06	1.66	19.25	1.83	15.4	1.98
200	110.0	1.34	48.88	1.53	36.66	1.69	27.5	1.81	22.0	1.97	17.6	2.06
225	......	......	55.0	1.68	41.25	1.84	30.94	1.96	24.75	2.12	19.8	2.104
250	......	......	61.11	1.83	45.83	1.99	34.38	2.11	27.5	2.26	22.0	2.34
300	......	......	73.32	2.12	55.0	2.30	41.25	2.40	33.0	2.56	26.4	2.62
350	......	......	85.4	2.41	64.16	2.50	48.13	2.69	38.5	2.85	30.8	2.88
400	......	......	97.66	2.70	73.33	2.88	55.0	2.98	44.0	3.15	35.2	3.18
450	......	......	......	......	82.5	3.19	61.88	3.28	49.5	3.44	39.6	3.36
500	......	......	......	......	......	......	68.75	3.57	55.0	3.73	44.0	3.23

free air. Pressure required on the pump is in the column directly opposite.

EXAMPLE.—The ratio between cylinders being 2 to 1, it is required to pump 100 gal. to a height of 250 ft. Opposite 250 at ratio 2 to 1, is 2.11; 2.11 × 100 = 211 cu. ft. free air. Pressure required, 34.38 lb. delivered at pump piston.

5. REHEATING COMPRESSED AIR

INCREASE OF PRESSURE due to reheating of compressed air is shown by the chart Fig. 14, the volume being constant. The air is assumed to be heated from initial temperatures of 0°, 32°, 60°, and 100° F.

INCREASE OF VOLUME due to reheating is shown by the chart, Fig. 15. The air is assumed to be heated from the initial temperatures of 0°, 32°, 60°, and 100° F., the pressure remaining constant. The relative volume at any temperature is given by the height of the vertical line corresponding to that temperature, the height from *AB* to *CD* representing one volume, and each horizontal line above *CD* indicating successively an added one-tenth volume. The original volume is doubled at the line *EF*. Figures below *AB* indicate the sensible temperatures; those above *EF* indicate the corresponding absolute temperatures.

TEMPERATURE OF INTAKE AIR, for economy, should be as low as possible. Hence, it is best taken, not from the warm engine room, but from outside of the building, preferably at a point shielded from direct rays of the sun, and free from dust. For every 5° F. lower temperature, the gain is approximately 1%.

Table 59.—Effect of Intake Temperature on Efficiency and Capacity of Compressors

Unit Capacity and Efficiency Assumed at 60° F.

Initial Temp.		Relative Capacities and Efficiencies	Initial Temp.		Relative Capacities and Efficiencies	Initial Temp.		Relative Capacities and Efficiencies
Deg. F.	Deg. Abs.		Deg. F.	Deg. Abs.		Deg. F.	Deg. Abs.	
−20	441	1.18	40	501	1.040	110	571	0.912
−10	451	1.155	50	511	1.020	120	581	.896
0	461	1.13	60	521	1.000	130	591	.880
10	471	1.104	70	531	0.980	140	601	.866
20	481	1.083	80	541	.961	150	611	.852
30	491	1.061	90	551	.944	160	621	.838
32	493	1.058	100	561	.928			

The saving shown in Table 59 is obtained at practically no cost. The area of the intake duct should be at least half the area of the compressor cylinder, and to reduce frictional resistance it is well to increase the area by 1 sq. in. for every 10 ft. in length. Tile pipe is a good material for intake ducts, due to its low conductivity.

Table 60.—Relative Volume of Air Discharged by a Compressor of an Intake Capacity of 1000 Cu. Ft. per Min., Reduced to Cu. Ft. at Atmospheric Pressure and 62° F.

Temperature of intake, deg. F.	0	32	62	75	80	90	100	110
Relative volume discharged, cu. ft.	1135	1060	1000	975	966	949	932	915

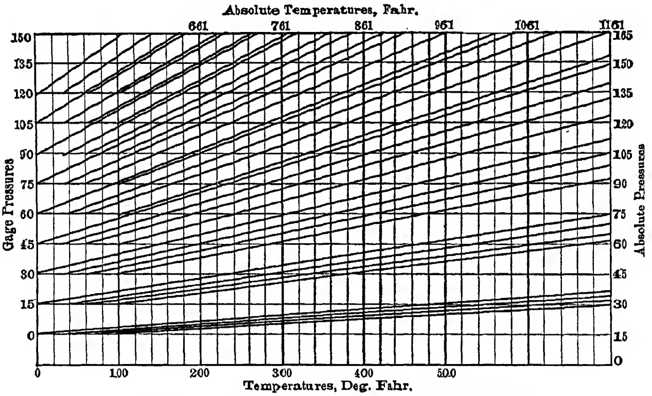


FIG. 14. Increase of Pressure Due to Reheating

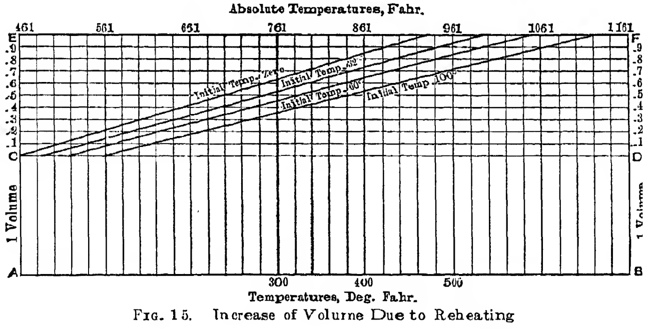


FIG. 15. Increase of Volume Due to Reheating

AFTER-COOLING.—By passing the air from the compressor through an after-cooler and reducing its temperature approximately to normal, much of the moisture will be condensed and may be removed before it can enter the pipe line, since compressed air has less capacity for moisture than air at atmospheric pressure. An after-cooler gives better results with a moderate quantity of cold water, rather than with an unlimited supply at higher temperature.

With the amounts of water given in Table 61 for ordinary working conditions, the temperature of the air leaving the inter-cooler or after-cooler should be within 20° F. of the temperature of the water used, when the cooler takes air from a two-stage compressor (100 lb. per sq. in. discharge pressure), and the initial water temperature is about 70° F.

Table 61.—Cooling Water Required for After-coolers, Inter-coolers and Cylinder Jackets
Gal. per 100 cu. ft. of free air for different water temperatures

Temperature of Water, deg., F. =	60	70	80	90
After-cooler or inter-cooler separate (80–100 lb. 2-stage compression)	2.5	3	3.5	4
Inter-cooler and jackets in series (80–100 lb., 2-stage compression)	2.9	3.4	4.0	4.5
After-cooler for 80–100 lb., single-stage compression	4.0	4.5	5.2	6.0
Both low- and high-pressure jackets with water supply separate from inter-cooler (80–100 lb., 2-stage compression)	0.085	1.0	1.2	1.4
Jacket for single-stage compression, 40 lb. air pressure	0.5	0.6	0.7	0.9
Jacket for single-stage compression, 60 lb. air pressure	0.6	0.7	0.8	1.0
Jacket for single-stage compression, 80 lb. air pressure	0.7	0.8	0.9	1.1
Jacket for single-stage compression, 100 lb. air pressure	0.8	0.9	1.0	1.2

Table 62.—Air After-cooler Capacities. (Air Pressure, 80 to 100 lb. per sq. in.)

Cooling Surface, sq. ft.	Maximum Capacity in cu. ft. Free Air per Minute, with Cooling Water at Different Temperatures							
	60° F.		70° F.		80° F.		90° F.	
	1-Stage	2-Stage	1-Stage	2-Stage	1-Stage	2-Stage	1-Stage	2-Stage
Vertical After-cooler								
150	690	1,160	630	1,035	570	960	510	850
200	920	1,540	840	1,380	755	1,280	680	1,140
300	1,400	2,310	1,260	2,070	1,140	1,920	1,020	1,700
500	2,300	3,850	2,100	3,450	1,900	3,200	1,700	2,850
750	3,450	5,750	3,150	5,400	2,850	4,800	2,550	4,270
1,000	4,650	7,700	4,200	6,900	3,780	6,400	3,400	5,700
2,000	9,290	15,400	8,400	13,800	7,550	12,800	6,800	11,400
Horizontal After-cooler								
50	230	385	210	345	190	320	170	285
150	690	1,160	630	1,035	570	960	510	850
200	920	1,540	840	1,380	755	1,280	680	1,140
300	1,400	2,310	1,260	2,070	1,140	1,920	1,020	1,700
500	2,300	3,850	2,100	3,450	1,900	3,200	1,700	2,850
750	3,450	5,750	3,150	5,400	2,850	4,800	2,550	4,270
1,000	4,650	7,700	4,200	6,900	3,780	6,400	3,400	5,700
2,000	9,200	15,400	8,400	13,800	7,550	12,800	6,800	11,400

SIZE OF AIR RECEIVER FOR GIVEN CAPACITY OF AIR COMPRESSOR.—1. Determine the maximum capacity of the compressor in free air per min., or piston displacement per min. 2. Compute the volume occupied by this air at the working pressure; this is the required volume of the receiver.

This computation gives approximately the minimum size of receiver necessary; a larger size is preferable. The receiver should be sufficiently large to prevent fluctutations in pressure and to retain the air long enough for it to cool and deposit part of its moisture.

6. CENTRIFUGAL COMPRESSORS

The theory underlying the design and operation of turbo- or centrifugal compressors, and their fundamental mechanical features (shape and number of impellers in series, and number of stages), are the same as for centrifugal fans and pumps. See p. 1–55. The inlets may be single or double, and axial or radial, according as the air or gas enters parallel to or at right angles to the shaft on which the impellers are mounted; for large volumes, they are generally axial. To minimize eddies, the air issuing from the impellers passes into the casing in streams between stationary vanes or shrouds. Means are provided to prevent leakage of air from one impeller or stage to another, or to the

atmosphere. Rotative speeds are generally from 3000 to 6000 r.p.m.; discharge velocity of the air, rarely less than 300 ft. per sec. In passing from each impeller into the discharge passages leading to the next in the series, the air velocity is converted into head or pressure, each impeller adding an increment of say $2\,1/2$ to 6 lb. per sq. in.

Centrifugal compressors designed for pressures up to about 5 lb. per sq. in. are called blowers, and have a single impeller; for higher pressures a series of impellers is used. For pressures above, say, 30 or 35 lb. per sq. in., stage compression is customary, several series of impellers being mounted on the same driving shaft, each series having its own casing. The last impeller of one series delivers air to the first of the next series. By four stages, the pressure may be built up to 150 lb. per sq. in. or more. Multi-stage compressors have intercoolers between the stages, and water-jackets for the individual impellers. In the first impellers, where the pressure is low, the temperature rises rapidly, notwithstanding the water-jackets, but, as the density of the air increases, cooling becomes more effective, with good isothermal efficiency. The volume of cooling water, at 70° F., is roughly 165 gal. per min. per 1000 Hp., sometimes estimated at 3000 gal. per hr. per 1000 cu. ft. of free air compressed per min. to 100 lb. per sq. in.

CAPACITY ranges from small units of 250–700 cu. ft. per min. up to about 60,000 cu. ft. of free air per min.; delivery pressures are, roughly, 1 to 35 lb. per sq. in. for cupola and blast-furnace blowers, to 100 to 170 lb. per sq. in. for multi-stage compressors. A number of high-pressure units, of 12,000 to 13,000 Hp., are used at central distributing plants in the South African gold fields (see Peele's Compressed Air Plant, 5th ed., pp. 97, 506–511). In general, compressors delivering 5000 to 12,000 cu. ft. of free air per min. at pressures of 85 to 120 lb. per sq. in. gage, should run 3500–3800 r.p.m.; those for, say, 2500 cu. ft. per min. at 70 to 85 lb. per sq. in., and for 1750 cu. ft. per min. at 60 lb. per sq. in. should run at 4200–4500 r.p.m.

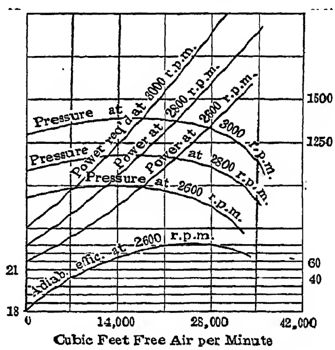


Fig. 16. Performance of 3-stage Centrifugal Compressor

Among the principal makers of centrifugal compressors are Ingersoll-Rand Co., N. Y. City; General Electric Co., Schenectady, N. Y.; Westinghouse Electric, Thomson-Houston and General Electric Cos., Great Britain; and Sulzer Bros., Winterthur, Switzerland. An instruction book of 60 pp., issued by the General Electric Co., Schenectady, N. Y., gives details of construction of multi-stage compressors, with directions for installation and operation.

APPLICATIONS.—1. Power transmission in general. 2. For blast- and other furnaces. When air must be supplied to these in uniform volume against a variable resistance, the speed of rotation is adjusted to air demand by a special governor, controlled by the varying air velocity in the compressor inlet. The weight of oxygen supplied to the furnace is thus quite accurately regulated. A nearly constant pressure also can be maintained, even with large fluctuations in volume of air used. For example, a compressor running normally at 4600 r.p.m., and 35 lb. per sq. in. pressure, showed only 2 to 3 lb. variation for outputs ranging from 1500 to 6000 cu. ft. of free air per min. Metallurgical service requires no higher pressures than 35 to 40 lb. per sq. in. 3. Exhauster service, for conveying comminuted materials, as cement, sawdust, ash, products of the flotation ore concentration process, etc. 4. Agitation of liquids, as in oil refining and treatment of sewage at disposal plants. 5. Gas making, ventilating, cooling and sundry manufacturing uses. 6. A low-pressure centrifugal may be used to deliver air at, say, 30 lb. per sq. in. to a high-pressure reciprocating compressor. This reduces about 60% the volume handled by the latter, decreases its size and increases the total efficiency.

ADVANTAGES are, for the larger units, lower initial and maintenance costs than for a reciprocating compressor of equal capacity, less floor space, freedom from vibration, non-pulsating discharge, good efficiency under quite wide load variation. Furthermore, as there is no internal lubrication, the discharge air is free from oil vapors, often a desirable condition. For the same reason, there can be no explosions in receivers or piping from mixtures of air and oil vapor, which may occur with reciprocating compressors.

Except for very low pressures, it is not generally advisable, in point of first cost, to adopt centrifugal compressors for capacities smaller than 2200–2500 cu. ft. of free air

per min. Also, except for very large units, like those of the South African plants already cited, reciprocating compressors are best for air pressures above, say, 70–85 lb. per sq. in. The air end of centrifugal compressors delivering more than 6000 cu. ft. of free air per min. should work under full load at 70 to 80% efficiency; those between 3000 and 6000 cu. ft., 65 to 70% efficiency at full load, and below 50% at half load. In general, the efficiency increases with the number of stages. Fig. 16 shows the performance of a three-stage compressor at different speeds.

COMPRESSOR CONSTANT, K may be expressed as $K = QN^2 \div \sqrt{p^3}$, where Q and p are, respectively, the volume and mean effective pressure of the air, and N = r.p.m. For different combinations of Q and p, the efficiency is practically unaltered. Compressors having the same constant are said to be *similar;* that is, their impeller and discharge vane angles are respectively equal, while the linear dimensions of these parts are in the same ratio as the impeller diameters. At the same speed, the volumes delivered by similar compressors vary as the cubes, the pressures as the squares, and the shaft horsepowers as the fifth powers of their diameters. At the same r.p.m. and constant pressure, the volume and horsepower vary as the square of the impeller diameter. In making tests, the coefficients found from observations are plotted as characteristic curves, like those of centrifugal pumps (see p. 2–80), from which the power, pressure, and efficiencies are computed for any given volume.

The power required by a centrifugal compressor, at from about one-half load to 25% overload, varies approximately with the load. If the demand for air ceases while the compressor is at full speed, the casing will heat, but no other hurtful strains will occur. The theoretical Hp. to compress adiabatically and deliver 100 cu. ft. of free air per min. is

$$\text{Hp.} = 1.511p_1\left[\left(\frac{p_2}{p_1}\right)^{0.29} - 1\right] \quad . \quad . \quad . \quad . \quad . \quad . \quad . \quad . \quad [16]$$

where p_1 and p_2 are initial and delivery pressures, lb. per sq. in. absolute (see Theory of Air Compression, p. 1–29). For the largest range of pressures likely to occur in practice, the ratios of isothermal to adiabatic work, from which approximate actual horsepower for a given delivery pressure can be estimated, are as follows (L. C. Lowenstein):

p_2/p_1..........	1.5	2.0	2.5	3	4	5	6	7	8	9	10
Isoth. ÷ Adiab.	0.94	.904	.875	.85	.812	.784	.763	.744	.728	.715	.703

DRIVING UNIT.—This is generally a direct-connected, high-pressure or mixed-pressure steam turbine of two or more stages, or occasionally, an electric motor. Gearing is sometimes used for small units. A mixed-pressure turbine is designed to carry its rated load with either live or low-pressure (exhaust) steam, or a combination of the two. Its operation is determined by the available volume of exhaust-steam, through control of a special governor. Sometimes, live steam is used in the first stage of the turbine, low-pressure steam, as from the exhaust of another engine, in the second stage. When the steam supply is used also for a heating system, subject to large fluctuations, the bleeder type of turbine is advisable. Boiler steam is expanded in the turbine to a predetermined pressure, at which part of the steam passes to the heating system, the remainder expanding further in the lower stages. A bleeder turbine can work at high pressure, if no steam is wanted for heating. According to a test on a 1000-Hp. mixed-pressure turbine, driving a compressor delivering 4400 cu. ft. of free air per min., compressed to 99 lb. per sq. in., there were used per 100 cu. ft. free air, 5.05 lb. of live steam at 85 lb. per sq. in., and 7.98 lb. exhaust steam, entering at 15.6 lb. per sq. in. and discharged at 1.14 lb. per sq. in.

INFORMATION REQUIRED BY MAKERS for submitting specifications and estimates includes:

For Compressor: Character of service, *i.e.*, for compressor or blower; nature of gas, if other than air; whether it is saturated, dry, or corrosive; kind of service; altitude of installation; intake temperature and volume (cu. ft. per min.); normal, minimum and maximum discharge pressure; hand or automatic control; if automatic, whether regulation is for constant volume or pressure.

For Driving Unit: If turbine, initial steam pressure and temperature; exhaust and back pressure; if of bleeder type, pounds of steam bled per hour, with its corresponding pressure. If electric motor, type, as induction, synchronous or direct-current; voltage, phase and cycles; squirrel-cage or slip ring type; starting equipment, manual or automatic.

7. HYDRAULIC AIR COMPRESSION

HYDRAULIC AIR COMPRESSOR.—When air in small bubbles is intimately mixed with water, the water breaks into foam, through which the bubbles tend to rise and escape. But if the mixed air and water be drawn downward by a strong falling current, as in a vertical pipe, the air is compressed. Then if, after reaching the depth and head

of water-column necessary to produce the compression desired, the direction of flow be changed and the velocity diminished, the bubbles of compressed air will be liberated and may be collected in a suitable chamber. The air pressure in this chamber corresponds to the effective head of water, that is, its depth below the level at the outflow. As the bubbles are thoroughly disseminated through the water during its descent, the total cooling surface is large and isothermal compression results. The moisture-carrying capacity of the air is therefore smaller than if it were compressed adiabatically. During compression the percentage of moisture in each globule of air increases until the point of saturation is reached; on further compression, the excess moisture is deposited, so that when re-expanded the air is relatively dry.

A typical apparatus consists of a vertical down-flow or compressing pipe, having a receiving tank at the top and a separating chamber at the bottom. Partly immersed in the receiving tank is a headpiece, containing a number of small vertical pipes, open to the air at the top and closed at the bottom. Inserted in these are numerous short horizontal tubes, directed towards the middle of the down-flow pipe. As the entering water passes among the horizontal tubes, air is entrained, carried as bubbles into the down-flow pipe and is thus compressed. On reaching the separating chamber at the bottom, the mixture of air and water is deflected radially outward by a cone and horizontal baffle plate, and, as velocity of flow decreases, the compressed air is liberated and accumulates in a receiving or storage chamber, from which it is drawn off as required. Finally, the water is discharged into an ascending pipe to a tail-race; the effective compressing head being equal to the difference in level between the water in the air storage chamber and that in the tailrace.

A number of these plants have been installed by the Taylor Hydraulic Air Compressing Co., Montreal. The first was at the Magog Cotton Mills, Prov. Quebec (1896). The principal dimensions are: Supply penstock, 60 in. diam.; supply tank, 8 ft. diam., 10 ft. high; air inlets (feeding numerous small tubes), 34 2-in. pipes; down-flow pipe 44 in. diam., increasing at lower end to 60 in. diam.; separating chamber 16 ft. diam.; total depth of shaft below normal level of head water, about 150 ft.; motive head, between water levels in receiving tank and tailrace, about 22 ft.; air discharge pipe, 7 in. diam.

Table 63.—Tests of Hydraulic Air Compressors at Magog Cotton Mills *

Test No.	1	3	4	5	7	8	10
Flow of water, cu. ft. per min.	3772.	3628.	4066.	4292.	4408.	4700.	5058.
Available head in ft.	20.54	20.00	20.35	19.51	19.93	19.31	18.75
Gross water-Hp.	146.3	136.9	156.2	158.1	165.8	171.4	179.1
Cu. ft. of air, at atmospheric pressure per minute	864.	901.	967.	1148.	1091.	1103.	1165.
Pressure of air at comp., lb.	51.9	53.7	53.2	53.3	53.7	52.9	53.3
Effective work in compressing, Hp	83.3	88.2	94.3	111.74	107.	106.8	113.4
Efficiency of compressor, %	56.8	64.4	60.3	70.7	64.5	62.2	63.3
Temp. of external air, deg. F.	68.3	57.7	66.4	65.2	59.7	65.	64.2
Temp. of water and compressed air, deg. F	66.	65.5	66.4	66.5	67.	66.5	66.
Ratio of water to air, volumes.	4.37	4.03	4.20	3.74	4.04	4.26	4.34
Moisture in external air, percent of saturation	61.	77.5	71.	68.	90.	60.5	63.
Moisture in comp. air, percent of saturation	51.5	44.	38.5	35.	29.	31.2	30.

* Tests 1, 4, and 7 were made with the original air inlets; 1, 5, 8 and 10 with the inlets increased by 15 3/4-in. pipes, and 3, 6, 9 and 11 with the inlets increased by 30 3/4-in. pipes. Tests 2, 6, 9 and 11 are omitted here. They gave, respectively, 55.5, 61.3, 62, and 55.4% efficiency.

Tests: (1) Three tests were made at different rates of flow of water; (2) Four tests at different rates of flow, the air inlet tubes being increased by 30 3/4-in. pipes; (3) Four tests, with the inlet tubes being increased by 15 3/4-in. pipes (see Table 63).

The water was measured by a weir, and the compressed air by meters.

Test 1, with flow about 3800 cu. ft. per min., showed a decided advantage by the use of 30 3/4-in. extra air inlet pipes. Test 5 shows, when flow of water is about 4200 cu. ft. per min., that the economy is highest when only 15 extra air tubes are employed. Tests 8 and 9 show, when flow is about 4600 cu. ft. per min., that there is no advantage in increasing the air-inlet area. Tests 10 and 11 show that a flow of 5000 or more cu. ft. of water is in excess of the capacity of the plant.

Summary: (1) The most economical rate of flow of water in this installation is about 4300 cu. ft. per min. (2) This plant has shown an efficiency of 70.7% under such a flow. (3) The compressed air contains only from 30 to 20% as much moisture as does the atmosphere. (4) The air is compressed at the temperature of the water.

Using an old Corliss engine, without changes in the valve gear, there was recovered 81 Hp. This would represent a total efficiency of work recovered from the falling water, of 51.2%. When the compressed air was preheated to 267° F. before being used in the engine, 111 Hp. was recovered,

using 115 lb. of coke per hour, which would equal about 23 Hp. The efficiency of work recovered from the falling water and the fuel burned would be, therefore, about 61 1/2%. Other hydraulic air-compressor plants are:

	Peterboro, Ont.	Norwich, Conn.	Cascade Range, Wash.	Victoria Mine, Mich.	Cobalt, Ont.
Head of water..........	14 ft.	18 1/2 ft.	45 ft.	70.5 ft.	54 ft.
Gage pressure...........	25 lb.	85 lb.	85 lb.	113 lb.	120 lb.
Diam. of shaft..........	42 in.	24 ft.	....	....	....
Diam. compressor pipe...	18 ft.	13 ft.	3 ft.	5 ft.	8 1/2 ft.
Depth below tailrace.....	64 ft.	215 ft.	....	270 ft.	276 ft.
Horsepower.............	....	1365	200	4000	5500

In the Cascade Range plant there is no shaft, as the apparatus is constructed against the vertical walls of a canyon. The upflow pipe is 4 ft. 9 in. diam.

A description of the Norwich plant is given by J. Herbert Shedd (*Compressed Air*, April, 1906). The shaft, 24 ft. diam., is enlarged at the bottom into a chamber 52 ft. diam., leading to an air reservoir 100 ft. long, 18 ft. wide and 15 to 20 ft. high. The downflow pipe is 14 ft. diam. A 16-in. main conveys the air 4 miles to Norwich, where it is used in engines in several establishments.

FANS AND BLOWERS *

By Robert Thurston Kent

1. CENTRIFUGAL

The essential parts of a centrifugal fan are: A rotating wheel or impeller to set up centrifugal action in the air or gas to be moved; a housing, usually spiral, to convert part of the kinetic energy in the air to static or potential energy to overcome friction, and also to connect the fan outlet to the discharge duct or chamber; an inlet connecting the wheel inlet with the duct or chamber being evacuated, and also directing the air into the wheel; a shaft and bearings to carry the wheel, and means to rotate the shaft.

TYPES OF FANS.—Radial Fans, the blades lying on radii of the wheel; number of blades 4 to 24, usually 6 to 12.

Forward Curved Blades or Multiblade Fans, the blade being concave in the direction of rotation; number of blades 48 to 64.

Backward Curved Blade Fans, the blades being inclined backward, and convex in the direction of rotation; number of blades, 8 to 12. Operating characteristics of a fan depend largely on the shape of blade. See Fig. 1. All three types have been developed to about the same maximum mechanical efficiency, approximately 80%, determined by the minimum losses of bending the air stream through the wheel, and of air, skin and journal friction. Each type is made single or double inlet, single, double or fractional width, and light, medium or heavy construction, depending on the use of the fan.

For the same work, relative speed varies widely in the three types. To move *standard air, i.e.* air at 68° F. and 29.92 in. Hg, and 50% relative humidity, against 2.5 in. static pressure, the average tip speeds in ft. per min. will be: Radial fan, 5500; forward curved, 4100; backward curved, 7700. At uniform tip speed the pressure produced by the three types also varies widely. Maximum tip speed with standard materials is about 24,000 ft. per min. At this speed and single-stage operation on standard air, expansion and compression being neglected, the pressures produced are: Radial, 48 in. = 1.73 lb. per sq. in.; forward curved, 86 in. = 3.1 lb. per sq. in.; backward curved, 24 in. = 0.87 lb. per sq. in. The number of blades affects speed characteristics more than it does efficiency. The following is the relative rotative and tip speeds of the three types: Radial, lowest rotative, medium tip; forward curved, medium rotative, lowest tip; backward curved, highest rotative, lowest tip. In the forward curved fan the rotative speed is higher than in the radial, despite low tip speed, since for a given duty wheel diameter is lower.

For low volumes at high pressure the radial fan is more practical. For very large volumes at low pressure, either the forward or backward curved fan is preferable. The backward curved fan for ventilation work is best adapted to direct-connected motors, as it can use smaller frame motors than the forward curved type. The radial fan is usually used with steam engine drive.

The Sources of Pressure in a centrifugal fan are the centrifugal force due to rotation of the column of air enclosed between each pair of blades, and the energy in the air due to its velocity when leaving the periphery of the fan wheel. Of two straight-blade fans, of equal diameter and peripheral velocity, that having the longer blades will develop the greater pressure, due to greater centrifugal force.

SIZE OF INLET.—The number of blades has a direct relation to size of inlet. Inlet is made as large as possible, to reduce loss due to friction of air entering the fan. In a wheel of given diameter, more power will be consumed to deliver a given volume of air with a small inlet than with a large one. If d = diam. of inlet, and P_v = pressure due to velocity V, P_v varies as V^2 and inversely as d^4. A small increase in diameter thus greatly reduces inlet loss for a given air delivery. An increase in diameter of inlet decreases the depth of fan blade, thus reducing capacity and pressure. To overcome this, the number of blades is increased to the limit placed by constructional considerations. In a properly proportioned fan, a balance is obtained between the two features

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* The author acknowledges the assistance of Messrs. M. S. Kice, Jr., of the American Blower Corp., and A. A. Criqui of the Buffalo Forge Co., in the preparation of this chapter.

of maximum inlet and maximum number of blades. With single-width fans, a double intake, where possible, is preferable, as the area of inlet opening is thus doubled and inlet losses are further decreased, resulting in gain of mechanical efficiency. With double-width fans, a double inlet is absolutely necessary.

THE TOTAL PRESSURE in the air delivered by a fan comprises the static pressure or compression, and the velocity pressure due to the kinetic energy of the air. Static pressure is used to overcome resistance of ducts, heaters, etc. Unless used for conveying materials held in suspension in the air, velocity pressure, to be of service, must be converted largely to static pressure, usually by a scroll-shaped casing around the fan wheel and a divergent nozzle on the outlet of the casing, if the velocity of air leaving the outlet is high.

RULES FOR FAN DESIGN.—It is impossible to give any general rules or formulas covering the proportions of parts of fans and blowers. No less than 14 variables are involved in the construction and operation of fans, a slight change in any one producing wide variations in performance. The design of a new fan is largely a matter of trial and error, based on experiments, until a compromise with all the variables is obtained which most nearly conforms to given conditions.

SHAPE OF BLADE.—Fig. 1 shows the effect of curvature of the fan blade. In these velocity diagrams V_2 represents the radial component of the air leaving the wheel, and V_1 its direction relative to the blade. U_2 represents tangential velocity due to rotation of the fan wheel and is equal to the velocity of the periphery of the wheel. R is the resultant velocity, and is the velocity relative to the housing. At a given speed, the forward-curved blade gives a higher pressure than that corresponding to the velocity of blade tip, and is best adapted to wheels in spiral casings. The backward-curved blade gives a lower corresponding pressure and can be used either in a spiral casing or with

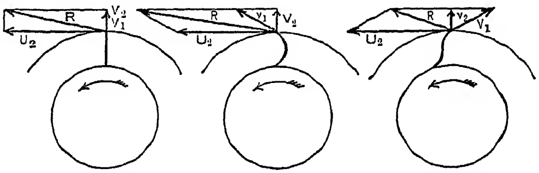


Fig. 1. Types of Fan Blades

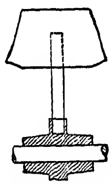


Fig. 2. Radial Fan Blade

a radial diffuser discharging directly to atmosphere. Resolving resultant R into radial and tangential components shows the radial component, as compared to R, to be relatively smaller with forward than with backward curved blades. Without a spiral around the fan wheel the resistance of back pressure is radially inward, hence the blade with the larger radial component is the more efficient.

The standard steel plate fan, in comparison with the multiblade fan, is essentially a straight-blade fan, although the tips of the blades may have a slight curvature, either backward or forward. Straight, radial-blade fans usually have trapezoidal-shaped blades, Fig. 2, to equalize the area of the ports between blades at inlet and outlet. If for constructional reasons, the blade is made rectangular, the area of the ports at inlet and outlet can be kept equal by curving the blades to an involute of a circle; in this case the axial length of blade is $1/4 \times$ diameter of wheel inlet. With blades whose cross-section is the arc of a circle, the minimum width of port between blades is on a line drawn through the tip of one blade normal to the next adjacent blade. The axial length of the blade, therefore, will exceed $1/4 \times$ inlet diameter, depending on the depth of the blade, being greater for shallow than for deep blades. In multiblade fans the inlet diameter closely approaches the diameter of the wheel with a correspondingly greater inlet area. The blades being shallow, their axial length may be from 4 to 8 times the radial depth. The shallowness of the blade is limited by its strength. The principal load on a blade is the centrifugal force due to the weight of the blade. Calculations for strength may be based on the assumption that the blade acts as a beam uniformly loaded.

A later development in blade construction for multiblade fans is described by F. W. Bailey and A. A. Criqui (Some Developments in Centrifugal Fan Design, *Trans*. A. S. H. V. E., 1921). The blade, in cross-section, is a reverse curve, each curve being the arc of a circle. At the heel the blade is concave in the direction of rotation, while at the tip it is convex. This type of blade is claimed to be stronger than the simple curved blade, and better able to resist the stresses due to centrifugal force. It is also claimed

to give pressure characteristics approaching those of the radial-blade fan. (See Pressure Characteristics of Fans, p. 1–58.)

FAN INLETS.—The shape of the inlet to the casing has considerable influence on fan efficiency. If the diameter of the inlet connection is less than the diameter of inlet of the fan wheel, a higher velocity of air is required through the inlet connection than when it is equal to or greater than the inlet of the wheel. Connections for attaching inlet pipes to the casing aggravate this condition. The preferable form of inlet to the casing is a stream-lined conduit which conforms to the natural direction of the flow of air into the inlet. Experiments have indicated a range of mechanical efficiency of from 65 to 70% in a fan with restricted inlet connection, compared to 70 to 80% in the same fan with stream-lined inlet, the volume of air and the static pressure being the same in both cases. The shape of the stream-lined inlet also exercises considerable influence on the mechanical efficiency. Stream-lined inlets are of particular advantage with the wider fans, where the inlet air velocity is high and inlet losses consequently greater.

The effect of obstructions in the inlet, such as bearings and their supports, is marked. The capacity may be decreased as much as 20% and the mechanical efficiency by from 2 to 3%. Due to this fact double-width, double-inlet fans, which require bearings in the inlets, have a lower capacity and efficiency than two single-inlet fans with overhung wheels and unobstructed inlets.

When a fan is used for exhausting, the duct is led to an inlet box fitted to the fan casing. The shape of the inlet box may have a marked effect on the performance of the fan. See *Trans.* A. S. M. E., FSP-54-16, 1932 and FSP-55-9,1933.

M. S. Kice, Jr., says that when double inlet fans are used as exhausters, as for induced draft work, the design of the box must be co-ordinated with the particular fan to which it is applied to enable air to pass through the fan inlet with minimum loss. Properly proportioned rectangular boxes have the advantage of minimum erosion as contrasted with spiral inlet boxes. The fan manufacturer develops these proportions along with the development of the fan.

Some types of multiblade fan inlets (1935) use stationary inlet vanes, curved to direct the air into the fan wheel with a minimum shock loss. Such vanes reduce the capacity, pressure and horsepower of these fans at a given speed, and increase overall efficiency by decreasing shock. Capacity and pressure desired may be obtained by a slight increase in speed, but the increased efficiency results in considerable reduction of horsepower. The vanes increase efficiency proportionately more for the higher capacities, and horsepower is at a maximum and decreases before free delivery condition is reached. This characteristic (see Fig. 5) provides a limit load for a given fan at a specified speed.

The effect of the ratio of diameter of fan wheel to diameter of inlet on the other dimensions and on the efficiency of a steel plate fan is shown by Table 1. In this table the standard fan inlet is 62.5% of the diameter of the wheel. The relative dimensions of fans of other ratios are based upon this standard. A high-efficiency fan, as shown by the table, is tall and narrow, and is therefore more expensive than the ordinary standard commercial fan. This type of fan frequently is used for induced draft work and for direct connection to steam engines.

Table 1.—Relative Dimensions of Steel Plate Fans Operating at the Same Capacity and Pressure

(From Fan Engineering, Buffalo Forge Co., Buffalo, N. Y.)

Diam. Inlet / Diam. Wheel	Diameters, Percent of Diameters of Standard Fan		Width, Percent of Width of Standard Fan	Horsepower, Percent of Horsepower of Standard Fan	Speed, Percent of Speed of Standard Fan
	Wheel	Inlet			
0.700	82.0	91.9	108.9	112.3	123.0
.650	93.2	97.5	102.5	104.0	109.5
.625	100.0	100.0	100.0	100.0	100.0
.600	106.9	102.6	97.5	96.7	92.7
.550	123.5	108.8	92.1	91.0	78.2
.500	144.9	116.6	85.9	86.8	64.5
.450	170.8	123.0	81.3	83.4	53.3
.400	206.5	132.4	75.5	80.0	43.1
.350	255.0	142.8	70.1	77.5	34.6

FAN OUTLETS.—The head of air developed by a fan consists of both static and velocity pressure, the latter ranging from 25% to 50% of the total pressure at outlet. If the fan discharges directly to atmosphere, all velocity pressure at outlet is lost. If the fan is fitted with an evasé discharge piece of proper construction, which will reduce the velocity of the air to from 1000 to 5000 ft. per min., a large percentage of this velocity pressure may be converted into static pressure. This evasé discharge piece should be tapered at an angle of from 7° to 10° from the axis, and should be of sufficient length to

reduce the velocity to the figures above given. Such an evasé discharge piece makes possible a decrease in speed of the fan, and increases the volumetric capacity by from 8% to 12% with an increase in horsepower of but from 3% to 5%, static pressure remaining the same.

PRESSURE CHARACTERISTICS OF FANS.—Pressure characteristics of centrifugal fans are determined by measuring, with a Pitot tube, the total pressure (also known as dynamic or impact pressure) and the static pressure at the fan outlet, at several different percentages of fan outlet opening, ranging from full opening to outlet entirely closed.

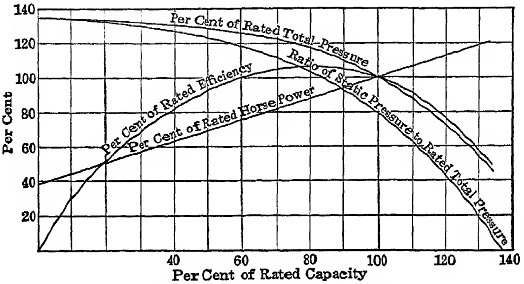


FIG. 3. Characteristic Curves of Radial Blade Fan

The fan is run at a constant speed during the entire test. The horsepower required to drive the fan at each condition of fan-outlet opening is also measured. From the readings, curves similar to those in Figs. 3, 4, and 5 are plotted. In making a fan test, actual quantities, as volume of air, horsepower, pressures in inches of water, etc., are plotted. If, however, these values are converted into percentages of rated performance of the fan under test, the curves may be used to determine the characteristics of any size of fan of similar design.

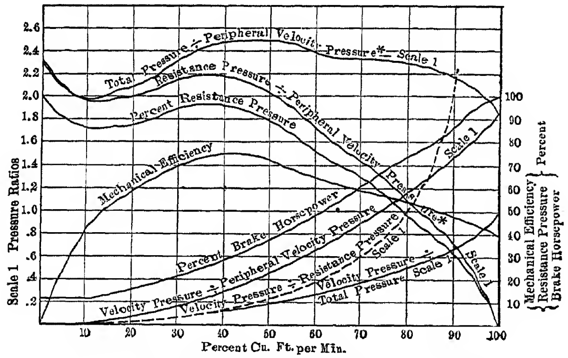


FIG. 4. Characteristic Curves of Forward-curved Blade Fan

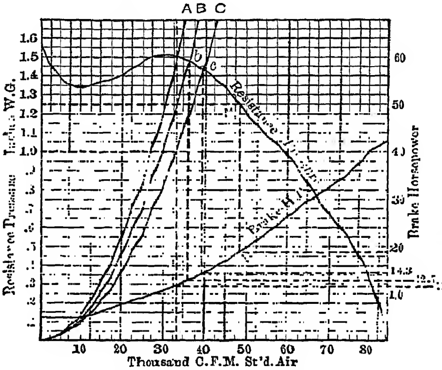
The shape of fan blade greatly influences the form of the characteristic curves. Fig. 3 represents the performance of a straight radial-blade fan. At constant speed, pressure increases as the load on the fan is reduced.

Fig. 4 shows characteristic curves for one type of fan with forward-curved blades. Fig. 5 shows the characteristic curve of a forward-curved blade fan on which are superposed three system curves, which are of value in determining the suitability of a given fan for a given set of conditions. A system curve is plotted on the basis of volume of air required *vs.* the resistance pressure necessary to force it through the system. The inter-

section of the system curve and the fan resistance-pressure curve shows conditions as they will actually occur, and which satisfies both fan and system characteristics. The volume of air delivered is read on the scale of abscissas at the foot of the ordinate drawn through the intersection. The horsepower required is read on the horsepower scale opposite the intersection of the aforesaid ordinate with the brake horsepower curve.

The use of system curves also permits the performance of the fan to be checked against a set of variable conditions, to determine its suitability for a wide variation in volume or resistance pressure.

Example.—A No. 10 Sirocco fan is chosen to deliver 32,950 cu. ft. per min. against a system resistance pressure of 1 1/2 in., requiring 11.71 Hp. at 213 r.p.m. (See Table 6.) If the system resistance pressure is 1 1/4 in. for 32,950 cu. ft. per min. instead of 1 1/2 in. for this same volume, the fan is capable of delivering approximately 50,000 cu. ft. per min. and requires approximately 19 Hp. (See points M and N, Fig. 5), overloading the 15 Hp. motor originally selected to drive the fan. This, however, cannot occur; if 50,000 cu. ft. per min. will flow through the system, the resistance pressure will rise to 2.65 in. Actually, the fan will handle a lesser quantity of air until fan and system characteristics coincide. Curve A shows the system conditions as originally estimated, *i.e.*, with resistance pressure 20% too high. Curve B shows the system conditions as they actually exist. Curve C shows system conditions estimated 20% too low. The intersection of curve B with the fan curve shows at point *b*, that the fan will deliver 36,000 cu. ft. per min. at a resistance pressure of 1.48 in., requiring 12.7 B.Hp. at 213 r.p.m. Curve C similarly shows (point *c*) that the fan under the given conditions will deliver 39,600 cu. ft. per min. at a resistance pressure of 1.44 in., requiring 14.3 B.Hp. at 213 r.p.m. These curves show that had curve *C* been the original estimate, and had the resistance pressure been estimated 50% too low, the fan selected would be capable of delivering the required amount of air without overloading the motor. They further show the importance of considering the system in connection with fan performance.



G. 5. Use of System Curves

Fig. 6 shows characteristics of a double-curved blade, multiblade fan with stationary inlet vanes. (See Fan Inlets.) In this particular fan the blade curves forward from its

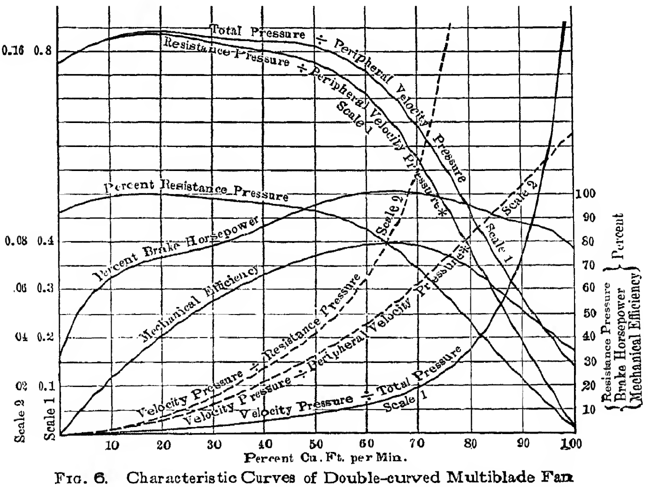


Fig. 6. Characteristic Curves of Double-curved Multiblade Fan

inner edge to about the middle of the blade, at which point the curve reverses, and at the periphery of the wheel the blade has a distinctly backward curvature. The pressure curve rises with a decrease in load in a manner similar to that of a straight radial-blade fan. This type of fan must run at higher speed for a given pressure than a forward-curved blade fan of the same size. The horsepower required is proportionately lower at the higher loads in comparison with the power required at the lower loads. In combination with the rising pressure curve, this indicates that the possibility of overloading the motor is eliminated. For a discussion of the characteristics of the double-curved blade fan, see Some Developments in Centrifugal Fan Design, by F. W. Bailey and A. A. Criqui (*Trans.* A.S.H.V.E., 1921). Fig. 7 shows the characteristics of a backward-curved blade fan.

For every fan running at constant speed there is a pressure and corresponding volume at which a fan will operate at its maximum efficiency (see Characteristic Curves), and a wide variation in these conditions will give a great drop in efficiency. In selecting a fan for any purpose the catalogs and bulletins issued by manufacturers should be examined and a tabular comparison made of the sizes, speed, etc., of different fans which may be used for the given purpose and conditions. (See Fan Tables and Fan Charts, pp. 1–63 to 1–79.)

THE RELATION OF CAPACITY, SPEED, PRESSURE AND HORSEPOWER for a given fan, piping system and density of air are: Let N = speed of fan, r.p.m., or ft. per min. of peripheral velocity; Q = capacity, cu. ft. per min.; V = outlet velocity of

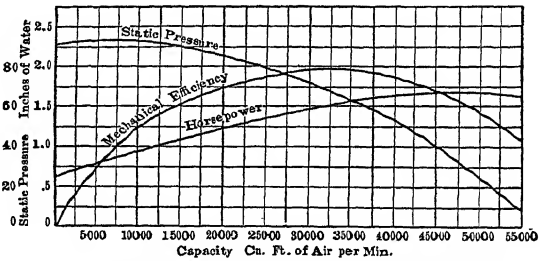


FIG. 7. Characteristic Curves of Backward-curved Blade Fan

air, ft. per min.; h = pressure, in. of water; Hp. = horsepower; D = diam. of fan wheel; W = density of air at absolute temperature t. Then

$$Q \propto N, \sqrt{h}. \quad V \propto N, Q. \quad N \propto \sqrt{h}. \quad h \propto N^2. \quad \text{Hp.} \propto N^3, Q^3, h^{3/2}.$$

For a given symmetrical line of fans, the following relations hold:

	Capacity varies as	Pressure varies as	Horsepower varies as	R.p.m. varies as
At constant r.p.m.	D^2	D^2	D^5	....
At varying r.p.m.	$D^3 \times R$ †	$D^2 \times R$ †	$D^5 \times R$ †	....
At constant tip speed	D^2	Unchanged	D^2	$1/D$
At constant capacity and speed	....	W, $1/t$ *	W, $1/t$ *	....
At constant pressure	$\sqrt{t}$	....	$\sqrt{t}$ *	$\sqrt{t}$

*Approximate. † R = ratio of r.p.m.

RELATION OF VELOCITY AND PRESSURE.—The pressure of the air due to the velocity of the fan blades may be determined by the formula $H = v^2/2g$, or $v = \sqrt{2gH}$, where H = "head" of air, ft., v = velocity of air leaving fan, ft. per second, and g = acceleration due to gravity = 32.2. The pressure of the air is increased by increasing the revolutions per minute of the fan.

The pressure as measured by a Pitot tube is given in inches of water, which may be converted to feet of head H, or to other units as follows:

Let P = pressure, lb. per sq. ft.; P_1 = pressure, lb. per sq. in.; p = pressure, oz. per sq. in.; h = pressure, in. of water; W = weight of air, lb. per cu. ft.; v = velocity of air, ft. per sec.; V = velocity of air, ft. per min. Let the humidity in the air be neglected. Then, since water at 70° F. weighs 62.30 lb. per cu. ft., $P = HW = 5.192\,h$, and $H = 5.192\,(h/W)$.

Substituting the value of H in $v = \sqrt{2\,gH}$, we have $v = \sqrt{64.32 \times 5.192\,(h/W)} = 18.27\sqrt{h/W}$, or $V = 1096.2\sqrt{h/W}$. At 70°F., $W = 0.07494$ (See Table 6, p. 1–05), whence $H = 5.192(h/W) = 69.282\,h$, from which the values of v and V are determined as follows: $v = \sqrt{2gH} = \sqrt{64.32 \times 69.282h} = 66.754\sqrt{h}$ ft. per sec.; $V = 4005\sqrt{h}$ ft. per min.

Since 1 oz. per sq. in. = 1.728 in. of water at 70°F., velocity, when pressure is measured in oz. per sq. in., is $v = 66.754\sqrt{h} = 66.754\sqrt{1.728\,p} = 87.71\sqrt{p}$ ft. per sec.; $V = 5263\sqrt{p}$ ft. per min.

If pressure is measured in lb. per sq. in., at 70°F., 1 lb. per sq. in. = 27.74 in. of water, and $v = 66.761\sqrt{h} = 66.761\sqrt{27.74\,P_1} = 350.89\sqrt{P_1}$ ft. per sec., $V = 21{,}053\sqrt{P_1}$ ft. per min.

The velocity at any other temperature than 70°F., can be found from the formula, $V_2 = V_1\sqrt{t_2/t_1}$, where V_1 and V_2 are velocities and t_1 and t_2 corresponding absolute temperatures, V_1 being the velocity at 70° F. For any other barometric pressure, velocity can be determined from the formula $V_2 = V_1\sqrt{p_1/p_2}$, where p_1 and p_2 are barometric pressures corresponding to velocities V_1 and V_2, respectively.

EFFECT OF HUMIDITY ON FAN PERFORMANCE.—Tables and charts relating to fan performance are based on dry air. The presence of moisture in the air requires, for strict accuracy, a correction according to the formula $V = V_1\sqrt{W/W_1}$, where V and V_1 = respectively, velocity, ft. per min., of air as observed and of dry air; W and W_1 = respectively, cu. ft. per lb. of air as observed, and of dry air. This correction is relatively unimportant for all ordinary cases of fan operation.

EFFECT OF TEMPERATURE ON FAN PERFORMANCE.—In a centrifugal fan, at constant speed and capacity, the horsepower required and the pressure developed will vary directly as the density and inversely as the absolute temperature of the air. If t_1 and t_2 be temperatures in deg. F., the corresponding absolute temperatures will be $(t_1 + 460)$ and $(t_2 + 460)$, and $(t_1 + 460)/(t_2 + 460) = K$. Let p_1 and p_2 be the respective volumes at these temperatures. Then if the volume of air to be delivered at temperatures t_1 and t_2 is constant, $p_2 = p_1/K$ and $\text{Hp.}_2 = \text{Hp.}_1/K$.

Table 2.—Velocity of Dry Air Due to Pressure

(Air at 70° F. and 29.92 in. barometer)

Pressure, in. of Water	Velocity, ft. per min.	Pressure, in. of Water	Velocity, ft. per min.	Pressure, in. of Water	Velocity, ft. per min.	Pressure, in. of Water	Velocity, ft. per min.	Pressure, in. of Water	Velocity, ft. per min.	Pressure, in. of Water	Velocity, ft. per min.
0.05	896	0.7	3351	2.00	5664	4.00	8,010	7.0	10,595	14.0	14,985
.1	1266	.75	3468	2.25	6007	4.25	8,256	7.5	10,968	15.0	15,510
.2	1791	.8	3582	2.50	6332	4.50	8,496	8.0	11,328	16.0	16,020
.25	2003	.9	3800	2.75	6641	4.75	8,729	9.0	12,015	17.0	16,513
.3	2193	1.00	4005	3.00	6937	5.00	8,943	10.0	12,665	18.0	16,990
.4	2533	1.25	4478	3.25	7220	5.50	9,392	11.0	13,282	19.0	17,456
.5	2832	1.50	4905	3.50	7492	6.00	9,810	12.0	13,875	20.0	17,910
.6	3102	1.75	5298	3.75	7756	6.50	10,210	13.0	14,440	....	....

Example.—Required the size, speed and horsepower of fan to handle 30,000 cu. ft. of air per min. at 3/4 in. pressure and at 425° F. *Solution.*—From Table 4, a No. 10 Limit Load Conoidal fan at 350 r.p.m. will deliver 30,312 cu. ft. at 1 1/4 in. static pressure, requiring 8.48 Hp.; it will deliver the same volume at 3/4 in. pressure and 425° F. Horsepower required at 425° F. = $\text{Hp.}_2 = \text{Hp.}_1/K = 8.48/1.67 = 5.08$. For maximum Hp. required at above speed, see limit load figures at bottom of Table 4. At 350 r.p.m., limit load or maximum Hp. will be 9.00, based on 70° air. Correcting for temperatures as above, limit load at 425° F. = 9.00/1.67 = 5.39 Hp.

HORSEPOWER OF FANS.—For a fan of perfect efficiency, the work expended in moving a column of air is the volume of air at 70° F. multiplied by the pressure, or $Q \times P = Q \times 5.193\,h_t$, where Q = volume, cu. ft. per min; P = pressure, lb. per sq. ft.; h_t = total pressure, in. of water. Then $\text{Hp.}_a = QP/33{,}000 = 5.193\,Qh_t/33{,}000$. If Q be taken as 1 cu. ft. and h_t as 1 in., $\text{Hp.}_a = 5.193\,Q/33{,}000 = 0.00015736\,Q$. If $\text{Hp.}_a = 1$ and $h_t = 1$ in., then $Q = 1/0.00015736 = 6355$, whence $\text{Hp.}_a = Qh_t/6355$.

From these equations the brake horsepower of a fan may be determined by the formulas $\text{Hp.}_b = 0.00015736\,Qh_t/E_t = 0.00015736\,Qh_s/E_s$, where h_t and h_s = total and static pressures, respectively, and E_t and E_s = total and static mechanical efficiencies, respectively.

FAN EFFICIENCY.—The *mechanical efficiency* of a fan is $E = \text{Hp.}_a/\text{Hp.}_b$, where Hp._a and Hp._b are air and brake horsepowers respectively. The value of Hp._a is usually

computed by means of the total pressure against which the fan operates, although sometimes the static pressure is used.

Static Efficiency. Total Efficiency. Mechanical efficiency may be stated either as static or as total efficiency. Static efficiency is the ratio of air Hp., calculated by means of static pressure, to brake Hp. That is, $E_s = (Q \times 5.193\ h_s/33{,}000)/\text{B.Hp.}$ Total efficiency is the ratio of air Hp., calculated by means of the total pressure of the air, to brake Hp. That is, $E_t = (Q \times 5.193\ h_t/33{,}000)/\text{B.Hp.}$; Q = cu. ft. of air per min., and h_s and h_t = static and total pressures, respectively, in. of water.

Manometric Efficiency, E_m is variously given by different writers as $E_m = gH/V_b{}^2$ and $E_m = 2\ gH/V_b{}^2$, where H = head of air, ft.; V_b = peripheral speed of fan-blade tips, ft. per sec.; and g = acceleration due to gravity = 32.2. It is defined as the ratio of the pressure developed by the fan to the pressure against a plane surface due to a velocity equal to the peripheral velocity of one fan blade.

Volumetric Efficiency, E_v is the ratio of volume of air delivered per revolution of the fan and the cubical contents of the fan wheel. The term is a misnomer, since it does not fulfill the definition of efficiency, *viz.*: energy recovered ÷ energy expended. The formula for volumetric efficiency is $E_v = Q/\pi r^2 wn$, where Q = volume of air delivered, cu. ft. per min.; r = radius of fan wheel, ft.; w = width of fan wheel, ft.; n = rev. per min.

The terms Manometric and Volumetric efficiency are without much significance and rarely are used in present day fan practice.

SELECTION OF FANS.—The determining features in selecting the proper size of fan for a particular installation are the power requirements and the velocity of air at fan outlet. Some classes of work require absolute quietness of operation, while others do not. The lower outlet velocities are used where quietness is essential. Table 3 represents outlet velocities covering the average range of practice, and can be used in making fan selections. These outlet velocities are lower than those recommended several years ago, since present (1935) types of fans have larger outlets in proportion to other dimensions. Comparative noise tests show fans with fixed inlet vanes to operate at lower noise levels than the same fans without vanes. Experience will assist the engineer in making the best selection for any specific condition. See also notes on Fan Tables and Fan Charts.

Table 3.—Permissible Fan Outlet Velocities

	Supply System	Exhaust System
Churches	800–1000	1000–1400
Schools and Theaters	1000–1400	1200–1600
Hotels and Offices	1200–1600	1400–1800
Factories	1500–2300	1700–2500

FAN TABLES are published in two forms: 1. A rated capacity table giving capacity, speed and horsepower of the fan when operating at point of best efficiency. As a fan frequently operates under conditions other than those of best efficiency, tabular figures should be modified by reference to characteristic curves, when such are available, or by calculation (see below) to ascertain probable performance at other than rated loads. 2. A multi-rating table giving performance of each size of fan at each static pressure at which the fan may operate, showing directly horsepower required by any size of fan to deliver any given volume against any static pressure, and enabling performance of different fans under identical conditions to be compared. Tables 4 to 7 are such tables. Figures for any other size of fan may be calculated from the data in these tables.

Examples in Use of Fan Tables.—In fans operating at constant pressure and with the same outlet velocity: 1. Capacity and horsepower vary as square of diam. D of fan wheel; 2. Speed varies inversely as D.

Example 1.—Required size of a Limit Load Conoidal fan to deliver 13,000 cu. ft. of air per min. at a static pressure of 1 1/2 in. *Solution.*—From Table 3 outlet velocity should be from 1200 to 1600 ft. per min. From Table 4, capacity of a No. 10 fan (60-in. fan wheel) at 1400 ft. per min. outlet velocity is 28,292 cu. ft. per min. From law (1) above $(13{,}000/28{,}292) = (D/60)^2$ and $D = 40.6$. The nearest commercial size of fan (Table 4) has a 42-in. wheel. To obtain exactly 13,000 cu. ft. per min. at 1 1/2 in. static pressure for a 42-in. fan, find from law (1) the corresponding capacity C at 1 1/2 in. static pressure for a No. 10 fan; thus, $(60/42)^2 = (C/13{,}000)$ and C = 26,500 cu. ft. per min. The nearest value of C in Table 4 to 26,500 is 26,271 at an outlet velocity of 1300 ft. per min., which is satisfactory. Opposite this velocity and under 1 1/2 in. static pressure, find the speed to be 361 r.p.m. and Hp. to be 8.6. Applying law (1) to find Hp. for a 42-in. fan, $8.6 \times (42/60)^2 = 4.22$ Hp. From law (2) the speed of the 42-in. fan is $361 \times (60/42) = 516$ r.p.m. The fan selected, therefore, will be a No. 7 (42-in.) Limit Load Conoidal fan operating at 516 r.p.m. and requiring 4.22 Hp.

If it is necessary to select a fan to operate at pressures not covered by the tables, the following laws are applied to convert the given figures to values in the tables. For a given size of fan and

Table 4.—Speed and Horsepower of No. 10 Limit Load Conoidal Fan at Various Static Pressures and Capacities

(Buffalo Forge 34)

0° F and 29. rome

Outlet Velocity, ft. per min.	Capacity, cu. ft. per min.	Static Pressure, in. of Water: 1/4 R.p.m.	1/4 Hp.	3/8 R.p.m.	3/8 Hp.	1/2 R.p.m.	1/2 Hp.	5/8 R.p.m.	5/8 Hp.	3/4 R.p.m.	3/4 Hp.	7/8 R.p.m.	7/8 Hp.	1 R.p.m.	1 Hp.	1 1/4 R.p.m.	1 1/4 Hp.	1 1/2 R.p.m.	1 1/2 Hp.	2 R.p.m.	2 Hp.	2 1/2 R.p.m.	2 1/2 Hp.	3 R.p.m.	3 Hp.
800	16167	168	0.965	190	1.35	211	1.77	230	2.21	249	2.67	265	3.14	281	3.62	310	4.63	336	5.72	382	7.99	424	10.4	462	13.1
900	18187	178	1.16	198	1.57	218	2.01	236	2.49	254	2.98	270	3.49	286	4.01	314	5.07	340	6.22	386	8.67	428	11.2	466	13.9
1000	20208	188	1.39	207	1.82	226	2.29	243	2.80	260	3.32	276	3.86	291	4.42	319	5.58	345	6.77	390	9.29	432	12.0	470	14.9
1100	22229	199	1.65	217	2.12	234	2.61	251	3.14	267	3.69	282	4.26	297	4.86	324	6.08	350	7.36	395	10.0	436	12.8	474	15.8
1200	24250	210	1.94	227	2.46	243	2.98	259	3.53	274	4.11	289	4.71	303	5.33	330	6.61	356	7.94	400	10.8	441	13.7	478	16.8
1300	26271	222	2.29	238	2.83	253	3.39	268	3.97	282	4.57	296	5.19	311	5.85	336	7.19	361	8.60	405	11.6	446	14.7	483	17.9
1400	28292	234	2.68	249	3.25	264	3.85	278	4.46	291	5.08	305	5.74	319	6.38	343	7.79	367	9.29	410	12.4	451	15.6	488	19.0
1500	30312	246	3.11	261	3.73	275	4.35	288	4.99	301	5.67	314	6.33	327	7.03	350	8.48	374	10.0	416	13.2	456	16.6	493	20.2
1600	32333	258	3.61	273	4.25	286	4.91	299	5.60	311	6.29	324	6.99	335	7.70	358	9.20	381	10.8	422	14.1	461	17.7	498	21.3
1700	34354	270	4.14	285	4.85	297	5.53	310	6.24	322	6.99	334	7.72	345	8.47	367	10.0	388	11.6	428	15.1	467	18.7	503	22.6
1800	36375	283	4.78	297	5.47	309	6.19	322	6.94	333	7.70	344	8.50	355	9.28	376	10.9	396	12.6	435	16.1	473	19.9	509	23.9
1900	38396	296	5.41	309	6.17	321	6.93	333	7.72	344	8.52	354	9.33	365	10.1	386	11.8	404	13.5	443	17.2	480	21.1	515	25.1
2000	40417	309	6.05	321	6.93	333	7.75	344	8.54	355	9.35	365	10.2	376	11.1	396	12.8	414	14.6	451	18.4	487	22.4	521	26.6
2100	42437	....	.....	333	7.78	345	8.61	356	9.48	366	10.3	376	11.2	387	12.2	406	13.9	424	15.8	460	19.6	494	23.7	528	28.0
2200	44458	....	.....	346	8.80	357	9.54	368	10.5	377	11.4	388	12.3	398	13.2	416	15.1	434	17.0	469	20.9	502	25.1	535	29.6
2300	46479	....	.....	359	9.72	369	10.6	380	11.5	389	12.5	400	13.4	409	14.3	427	16.3	444	18.3	478	22.4	510	26.6	542	31.1
2400	48500	....	.....	372	10.7	382	11.7	392	12.7	401	13.7	412	14.6	421	15.5	438	17.6	455	19.7	488	23.9	518	28.2	550	32.8
2500	50521	....	.....	....	.....	395	12.9	404	13.8	414	14.9	424	15.9	433	16.9	449	19.0	466	21.1	497	25.4	527	29.9	557	34.6
2700	54562	....	.....	....	.....	421	15.5	429	16.6	439	17.6	447	18.8	455	19.8	472	22.0	488	24.3	518	29.0	546	33.6	573	38.5
2900	58604	....	.....	....	.....	....	.....	454	19.7	463	20.7	471	22.0	479	23.2	495	25.4	511	27.9	539	32.8	567	37.6	593	42.9
3100	62646	....	.....	....	.....	....	.....	480	22.9	488	24.4	496	25.7	504	26.9	519	29.3	534	31.8	560	37.0	588	42.2	613	47.7
3300	66687	....	.....	....	.....	....	.....	....	.....	513	28.3	521	29.7	529	30.9	544	33.5	558	36.3	583	41.6	609	47.3	634	53.1
3500	70729	....	.....	....	.....	....	.....	....	.....	540	32.7	547	34.3	555	35.6	569	38.2	582	41.2	607	46.7	631	52.6	655	58.7

Limit Load for Various R.P.M.

R.p.m.	150	170	190	210	230	250	270	290	310	330	350	370	390	410	430	450	470	490	510	530	550	570	590	610	630	650	670
Hp.	0.708	1.03	1.44	1.95	2.55	3.28	4.14	5.12	6.26	7.55	9.00	10.7	12.5	14.5	16.7	19.2	21.8	24.8	27.9	31.3	35.0	39.0	43.2	47.7	52.6	57.7	63.2

piping system: *a.* Capacity varies directly as speed; *b.* Pressure varies as square of speed; *c.* Speed and capacity vary as square root of pressure; *d.* Horsepower varies as cube of capacity or speed.

Example 2.—Required the size of a Limit Load Conoidal fan to supply 40,000 cu. ft. per min. at $1/8$-in. static pressure, with maximum outlet velocity of 1200 ft. per min. *Solution.*—Find corresponding capacity at $1/4$ in. static pressure by applying law (*c*). $(\sqrt{1/4}/\sqrt{1/8}) = (C/40{,}000)$ whence $C = 56{,}570$ cu. ft. per min. By law (*c*), outlet velocity at $1/4$ in. static pressure is also found to be $1200 \times (\sqrt{1/4}/\sqrt{1/8}) = 1697$ ft. per min. From Table 4, capacity at 1697 ft. per min. may be taken as 34,354 cu. ft. per min. Applying law (1) as before, $(56{,}570/34{,}354) = (D/60)^2$, and $D = 77$ in. The nearest commercial size of fan (Table 5) is 78 in., and from law (1), $(60/78)^2 = (C/56{,}570)$, and $C = 33{,}470$ cu. ft. per min. Interpolating in Table 4, 33,470 cu. ft. per min. at $1/4$ in. static pressure will require 3.91 Hp. at 263 r.p.m., and the outlet velocity will be 1660 ft. per min. Converting these values to a 78-in. fan at $1/4$ in. static pressure, $C = 33{,}470 \times (78/60)^2 = 56{,}570$; r.p.m. $= 263 \times (60/78) = 202$; Hp. $= 3.91 \times (78/60)^2 = 6.61$.

Transforming to $1/8$ in. static pressure to comply with the given conditions,

From law (*c*) $C = 56{,}570 \times (\sqrt{1/8}/\sqrt{1/4}) = 40{,}000$ cu. ft. per min.;
" " (*c*) r.p.m. $= 202 \times (\sqrt{1/8}/\sqrt{1/4}) = 143$;
" " (*d*) Hp. $= 6.61 \times (143/202)^3 = 2.35$;
" " (*c*) Outlet velocity $= 1660 \times (\sqrt{1/8}/\sqrt{1/4}) = 1174$ ft. per min.

The fan selected, therefore, will be a No. 13 (78-in.) Limit Load Conoidal fan, operating at 143 r.p.m. and requiring 2.35 Hp.

Table 5.—Dimensions of Limit Load Conoidal Fans
(Buffalo Forge Co., Buffalo, N. Y., 1934)

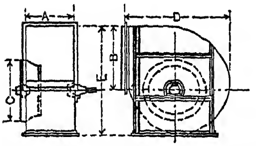


Fig. 8

Fan Size	Dimensions, in.					
	Wheel Diam., in.	Width, *A*	Outlet, *B*	Inlet, *C*	Length, *D*	Height, *E*
2	12	9 1/2	12 7/8	12 3/4	20 1/4	24 7/8
2 1/2	15	11 7/8	16	15 3/4	25 1/4	30 3/8
3	18	14 1/8	19 1/8	18 3/4	29 7/8	36 3/8
3 1/2	21	16 1/2	22 1/4	21 7/8	34 3/4	42 1/2
4	24	18 3/4	25 3/8	25	39 1/2	48 1/4
4 1/2	27	21 1/8	28 5/8	28	44 1/4	53 7/8
5	30	23 3/8	31 3/4	30 3/4	50 1/4	54 5/8
5 1/2	33	25 3/4	34 7/8	33 3/4	55	60 3/8
6	36	28	38	36 3/4	59 3/4	65 1/2
7	42	32 5/8	44 1/4	43	69 3/8	76 3/8
8	48	37 1/4	50 1/2	49 1/8	79	86 3/4
9	54	41 7/8	57	55 1/8	88 5/8	97 5/8
10	60	46 1/2	63 1/4	61 1/4	98 3/4	108 5/8
11	66	51 1/8	69 1/2	67 3/8	108 1/2	119
12	72	55 3/4	75 3/4	73 3/8	118	129 7/8
13	78	60 3/8	82	79 3/8	127 5/8	140 3/4
14	84	65	88 1/4	85 1/2	137 1/4	151 1/8
15	90	69 5/8	94 1/2	91 5/8	146 7/8	162
16	96	74 1/4	100 3/4	97 3/4	156 3/8	172 3/8
18	108	83 1/2	113 3/4	109 7/8	175 5/8	194 1/8
19	114	88 1/8	119 3/4	116	185 3/8	204 5/8
20	120	92 3/4	126 1/4	122 1/8	195 1/8	215 5/8

Tables 6 and 7 are multi-rating tables of a No. 10 Sirocco fan (forward curved multi-blade), and a No. 10 high-speed fan (backward curved blade), both the product of the American Blower Corp. Both types are built in two classes: Class I fans, for air conditioning or heating and ventilating duty, may be operated for a maximum duty of $3\,3/4$ in. total pressure at 70° F. and sea level without exceeding safe structural limits, and may be used only for the ratings above the heavy black lines in the capacity tables; Class II fans, built primarily for industrial work, may be operated up to total pressures of $6\,3/4$ in. at 70° F. and sea-level. The maximum total pressure for higher temperatures and elevations will be less than those shown in the tables. They *must* be used for all ratings below the heavy black lines in the capacity tables, but may be used for all ratings, if noise is not a factor. The determination for capacity for other sizes than the No. 10 may be accomplished by the same type of calculation as given above. Dimensions of both types are given in Tables 8 and 9.

Table 10 gives capacity of a No. 50 steel plate exhaust fan at various speeds and pressures. Capacities of other sizes can be determined as above described. This fan is essentially a straight radial blade fan, although it is fitted with other types of wheels

(*Continued on* p. 1-71.)

Table 6.—Capacity of No. 10 Single-width Single-inlet Sirocco Fan. (See Notes, page 1–64.)
(American Blower Corp., Detroit, Mich., 1934)

SINGLE WIDTH, SINGLE INLET. OUTLET AREA = 19.38 SQ. FT. WHEEL DIAM. = 4.948 FT.

Volume, cu. ft. per min.	Velocity, ft. per min.	1/8 in. R.P. R.p.m.	1/8 in. R.P. B.Hp.	1/4 in. R.P. R.p.m.	1/4 in. R.P. B.Hp.	3/8 in. R.P. R.p.m.	3/8 in. R.P. B.Hp.	1/2 in. R.P. R.p.m.	1/2 in. R.P. B.Hp.	5/8 in. R.P. R.p.m.	5/8 in. R.P. B.Hp.	3/4 in. R.P. R.p.m.	3/4 in. R.P. B.Hp.	7/3 in. R.P. R.p.m.	7/3 in. R.P. B.Hp.	1 in. R.P. R.p.m.	1 in. R.P. B.Hp.	1 1/4 in. R.P. R.p.m.	1 1/4 in. R.P. B.Hp.	1 1/2 in. R.P. R.p.m.	1 1/2 in. R.P. B.Hp.	1 3/4 in. R.P. R.p.m.	1 3/4 in. R.P. B.Hp.	2 in. R.P. R.p.m.	2 in. R.P. B.Hp.	2 1/4 in. R.P. R.p.m.	2 1/4 in. R.P. B.Hp.	2 1/2 in. R.P. R.p.m.	2 1/2 in. R.P. B.Hp.	3 in. R.P. R.p.m.	3 in. R.P. B.Hp.	4 in. R.P. R.p.m.	4 in. R.P. B.Hp.
19380	1000	73	0.93	91	1.36	108	1.78	123	2.32	...	...	...	...	...	...	...	...	...	...	...	...	...	...	...	...	...	...	...	...	...	...	...	...
21320	1100	75	1.12	94	1.62	109	2.06	124	2.57	...	...	...	...	...	...	...	...	...	...	...	...	...	...	...	...	...	...	...	...	...	...	...	...
23260	1200	78	1.35	97	1.91	111	2.40	125	2.90	139	3.49	...	...	...	...	...	...	...	...	...	...	...	...	...	...	...	...	...	...	...	...	...	...
25190	1300	81	1.61	100	2.24	114	2.78	127	3.30	140	3.88	151	4.52	163	5.26	...	...	...	...	...	...	...	...	...	...	...	...	...	...	...	...	...	...
27130	1400	84	1.91	103	2.59	116	3.20	129	3.77	141	4.33	153	4.97	164	5.68	174	6.46	...	...	...	...	...	...	...	...	...	...						
29070	1500	88	2.26	105	2.96	120	3.66	131	4.26	143	4.90	154	5.50	165	6.19	175	6.96	...	...	...	...	...	...	...	...	...	...						
31010	1600	91	2.64	108	3.37	123	4.14	134	4.83	145	5.47	156	6.11	166	6.80	176	7.55	195	9.23	...	...	...	...	...	...	...	...						
32950	1700	95	3.09	110	3.83	125	4.67	137	5.43	147	6.12	157	6.80	168	7.49	177	8.24	196	9.89	213	11.77	...	...	...	...	...	...						
34880	1800	99	3.57	113	4.36	128	5.23	140	6.03	150	6.86	160	7.57	169	8.27	179	9.00	197	10.64	214	12.49	230	14.48	...	...	...	...						
36820	1900	102	4.09	117	4.93	130	5.81	143	6.75	153	7.57	162	8.37	171	9.14	181	9.87	198	11.47	215	13.33	231	15.30	...	...	...	...	...	...	...	...	...	...
38760	2000	106	4.70	120	5.55	133	6.47	148	7.46	156	8.35	165	9.24	174	9.99	183	10.89	200	12.53	216	14.25	232	16.24	247	18.47	...	...	...	...	...	...	...	...
42640	2200	115	6.10	127	6.99	139	7.97	151	8.99	162	10.07	171	11.05	179	12.02	187	12.94	203	14.69	219	16.50	234	18.41	249	20.59	262	22.83	...	...	...	...	...	...
46510	2400	123	7.68	134	8.70	145	9.71	156	10.83	167	11.97	177	13.12	186	14.23	194	15.31	208	17.25	223	19.22	237	21.15	251	23.24	264	25.47	277	27.91	302	33.23	...	...
50390	2600	132	9.62	142	10.66	152	11.77	162	12.88	172	14.10	182	15.44	192	16.62	200	17.91	214	20.12	227	22.19	241	24.43	254	26.45	267	28.60	279	31.02	303	36.18	...	...
54260	2800	141	11.93	...	...	159	14.09	169	15.34	178	16.64	187	17.91	197	19.14	205	20.68	220	23.18	232	25.62	245	27.80	258	30.24	270	32.27	282	34.64	305	39.73	348	51.52
58140	3000	149	14.36	...	...	166	16.82	175	18.03	184	19.40	193	20.73	202	22.10	210	23.69	226	26.48	239	29.25	251	31.86	262	34.12	274	36.75	285	38.96	308	43.86	350	55.65
62020	3200	158	17.36	...	...	174	19.81	182	21.07	191	22.56	199	23.88	207	25.49	215	26.98	231	30.19	245	33.20	257	35.88	267	38.62	278	41.22	289	43.77	311	48.92	352	60.36
65890	3400	168	20.62	...	...	...	...	190	24.66	198	26.21	206	27.60	213	29.11	221	30.67	236	34.08	251	37.28	263	40.37	274	43.36	284	46.34	294	48.91	315	54.81	354	65.71
69770	3600	176	24.21	...	...	...	...	197	28.42	205	29.89	212	31.65	220	33.20	227	34.86	241	38.13	256	41.70	269	45.25	280	48.27	290	51.52	300	54.88	319	60.26	358	72.03
73640	3800	185	28.21	...	...	...	...	...	...	212	34.55	220	36.26	226	37.58	233	39.37	247	42.93	261	46.47	274	50.30	286	53.98	296	57.07	306	60.47	324	66.99	361	79.44
79520	4000	195	32.82	...	...	...	...	...	...	220	39.25	226	40.65	233	42.43	240	44.43	253	47.97	265	51.59	279	55.81	291	59.49	302	63.22	312	66.80	330	73.58	365	87.12
85270	4400	...	...	...	...	...	...	...	...	...	...	...	...	...	...	...	...	266	59.79	278	63.79	289	67.77	301	71.90	313	76.47	324	80.54	343	88.39	375	103.55
93020	4800	...	...	...	...	...	...	...	...	...	...	...	...	...	...	...	...	279	73.18	291	77.67	302	82.20	312	86.66	320	91.07	334	95.63	354	105.15	387	122.30

Values below the heavy black lines are for Class II fans, which should be used whenever these ratings are desired.

Table 6A.—Capacities of No. 10 Double-width, Double-inlet Sirocco Fan. (See Notes, p. 1–64)

(American Blower Corp., Detroit, Mich., 1934)

DOUBLE WIDTH, DOUBLE INLET. OUTLET AREA = 34.15 SQ. FT. WHEEL DIAM. = 4.948 FT.

Volume, cu. ft. per min.	Velocity, ft. per min.	1/8 in. R.P. R.p.m.	1/8 in. R.P. B.Hp.	1/4 in. R.P. R.p.m.	1/4 in. R.P. B.Hp.	3/8 in. R.P. R.p.m.	3/8 in. R.P. B.Hp.	1/2 in. R.P. R.p.m.	1/2 in. R.P. B.Hp.	5/8 in. R.P. R.p.m.	5/8 in. R.P. B.Hp.	3/4 in. R.P. R.p.m.	3/4 in. R.P. B.Hp.	7/8 in. R.P. R.p.m.	7/8 in. R.P. B.Hp.
34150	1000	69	1.54	90	2.33	106	3.13	...	...	...	...	...	...	...	...
37570	1100	72	1.87	91	2.75	108	3.59	122	4.94	...	...	...	...	...	...
40980	1200	75	2.25	93	3.20	109	4.13	123	5.51	136	6.15	...	...	...	...
44400	1300	78	2.71	95	3.72	111	4.73	125	6.16	137	6.80	148	7.97	160	9.16
47810	1400	81	3.24	98	4.30	113	5.40	126	6.92	139	7.54	150	8.75	161	10.00
51230	1500	85	3.91	100	4.93	115	6.11	128	7.76	140	8.44	151	9.60	162	10.91
54640	1600	89	4.66	103	5.61	117	6.89	130	8.66	142	9.37	153	10.57	163	11.89
58060	1700	94	5.55	106	6.41	119	7.78	132	9.64	144	10.42	154	11.68	165	13.01
61470	1800	102	6.46	109	7.33	122	8.70	134	10.62	146	11.53	157	12.90	166	14.24
64890	1900	108	7.54	112	8.31	125	9.66	136	11.78	148	12.66	158	14.21	168	15.61
68300	2000	113	8.78	116	9.48	127	10.80	139	13.11	150	13.93	160	15.48	170	17.06
75130	2200	122	11.56	...	...	134	13.35	144	15.61	154	16.72	164	18.43	174	20.17
81960	2400	132	14.82	...	...	140	16.51	150	19.54	159	19.91	169	21.83	178	23.66
88790	2600	141	18.84	...	...	...	...	156	23.32	165	23.56	174	25.67	182	27.52
95620	2800	152	23.41	...	...	...	...	163	27.80	171	27.94	179	29.80	188	32.03
102450	3000	161	28.20	...	...	...	...	...	...	178	32.78	185	34.69	193	36.90
109280	3200	172	34.64	...	...	...	...	...	...	185	38.69	192	40.17	199	42.56
116110	3400	...	...	...	...	...	...	...	...	...	...	...	...	205	48.96
122940	3600	...	...	...	...	...	...	...	...	...	...	...	...	212	56.19
129770	3800	...	...	...	...	...	...	...	...	...	...	...	...	219	63.89
136600	4000	...	...	...	...	...	...	...	...	...	...	...	...	...	...
150260	4400	...	...	...	...	...	...	...	...	...	...	...	...	...	...
163920	4800	...	...	...	...	...	...	...	...	...	...	...	...	...	...

Volume, cu. ft. per min.	Velocity, ft. per min.	1 in. R.P. R.p.m.	1 in. R.P. B.Hp.	1 1/4 in. R.P. R.p.m.	1 1/4 in. R.P. B.Hp.	1 1/2 in. R.P. R.p.m.	1 1/2 in. R.P. B.Hp.	1 3/4 in. R.P. R.p.m.	1 3/4 in. R.P. B.Hp.	2 in. R.P. R.p.m.	2 in. R.P. B.Hp.	2 1/4 in. R.P. R.p.m.	2 1/4 in. R.P. B.Hp.	2 1/2 in. R.P. R.p.m.	2 1/2 in. R.P. B.Hp.	3 in. R.P. R.p.m.	3 in. R.P. B.Hp.	4 in. R.P. R.p.m.	4 in. R.P. B.Hp.
34150	1000	...	...	...	...	...	...	...	...	...	...	...	...	...	...	...	...	...	...
37570	1100	...	...	...	...	...	...	...	...	...	...	...	...	...	...	...	...	...	...
40980	1200	...	...	...	...	...	...	...	...	...	...	...	...	...	...	...	...	...	...
44400	1300	...	...	...	...	...	...	...	...	...	...	...	...	...	...	...	...	...	...
47810	1400	171	11.27	...	...	...	...	...	...	...	...	...	...	...	...	...	...	...	...
51230	1500	172	12.24	...	...	...	...	...	...	...	...	...	...						
54640	1600	173	13.26	192	16.18	...	...	...	...	...	...	...	...						
58060	1700	174	14.42	193	17.40	210	20.53	...	...	...	...	...	...						
61470	1800	176	15.67	194	18.73	211	21.96	...	...	...	...	...	...						
64890	1900	177	17.05	195	20.14	212	23.42	227	26.89	...	...	...	...	...	...	...	...	...	...
68300	2000	179	18.66	196	21.75	213	25.04	228	28.54	243	32.28	...	...	...	...	...	...	...	...
75130	2200	183	21.96	199	25.20	215	28.71	230	32.35	245	36.20	258	40.18	...	...	...	...	...	...
81960	2400	187	25.68	203	29.40	218	33.03	233	36.77	247	40.68	260	44.76	273	49.18	298	57.92	...	...
88790	2600	191	29.71	207	33.87	222	37.83	236	41.92	249	45.73	262	50.05	275	54.44	299	63.67	...	...
95620	2800	196	34.33	212	39.11	226	43.26	239	47.60	253	51.37	265	56.01	277	60.29	300	69.82	344	90.2
102450	3000	201	39.33	216	44.09	230	48.82	243	53.87	256	58.17	268	62.96	281	67.21	303	76.77	345	97.9
109280	3200	206	44.91	220	49.99	234	55.08	247	60.70	260	65.22	272	70.48	284	74.98	305	84.81	347	106.1
116110	3400	212	51.24	226	56.57	239	62.18	252	67.92	264	72.50	276	78.70	287	83.39	309	94.19	349	115.3
122940	3600	218	58.35	231	63.45	244	69.60	256	75.60	268	80.52	280	87.04	291	92.25	312	102.8	351	125.4
129770	3800	225	66.42	237	71.57	249	77.07	261	83.96	273	89.72	284	96.21	295	101.3	316	113.8	355	136.7
136600	4000	232	75.82	243	80.46	255	86.24	266	92.93	278	98.73	288	106.2	299	111.4	320	124.2	358	149.3
150260	4400	...	...	257	101.1	267	107.1	277	113.8	288	119.7	298	127.8	308	133.7	329	148.1	365	176.0
163920	4800	...	...	...	...	281	132.6	290	138.7	299	144.3	309	152.3	319	159.3	338	175.4	...	...

Values below the heavy black lines are for Class II fans, which should be used whenever these ratings are desired.

Table 7.—Capacities of No. 10 High-speed Fans

(American Blower Corp., Detroit, Mich., 1934)

Wheel Diam., 4.948 ft. Max. B.Hp. = 324.2 (R.p.m./1000)³. For method of determining capacity of any other size of fan, see page 1–62.

Volume, cu. ft. per min.	Outlet Velocity, ft. per min.	1/8 in. R.P.		1/4 in. R.P.		3/8 in. R.P.		1/2 in. R.P.		5/8 in. R.P.		3/4 in. R.P.		7/8 in. R.P.		1 in. R.P.		1 1/4 in. R.P.		1 1/2 in. R.P.		1 3/4 in. R.P.		2 in. R.P.		2 1/4 in. R.P.		2 1/2 in. R.P.		3 in. R.P.		4 in. R.P.	
		R.p.m.	B.Hp.	R.p.m.	B.Hp.	R.p.m.	B.Hp.	R.p.m.	B.Hp.	R.p.m.	B.Hp.	R.p.m.	B.Hp.	R.p.m.	B.Hp.	R.p.m.	B.Hp.	R.p.m.	B.Hp.	R.p.m.	B.Hp.	R.p.m.	B.Hp.	R.p.m.	B.Hp.	R.p.m.	B.Hp.	R.p.m.	B.Hp.	R.p.m.	B.Hp.	R.p.m.	B.Hp.
		SINGLE WIDTH—SINGLE INLET. OUTLET AREA = 19.38 SQ. FT.																															
11630	600	138	0.37	162	0.63	191	0.99	208	1.27	230	1.64	251	2.03	270	2.30	287	2.94	320	3.93	349	5.09	...	.....	...	.....	...	.....	...	.....	...	.....	...	.....
13570	700	154	.51	175	.77	197	1.09	215	1.45	233	1.85	253	2.26	272	2.62	289	3.19	322	4.19	351	5.44	378	6.55	403	7.60	427	8.86	...	.....	...	.....	...	.....
15500	800	171	.66	188	.95	207	1.28	224	1.66	243	2.06	261	2.47	275	2.96	292	3.44	324	4.51	354	5.56	380	6.94	405	8.29	429	9.18	452	10.47	493	13.52	...	.....
17400	900	189	.85	204	1.18	220	1.53	235	1.91	252	2.33	268	2.78	282	3.26	297	3.76	327	4.82	356	5.86	382	6.98	407	8.64	431	9.91	453	11.51	495	13.77	...	.....
19380	1000	204	1.06	221	1.47	234	1.79	250	2.23	264	2.65	278	3.12	292	3.61	306	4.13	332	5.22	358	6.39	385	7.60	410	8.80	433	10.43	456	11.84	497	14.66	570	21.81
21320	1100	221	1.34	239	1.78	249	2.16	263	2.56	276	3.02	289	3.50	302	4.01	315	4.54	338	5.69	363	6.90	388	8.16	411	9.54	435	10.99	457	12.54	499	15.63	572	22.48
23260	1200	237	1.65	254	2.12	266	2.59	278	2.99	290	3.41	301	3.94	314	4.46	326	5.02	349	6.19	373	7.46	394	8.83	415	10.19	438	11.57	460	13.20	501	16.24	574	23.49
25190	1300	254	2.02	272	2.53	285	3.05	292	3.47	304	3.95	315	4.45	325	4.98	338	5.55	359	6.75	382	8.03	402	9.39	423	10.78	442	12.37	463	13.83	503	17.10	577	24.18
27130	1400	270	2.42	288	2.98	301	3.53	309	4.06	318	4.51	330	5.03	340	5.59	350	6.17	371	7.39	392	8.70	411	10.11	431	11.56	449	13.10	468	14.67	506	18.09	579	25.25
29070	1500	...	.....	304	3.50	318	4.07	328	4.67	336	5.24	343	5.68	352	6.22	364	6.85	382	8.09	401	9.45	422	10.86	441	12.35	458	13.93	475	15.56	511	18.90	581	26.30
31010	1600	...	.....	319	4.06	333	4.66	345	5.32	352	5.88	358	6.43	368	6.99	376	7.56	396	8.89	414	10.26	433	11.74	450	13.25	468	14.83	486	16.49	517	19.97	583	27.50
32950	1700	...	.....	335	4.70	350	5.37	360	6.01	371	6.70	376	7.33	383	7.85	393	8.49	409	9.78	427	11.18	443	12.67	461	14.22	478	15.83	494	17.52	526	21.08	587	28.89
34880	1800	...	.....	353	5.43	366	6.12	375	6.76	389	7.54	395	8.23	398	8.83	406	9.41	426	10.77	440	12.17	459	13.77	474	15.34	490	16.96	507	18.67	536	22.28	595	30.07
36820	1900	...	.....	370	6.19	382	6.94	393	7.64	402	8.37	414	9.19	418	9.89	422	10.52	438	11.81	453	13.24	468	14.79	485	16.45	499	18.12	514	19.86	543	23.54	601	31.55
38760	2000	...	.....	...	.....	398	7.91	409	8.60	419	9.34	428	10.18	435	10.94	443	11.78	456	13.14	468	14.42	484	16.04	497	17.67	512	19.41	528	21.20	555	24.90	611	32.97
40700	2100	...	.....	...	.....	415	8.86	425	9.62	435	10.45	443	11.23	452	12.08	460	12.94	471	14.46	483	15.84	495	17.32	512	19.05	524	20.80	538	22.58	564	26.38	621	34.61
42640	2200	...	.....	...	.....	432	9.92	440	10.66	452	11.54	460	12.39	469	13.27	478	14.22	485	15.81	497	17.24	511	18.78	524	20.43	536	22.25	550	24.10	578	27.97	630	36.35
44570	2300	...	.....	...	.....	448	11.08	455	11.94	470	12.89	476	13.65	484	14.55	493	15.52	508	17.41	511	18.80	526	20.44	540	22.09	551	23.89	566	25.79	589	29.68	640	38.15
46510	2400	Values below the heavy black lines are for Class II fans, which must be used whenever these ratings are desired.						472	13.19	483	14.13	493	15.04	501	15.97	509	16.96	523	18.92	531	20.65	541	22.15	553	23.85	562	25.51	576	27.46	604	31.40	652	40.13
50390	2600							506	16.05	516	17.15	526	18.18	532	18.98	541	20.14	557	22.21	568	24.31	574	26.17	584	27.83	595	29.61	608	31.52	630	35.61	...	.....
54260	2800							...	.....	551	20.65	559	21.70	...	.....	575	23.82	589	25.89	603	28.26	613	30.03	619	32.50	625	34.26	635	36.05	656	40.10	...	.....
58140	3000							...	.....	585	24.40	589	25.50	...	.....	...	.....	618	30.04	634	32.58	644	34.90	655	37.44	...	.....	...	.....	...	.....	...	.....
62020	3200							...	.....	...	.....	623	29.94	...	.....	...	.....	...	.....	...	.....	...	.....	...	.....	...	.....	...	.....	...	.....	...	.....

Table 7A.—Capacities of Double-width Double-inlet No. 10 High-speed Fans.

(American Blower Corp., Detroit, Mich., 1934)

Wheel Diam. 4.948 ft. Max. B.Hp. =324.2 (R.p.m./1000)³. For method of determining capacity of any other size of fan, see page 1–62.

DOUBLE WIDTH—DOUBLE INLET. OUTLET AREA = 34.15 SQ. FT.

Volume, cu. ft. per min.	Outlet Velocity, ft. per min.	1/8 in. R.P.		1/4 in. R.P.		3/8 in. R.P.		1/2 in. R.P.		5/8 in. R.P.		3/4 in. R.P.		7/8 in. R.P.	
		R.p.m.	B.Hp.	R.p.m.	B.Hp.	R.p.m.	B.Hp.	R.p.m.	B.Hp.	R.p.m.	B.Hp.	R.p.m.	B.Hp.	R.p.m.	B.Hp.
20490	600	127	0.63	154	1.11	182	1.65	208	2.24	230	2.86	251	3.49	271	3.98
23910	700	138	.81	162	1.34	187	1.92	212	2.57	233	3.25	253	3.96	273	4.59
27320	800	151	1.05	173	1.63	194	2.25	215	2.98	237	3.65	257	4.37	275	5.21
30740	900	165	1.35	185	1.97	204	2.64	221	3.35	241	4.12	260	4.93	278	5.78
34150	1000	178	1.69	197	2.37	215	3.09	231	3.85	248	4.65	265	5.50	282	6.40
37570	1100	192	2.10	210	2.83	226	3.60	242	4.41	257	5.27	272	6.15	288	7.08
40980	1200	206	2.59	223	3.36	238	4.19	253	5.05	267	5.95	280	6.89	293	7.85
44400	1300	221	3.15	238	3.99	251	4.85	262	5.75	278	6.72	292	7.69	300	8.69
47810	1400	235	3.78	254	4.68	264	5.61	276	6.55	290	7.60	303	8.64	311	9.67
51230	1500	...	.....	265	5.48	279	6.44	289	7.56	302	8.53	312	9.57	324	10.67
54640	1600	...	.....	278	6.35	292	7.38	303	8.47	313	9.53	324	10.66	336	11.83
58060	1700	...	.....	292	7.34	304	8.46	316	9.53	326	10.69	336	11.87	347	13.07
61470	1800	...	.....	307	8.32	319	9.55	329	10.70	341	12.02	350	13.20	358	14.39
64890	1900	...	.....	322	9.36	332	10.85	344	12.06	354	13.28	364	14.65	372	15.93
68300	2000	...	.....	...	.....	345	12.26	357	13.55	367	14.81	375	16.13	383	17.42
71720	2100	...	.....	...	.....	361	13.77	371	15.11	381	16.45	389	17.81	397	19.21
75130	2200	...	.....	...	.....	374	15.49	383	16.79	396	18.37	402	19.59	411	21.09
78550	2300	...	.....	...	.....	391	17.27	398	18.80	407	20.18	417	21.60	424	23.07
81960	2400	Values below the heavy black lines are for Class II fans, which must be used whenever these ratings are desired.						414	20.84	419	22.09	431	23.78	437	25.20
88790	2600							441	25.17	448	26.76	459	28.58	464	29.95
95620	2800							...	.....	478	32.25	485	33.85	...	.....
102450	3000							...	.....	508	38.12	512	39.92	...	.....
109280	3200	...	.....	...	.....	...	.....	...	.....	...	.....	541	46.67	...	.....
116110	3400	...	.....	...	.....	...	.....	...	.....	...	.....	...	.....	...	.....

Volume, cu. ft. per min.	Outlet Velocity, ft. per min.	1 in. R.P.		1 1/4 in. R.P.		1 1/2 in. R.P.		1 3/4 in. R.P.		2 in. R.P.		2 1/4 in. R.P.		2 1/2 in. R.P.		3 in. R.P.		4 in. R.P.	
		R.p.m.	B.Hp.	R.p.m.	B.Hp.	R.p.m.	B.Hp.	R.p.m.	B.Hp.	R.p.m.	B.Hp.	R.p.m.	B.Hp.	R.p.m.	B.Hp.	R.p.m.	B.Hp.	R.p.m.	B.Hp.
20490	600	288	4.90	321	6.46	351	8.18	...	.....	...	.....	...	.....	...	.....	...	.....	...	.....
23910	700	290	5.46	323	7.03	353	8.92	380	10.67	406	12.36	429	16.24	...	.....	...	.....	...	.....
27320	800	292	6.02	325	7.75	354	9.46	381	11.45	407	13.59	431	16.40	454	17.19	496	22.72	...	.....
30740	900	295	6.65	327	8.45	357	10.22	384	12.35	409	14.48	432	16.54	455	18.91	498	23.23	...	.....
34150	1000	299	7.30	330	9.22	359	11.22	386	13.27	411	15.24	434	17.75	457	20.24	499	24.49	574	35.21
37570	1100	303	8.04	333	10.07	362	12.19	388	14.35	413	16.64	436	18.99	459	21.44	500	26.32	575	36.94
40980	1200	309	8.84	337	10.96	365	13.19	391	15.46	415	17.91	439	20.30	461	22.87	503	27.91	577	39.24
44400	1300	316	9.74	342	11.94	369	14.25	394	16.66	419	19.11	441	21.77	463	24.34	505	29.71	578	41.00
47810	1400	326	10.74	349	13.00	374	15.37	398	17.90	422	20.18	444	23.14	466	25.95	507	31.66	580	43.37
51230	1500	333	11.85	356	14.19	379	16.62	403	19.24	426	21.95	446	24.69	469	27.64	510	33.39	583	45.71
54640	1600	346	13.00	367	15.46	388	18.02	408	20.66	431	23.44	452	26.29	474	29.16	513	35.29	585	48.19
58060	1700	359	14.34	379	16.91	397	19.53	416	22.24	436	25.05	457	28.03	477	31.07	516	37.33	587	50.78
61470	1800	369	15.67	392	18.43	408	21.11	427	24.02	443	26.89	463	29.89	483	32.99	521	39.58	591	53.20
64890	1900	380	17.21	400	20.00	419	22.87	435	25.70	452	28.76	469	31.82	487	34.96	524	41.68	594	55.74
68300	2000	394	18.91	412	21.84	430	24.74	446	27.73	462	30.80	478	33.96	495	37.21	530	44.04	598	58.55
71720	2100	406	20.71	425	23.69	441	26.73	457	29.76	474	33.02	488	36.26	503	39.54	538	46.55	602	61.38
75130	2200	420	22.60	434	25.64	451	28.79	468	31.97	484	35.35	497	38.63	512	42.14	543	49.16	606	64.37
78550	2300	433	24.69	450	27.86	462	30.91	480	34.36	494	37.90	510	41.20	524	44.85	551	51.97	611	67.40
81960	2400	447	26.82	462	30.13	475	33.49	491	36.87	506	40.41	519	43.88	534	47.46	562	55.12	617	70.72
88790	2600	473	31.70	489	35.29	501	38.65	514	42.32	528	46.10	543	49.87	559	53.86	584	61.61	632	77.91
95620	2800	502	37.54	516	41.11	527	44.95	541	49.34	553	51.88	565	56.39	580	60.38	604	68.72	...	.....
102450	3000	...	.....	541	47.36	554	51.54	567	56.31	578	59.52	590	63.67	602	68.88	625	76.56	...	.....
109280	3200	...	.....	...	.....	583	59.16	595	63.22	607	67.75	616	71.72	625	76.25	647	85.15	...	.....
116110	3400	...	.....	...	.....	608	67.33	623	71.57	631	76.16	643	80.81	...	.....	...	.....	...	.....

Table 8.—Dimensions, inches, of Sirocco Fans. (See Figs. 9 to 13)
(American Blower Corp., Detroit, Mich., 1934)

Fan Size	Wheel Diam.	Inlet Diam.	A	B	C	D	E	F	G	GG	H	J	JJ	K	KK	L
2	12	12 1/2	7 5/8	10 1/8	8 7/8	11 27/32	11 11/32	12 3/8	9 15/16	15 7/8	13	13 7/16	16 7/16	9 3/16	12 3/16	....
2 1/2	15	15 1/4	9 17/32	12 11/16	11 1/16	14 13/16	13 1/4	15 3/8	12 5/16	19 7/8	16	15 1/2	19 5/16	11	14 13/16	....
3	18	18 1/2	11 13/32	15 3/16	13 9/32	17 3/4	15 1/8	18 3/8	14 11/16	23 13/16	18 5/8	18 3/16	22 3/4	12 3/4	17 5/16	....
3 1/2	21	21 1/4	13 5/16	17 11/16	15 1/2	20 5/8	17 1/32	21 3/8	17 1/16	27 13/16	21 1/2	20 1/16	25 7/16	14 5/8	20	....
4	24	25	15 1/4	20 1/4	17 11/16	23 7/16	18 31/32	24 3/8	19 7/16	31 5/8	24 1/4	22 5/16	28 11/16	16 1/8	22 1/4	....
4 1/2	27	28	17 1/8	22 25/32	19 29/32	26 5/8	20 27/32	27 3/8	21 13/16	35 7/16	28	24 9/16	31 3/8	18 1/8	24 15/16	....
5	30	32	19	25 5/16	22 1/8	29 5/16	22 23/32	30 7/16	24 1/4	39 1/2	30	26 1/16	33 11/16	19 3/4	27 3/8	....
5 1/2	33	35	20 7/8	27 27/32	24 11/32	32 7/16	24 9/16	33 7/16	26 5/8	43 1/4	33 3/4	27	35 3/8	20 5/8	28 15/16	....
6	36	38	22 13/16	30 3/8	26 9/16	35 3/8	26 17/32	36 7/16	29	47 1/4	39 3/8	30 1/4	39 1/8	22 9/16	31 11/16	....
6 1/2	39 3/4	45 7/8	25	33 9/16	29 5/16	39 1/16	25	40 1/8	32	55 3/4	42 1/2	29 1/4	41 1/8	24 3/8	34 5/8	33
7	44	50 1/2	27 11/16	37 1/8	32 7/16	43 1/4	27 11/16	44 3/8	35 1/4	61 3/4	46 1/2	31 7/8	45 1/8	26	37 5/8	36
8	48 5/8	55 1/2	30 9/16	41 1/16	35 7/8	47 13/16	30 9/16	49	39	68 1/4	51	34 3/4	49 3/8	28 3/8	41 3/8	39 1/2
9	53 3/4	61	33 3/4	45 3/8	39 5/8	52 7/8	33 3/4	54 1/8	43	75 1/2	56 1/2	38 3/4	55	31 5/8	46	43 1/2
10	59 3/8	67 1/8	37 5/16	50 1/8	43 3/4	58 3/8	37 5/16	59 3/4	47 1/2	83 1/4	62 1/2	43	60 7/8	34 7/8	50 7/8	48
11	65 3/4	73 7/8	41 1/4	55 1/2	48 1/2	64 5/8	41 1/4	66 1/4	52 5/8	91 7/8	69	46 1/2	72	38 3/4	62	53 1/2
12	72 5/8	81 1/8	45 9/16	61 5/16	53 9/16	71 3/8	45 9/16	73 1/8	58 1/8	101 5/8	76	51 1/2	78 3/4	43 7/8	68 3/4	58 1/2
13 1/2	80 3/8	89 1/2	50 7/16	67 7/8	59 1/4	79	50 7/16	80 7/8	64 1/8	112 5/8	84	57 1/2	87 3/4	48	75 3/4	65
15	88 3/4	98 1/2	55 11/16	74 15/16	65 7/16	87 1/4	55 11/16	89 1/4	70 7/8	124 5/8	92 1/2	61 5/8	94 1/2	52 1/8	82 1/2	71
16 1/2	98 1/4	110	62 3/8	82 15/16	72 9/16*	96 5/8	62 3/8	98 1/4‡	77 3/4‡	137 7/8	....	87	111	73 1/2	95	78§
18	108 1/2	121	68 5/8	91 5/8	80 1/8†	106 3/4	68 5/8	108 1/4‡	85 13/16‡	152 1/4	....	94	123 1/2	79 1/2	105 1/2	85§

Fan Size	Wheel Diam.	Inlet Diam.	M	N	P	Q	R	S	T	U	V	W	X	Y	Z	ZZ
2	12	12 1/2	....	....	....	....	....	....	....	....	....	....	....	....	....	....
2 1/2	15	15 1/4	....	....	....	....	....	....	....	....	....	....	....	....	....	....
3	18	18 1/2	....	....	....	....	....	....	....	....	....	....	....	....	....	....
3 1/2	21	21 1/4	....	....	....	....	....	....	....	....	....	....	....	....	....	....
4	24	25	....	....	....	....	....	....	....	....	....	....	....	....	....	....
4 1/2	27	28	....	....	....	....	....	....	....	....	....	....	....	....	....	....
5	30	32	....	....	....	....	....	....	....	....	....	....	....	....	....	....
5 1/2	33	35	....	....	....	....	....	....	....	....	....	....	....	....	....	....
6	36	38	....	....	....	....	....	....	....	....	....	....	....	....	....	....
6 1/2	39 3/4	45 7/8	33 13/16	25 1/8	37 1/2	19 3/16	29 9/16	36 1/16	31 7/16	27 1/16	35 5/8	27 3/4	25 1/2	36 5/8	31 1/2	55 1/4
7	44	50 1/2	37 3/8	27 13/16	41	21 1/4	32 11/16	39 15/16	34 3/4	30 3/8	39 3/8	30 3/4	28	40 1/2	34 3/4	61 1/4
8	48 5/8	55 1/2	41 5/16	30 11/16	45	23 1/2	36 1/8	44 1/16	38 7/16	33 9/16	43 9/16	34 1/2	30 1/2	44 11/16	38 1/2	67 3/4
9	53 3/4	61	45 11/16	33 7/8	49	26	39 15/16	49 11/16	42 1/2	37 1/8	48 1/8	38 1/2	33	49 7/16	42 1/2	75
10	59 3/8	67 1/8	50 7/16	37 7/16	54 1/2	28 11/16	44 1/16	54	46 15/16	40 15/16	53 1/4	42 1/4	37	54 1/2	47	82 3/4
11	65 3/4	73 7/8	55 7/8	41 3/8	60 1/2	31 3/4	48 7/8	59 3/8	51 15/16	45 5/16	58 15/16	47	40 1/2	60 7/16	52	91 1/4
12	72 5/8	81 1/8	61 11/16	45 11/16	66 1/2	35 1/16	53 15/16	65 11/16	57 3/8	50 1/8	65 1/16	52 5/8	44	66 7/8	57 1/2	101
13 1/2	80 3/8	89 1/2	68 1/4	50 9/16	74	38 13/16	59 11/16	72 9/16	63 1/2	55 3/8	72	57 7/8	49	73 7/8	63 1/2	112
15	88 3/4	98 1/2	75 5/16	55 13/16	81	42 7/8	65 7/8	80 1/4	70 1/8	61 1/8	79 1/2	64 3/8	53 1/2	81 5/8	70 1/4	124
16 1/2	98 1/4	110	88 5/16¶	66 5/8	....	47 1/2	78	....	77 1/2	67 3/16	88 1/16	71 1/4	59	95	....	137
18	108 1/2	121	97¶	73 1/4	....	52 1/2	85 1/2	....	85 3/4	74 3/8	97 1/8	79	65	104 1/2	....	151 3/8

On Up-blast Fans, 7/8 Housing, * 73; † 80 5/8. ‡ Inside Dimensions. § On double width fan these values are: No. 16 1/2, 76 1/2; No. 18, 84. ¶ Top Horizontal Discharge Fans only.

Table 9.—Dimensions, inches, of High-speed, Backward-curved Blade Fans. (See Figs. 9 to 13.)

(American Blower Corp., Detroit, Mich., 1934)

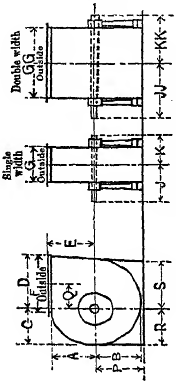
Fan Size	Wheel Diam.	Inlet Diam.	A	B	C	D	E	F	G	GG	H	J	JJ	K	KK	L
3	18	18 1/2	11 13/32	15 3/16	13 9/32	17 3/4	15	18 3/8	14 11/16	....	18 5/8	19 3/4	....	12 3/4	....	18 5/8
3 1/2	21	21 1/4	13 5/16	17 11/16	15 1/2	20 5/8	16 7/8	21 3/8	17 1/16	....	21 1/2	21 1/4	....	14 5/8	....	21 1/2
4	24	25	15 1/4	20 1/4	17 11/16	23 7/16	18 27/32	24 3/8	19 7/16	33 5/16	24 3/8	24 1/4	29 3/4	16 3/8	23 5/16	24 3/8
4 1/2	27	28	17 1/8	22 25/32	19 29/32	26 5/8	20 5/8	27 3/8	21 13/16	37 3/8	28	26 1/4	32	17 3/8	25 1/2	28
5	30	32	19	25 5/16	22 1/8	29 5/16	22 1/2	30 7/16	24 1/4	41 3/4	30 3/8	27 3/4	34 9/16	19 1/2	28 1/4	30 3/8
5 1/2	33	35	20 7/8	27 27/32	24 11/32	32 7/16	24 9/32	33 7/16	26 5/8	46 1/16	33 3/4	30 1/4	35 5/16	20 1/8	29 7/8	33 3/4
6	36	38	22 13/16	30 3/8	26 9/16	35 3/8	26 1/4	36 7/16	29	50 5/16	39 3/8	32 1/4	39 9/16	21 7/8	30 1/2	39 3/8
6 1/2	39 3/4	45 7/8	25	33 9/16	29 5/16	39 1/16	25	40 1/8	32	55 3/4	42 1/2	29 1/4	41 1/8	24 3/8	34 5/8	33
7	44	50 1/2	27 11/16	37 1/8	32 7/16	43 1/4	27 11/16	44 3/8	35 1/4	61 3/4	46 1/2	31 7/8	45 5/8	26	37 5/8	36
8	48 5/8	55 1/2	30 9/16	41 1/16	35 7/8	47 13/16	30 9/16	49	39	68 1/4	51	34 3/4	49 3/8	28 3/8	41 3/8	39 1/2
9	53 3/4	61	33 3/4	45 3/8	39 5/8	52 7/8	33 3/4	54 1/8	43	75 1/2	56 1/2	38 3/4	55	31 5/8	46	43 1/2
10	59 3/8	67 1/8	37 5/16	50 1/8	43 3/4	58 3/8	37 5/16	59 3/4	47 1/2	83 1/4	62 1/2	43	60 7/8	34 7/8	50 7/8	48
11	65 3/4	73 7/8	41 1/4	55 1/2	48 1/2	64 5/8	41 1/4	66 1/4	52 5/8	91 7/8	69	46 1/2	72	38 3/4	62	53 1/2
12	72 5/8	81 1/8	45 9/16	61 5/16	53 9/16	71 3/8	45 9/16	73 1/8	58 1/8	101 5/8	76	51 1/2	78 3/4	43 7/8	68 3/8	58 1/2
13 1/2	80 3/8	89 1/2	50 7/16	67 7/8	59 1/4	79	50 7/16	80 7/8	64 1/8	112 5/8	....	57 3/4	90 3/4	48 1/4	75 3/4	65
15	88 3/4	98 1/2	55 11/16	74 15/16	65 7/16	87 1/4	55 11/16	89 1/4	70 7/8	124 5/8	....	61 7/8	97 1/2	52 3/8	82 1/2	71
16 1/2	98 1/4	110	62 3/8	82 15/16	72 9/16	96 5/8	62 3/8	98 1/4*	77 3/4*	137*	....	87	111	73 1/2	95	78
18	108 1/2	121	68 5/8	91 5/8	80 1/8	106 3/4	68 5/8	108 1/2*	85 13/16*	151 3/8*	....	94	123 1/2	79 1/2	105 1/2	85

Fan Size	Wheel Diam.	Inlet Diam.	M	N	P	Q	R	S	T	U	V	W	X	Y	Z	ZZ
3	18	18 1/2	....	....	18 5/8	....	....	....	....	....	....	....	....	....	....	....
3 1/2	21	21 1/4	....	....	21 1/2	....	....	....	....	....	....	....	....	....	....	....
4	24	25	....	....	24 3/8	....	....	....	....	....	....	....	....	....	....	....
4 1/2	27	28	....	....	28	....	....	....	....	....	....	....	....	....	....	....
5	30	32	....	....	30 3/8	....	....	....	....	....	....	....	....	....	....	....
5 1/2	33	35	....	....	33 3/4	....	....	....	....	....	....	....	....	....	....	....
6	36	38	....	....	39 3/8	....	....	....	....	....	....	....	....	....	....	....
6 1/2	39 3/4	45 7/8	33 13/16	25 1/8	37 1/2	19 3/16	29 9/16	36 1/16	31 7/16	27 7/16	35 5/8	27 3/4	25 1/2	36 5/8	31 1/2	....
7	44	50 1/2	37 3/8	27 13/16	41	21 1/4	32 11/16	39 15/16	34 3/4	30 3/8	39 3/8	30 3/4	28	40 1/2	34 3/4	....
8	48 5/8	55 1/2	41 5/16	30 11/16	45	23 1/2	36 1/8	44 1/16	38 7/16	33 9/16	43 9/16	34 1/2	30 1/2	44 11/16	38 1/2	....
9	53 3/4	61	45 11/16	33 7/8	49	26	39 15/16	49 11/16	42 1/2	37 1/8	48 1/8	38 1/2	33	49 7/16	42 1/2	....
10	59 3/8	67 1/8	50 7/16	37 7/16	54 1/2	28 11/16	44 1/16	54	46 15/16	40 15/16	53 1/4	42 1/4	37	54 1/2	47	....
11	65 3/4	73 7/8	55 7/8	41 3/8	60 1/2	31 3/4	48 7/8	59 3/8	51 15/16	45 5/16	58 15/16	47	40 1/2	60 7/16	52	....
12	72 5/8	81 1/8	61 11/16	45 11/16	66 1/2	35 1/16	53 15/16	65 11/16	57 3/8	50 1/8	65 1/16	52 5/8	44	66 7/8	57 1/2	....
13 1/2	80 3/8	89 1/2	68 1/4	50 9/16	....	38 13/16	59 11/16	....	....	....	....	....	49	73 7/8	....	....
15	88 3/4	98 1/2	75 5/16	55 13/16	....	42 7/8	65 7/8	....	....	....	....	....	53 1/2	81 5/8	....	....
16 1/2	98 1/4	110	88 5/16	66 5/8	....	47 1/2	78	....	....	....	....	....	59	95	....	....
18	108 1/2	121	97	73 1/4	....	52 1/2	85 1/2	....	....	....	....	....	65	104 1/4	....	....

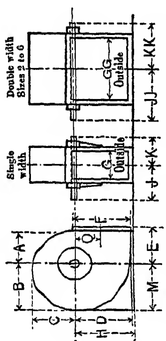
* Inside Dimensions.

for special work. For handling stringy or fibrous material it is fitted with a cast-iron wheel, with smooth curves and surfaces presenting no projections to catch fiber and clog. For long shavings a steel plate straight blade fan is used. The slow speed wheel has steel plate blades with the tips curved forward. For cotton gin work, and for moving dust, smoke or vapors, the so-called cotton wheel is used. This wheel has 18 forward-curved blades. The capacity of the fan fitted with the three latter types of wheel is given in Table 10. See Table 11 for dimensions of all sizes.

Table 12 gives the capacities of blowers for handling air or gases. They are designed to operate at pressures up to 20 in. water gage, and are used for forced draft, cooling molds, blowing scale from dies and forging hammers, ejector exhaust systems for removing inflammable or corrosive fumes, etc. Sizes 3 to 7 are of steel plate construction. Sizes, 0, 00 and 000 are of cast iron. Dimensions are given in Table 13.

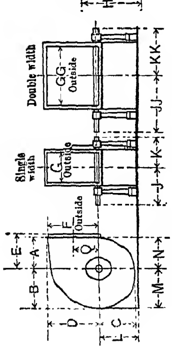


UP-BLAST DISCHARGE – FULL HOUSING



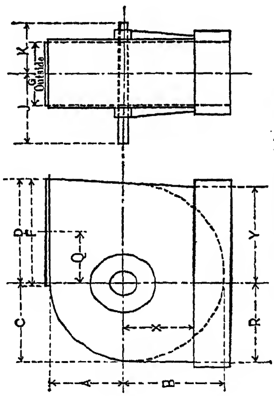
BOTTOM HORIZONTAL DISCHARGE – FULL HOUSING

Fig. 10



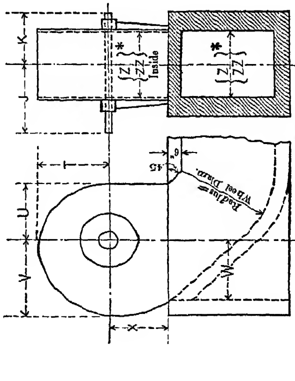
TOP HORIZONTAL DISCHARGE–FULL HOUSING

Fig. 9



UP-BLAST DISCHARGE – 7/8 HOUSING

Fig. 13



* Z = Single width fans
ZZ = Double width fans

BOTTOM HORIZONTAL DISCHARGE – 7/8 HOUSING

Fig. 12

Table 10.—Capacities of No. 50 Steel Plate Exhaust Fan with Various Types of Wheel

(American Blower Corp., Detroit, Mich., 1934)

Diam. of Wheel = Slow Speed, 34 in.; Long Shaving, 33 in.; Cotton, 34 in.; Diam. of Inlet = 21 in.; Area of Inlet = 2.40 sq. ft.; Area of Outlet = 2.25 sq. ft.

To determine capacity of any other size of fan, see p. 1–62.

1/2 in. Static Pressure

Cu. ft. per min.	Slow Speed R.p.m.	Slow Speed B.Hp.	Long Shaving R.p.m.	Long Shaving B.Hp.	Cotton R.p.m.	Cotton B.Hp.
964	260	0.147	290	0.138	261	0.14
1218	260	.172	295	.168	262	.166
1461	262	.20	300	.198	263	.193
1724	265	.234	307	.237	263	.226
2045	270	.282	315	.289	264	.273
2320	275	.329	324	.340	265	.32
2650	283	.395	336	.407	269	.384
2970	291	.468	350	.485	277	.459
3420	304	.585	366	.607	289	.58

1 in. Static Pressure

Cu. ft. per min.	Slow Speed R.p.m.	Slow Speed B.Hp.	Long Shaving R.p.m.	Long Shaving B.Hp.	Cotton R.p.m.	Cotton B.Hp.
1364	368	0.415	410	0.390	369	0.396
1723	368	.485	417	.475	371	.468
2070	370	.565	424	.560	372	.545
2440	375	.661	437	.670	372	.638
2890	382	.796	445	.818	373	.77
3280	389	.93	453	.962	374	.904
3750	400	1.115	475	1.150	381	1.085
4200	412	1.322	495	1.372	392	1.297
4835	430	1.652	518	1.716	409	1.64

1 1/2 in. Static Pressure

Cu. ft. per min.	Slow Speed R.p.m.	Slow Speed B.Hp.	Long Shaving R.p.m.	Long Shaving B.Hp.	Cotton R.p.m.	Cotton B.Hp.
1670	451	0.762	502	0.717	452	0.726
2110	451	.891	511	.872	454	.86
2530	454	1.038	520	1.03	455	1.002
2990	459	1.214	532	1.232	455	1.073
3540	467	1.462	546	1.50	456	1.416
4020	477	1.708	561	1.767	456	1.66
4590	490	2.05	582	2.11	466	1.992
5140	504	2.43	606	2.52	479	2.38
5920	526	3.03	634	3.155	500	3.01

2 in. Static Pressure

Cu. ft. per min.	Slow Speed R.p.m.	Slow Speed B.Hp.	Long Shaving R.p.m.	Long Shaving B.Hp.	Cotton R.p.m.	Cotton B.Hp.
1938	521	1.175	580	1.10	522	1.12
2436	521	1.374	590	1.34	524	1.33
2522	524	1.60	600	1.58	526	1.545
3448	530	1.873	614	1.89	526	1.81
4090	540	2.255	630	2.31	528	2.18
4640	550	2.635	648	2.72	530	2.56
5300	566	3.16	672	3.22	538	3.07
5940	582	3.75	700	3.88	554	3.67
6840	608	4.68	732	4.85	578	4.64

2 1/2 in. Static Pressure

Cu. ft. per min.	Slow Speed R.p.m.	Slow Speed B.Hp.	Long Shaving R.p.m.	Long Shaving B.Hp.	Cotton R.p.m.	Cotton B.Hp.
2155	583	1.64	648	1.54	585	1.56
2725	583	1.92	660	1.874	586	1.85
3270	586	2.23	671	2.21	587	2.16
3850	593	2.62	687	2.645	587	2.53
4570	604	3.15	705	3.22	588	3.04
5190	616	3.67	725	3.79	592	3.57
5925	633	4.4	752	4.54	601	4.28
6640	650	5.22	783	5.41	620	5.12
7650	680	6.52	818	6.77	647	6.47

3 in. Static Pressure

Cu. ft. per min.	Slow Speed R.p.m.	Slow Speed B.Hp.	Long Shaving R.p.m.	Long Shaving B.Hp.	Cotton R.p.m.	Cotton B.Hp.
2365	639	2.16	710	2.025	640	2.05
2985	639	2.52	722	2.465	642	2.44
3580	642	2.94	735	2.90	643	2.83
4220	650	3.44	752	3.47	643	3.32
5010	661	4.14	772	4.24	644	4.0
5690	675	4.83	794	4.98	649	4.69
6490	693	5.79	823	5.96	659	5.64
7280	713	6.87	857	7.11	678	6.73
8380	745	8.58	897	8.90	708	8.51

4 in. Static Pressure

Cu. ft. per min.	Slow Speed R.p.m.	Slow Speed B.Hp.	Long Shaving R.p.m.	Long Shaving B.Hp.	Cotton R.p.m.	Cotton B.Hp.
2728	736	3.32	820	3.12	738	3.16
3446	736	3.88	834	3.80	742	3.75
4140	740	4.52	848	4.48	744	4.36
4880	750	5.29	868	5.36	744	5.11
5780	764	6.37	890	6.54	746	6.16
6560	778	7.44	916	7.69	748	7.23
7500	800	8.91	950	9.20	762	8.68
8400	824	10.58	990	10.97	784	10.37
9670	860	13.2	1030	13.72	819	13.10

5 in. Static Pressure

Cu. ft. per min.	Slow Speed R.p.m.	Slow Speed B.Hp.	Long Shaving R.p.m.	Long Shaving B.Hp.	Cotton R.p.m.	Cotton B.Hp.
3050	824	4.64	918	4.35	826	4.42
3850	824	5.42	933	5.30	828	5.24
4620	829	6.31	949	6.24	829	6.09
5450	839	7.39	971	7.48	829	7.14
6470	854	8.9	996	9.11	831	8.61
7330	870	10.4	1025	10.72	837	10.10
8380	895	12.47	1064	12.84	851	12.12
9400	921	14.78	1108	15.30	876	14.48
10820	962	18.5	1159	19.15	915	18.3

6 in. Static Pressure

Cu. ft. per min.	Slow Speed R.p.m.	Slow Speed B.Hp.	Long Shaving R.p.m.	Long Shaving B.Hp.	Cotton R.p.m.	Cotton B.Hp.
3340	902	6.09	1004	5.73	904	5.91
4220	902	7.13	1022	6.97	907	6.89
5060	908	8.3	1040	8.24	910	8.01
5980	918	9.72	1064	9.85	910	9.39
7080	934	11.7	1092	12.00	912	11.32
8040	954	13.66	1122	14.13	916	13.28
9180	980	16.38	1164	16.88	932	15.95
10280	1008	19.42	1212	20.16	958	19.05
11840	1052	24.3	1268	25.24	1000	24.1

7 in. Static Pressure

Cu. ft. per min.	Slow Speed R.p.m.	Slow Speed B.Hp.	Long Shaving R.p.m.	Long Shaving B.Hp.	Cotton R.p.m.	Cotton B.Hp.
3610	974	7.67	1086	7.20	976	7.31
4560	974	8.97	1105	8.77	979	8.67
5470	980	10.44	1123	10.34	982	10.10
6450	991	12.24	1150	12.37	982	11.82
7650	1009	14.72	1180	15.10	984	14.26
8680	1030	17.2	1213	17.75	996	16.72
9920	1058	20.06	1258	21.27	1006	20.1
11120	1089	24.45	1310	25.3	1035	24.0
12800	1138	30.6	1370	31.70	1082	30.3

8 in. Static Pressure

Cu. ft. per min.	Slow Speed R.p.m.	Slow Speed B.Hp.	Long Shaving R.p.m.	Long Shaving B.Hp.	Cotton R.p.m.	Cotton B.Hp.
3876	1042	9.38	1160	8.80	1044	8.95
4872	1042	10.97	1180	10.72	1048	10.6
5044	1048	12.78	1200	12.64	1052	12.33
6896	1060	14.97	1228	15.12	1052	14.44
8180	1080	18.0	1260	18.48	1056	17.42
9280	1100	21.0	1296	21.76	1060	20.45
10600	1132	25.2	1344	25.76	1076	24.55
11880	1164	29.9	1400	31.04	1108	29.3
13680	1216	37.4	1464	38.80	1156	37.05

9 in. Static Pressure

Cu. ft. per min.	Slow Speed R.p.m.	Slow Speed B.Hp.	Long Shaving R.p.m.	Long Shaving B.Hp.	Cotton R.p.m.	Cotton B.Hp.
4092	1105	11.2	1230	10.53	1108	10.68
5169	1105	13.1	1251	12.84	1110	12.66
6210	1112	15.25	1272	15.13	1113	14.73
7320	1124	17.87	1302	18.10	1113	17.25
8670	1144	21.5	1335	22.10	1114	20.8
9840	1168	25.1	1374	26.00	1122	24.4
11250	1200	30.1	1425	31.1	1142	29.3
12600	1235	35.7	1485	37.05	1174	35.0
14505	1290	44.6	1554	46.30	1227	44.25

10 in. Static Pressure

Cu. ft. per min.	Slow Speed R.p.m.	Slow Speed B.Hp.	Long Shaving R.p.m.	Long Shaving B.Hp.	Cotton R.p.m.	Cotton B.Hp.
4310	1166	13.1	1296	12.32	1170	12.5
5450	1166	15.32	1320	14.99	1172	14.8
6540	1174	17.85	1342	17.68	1174	17.28
7700	1186	20.9	1374	21.16	1174	20.2
9140	1208	25.15	1410	25.76	1176	24.34
10380	1232	29.4	1450	30.32	1184	28.55
11850	1266	35.2	1504	36.32	1202	34.27

11 in. Static Pressure

Cu. ft. per min.	Slow Speed R.p.m.	Slow Speed B.Hp.	Long Shaving R.p.m.	Long Shaving B.Hp.	Cotton R.p.m.	Cotton B.Hp.
4525	1222	15.12	1361	14.2	1225	14.42
5720	1222	17.7	1385	17.26	1228	17.08
6850	1230	20.6	1407	20.35	1231	19.88
8090	1243	24.14	1440	24.30	1231	23.28
9580	1265	29.0	1479	29.70	1233	28.1
10900	1290	33.9	1520	34.90	1241	32.93
....	....	....	....	....	....	....

12 in. Static Pressure

Cu. ft. per min.	Slow Speed R.p.m.	Slow Speed B.Hp.	Long Shaving R.p.m.	Long Shaving B.Hp.	Cotton R.p.m.	Cotton B.Hp.
4730	1278	17.25	1420	16.20	1280	16.4
5970	1278	20.2	1444	19.72	1284	19.5
7160	1284	23.5	1470	23.20	1286	22.7
8440	1300	27.5	1504	27.76	1286	26.6
10020	1322	33.1	1544	33.92	1288	32.0
....	....	....	....	....	....	....
....	....	....	....	....	....	....

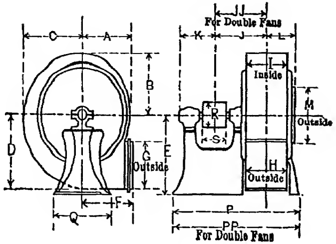


FIG. 14
Sizes 25–60

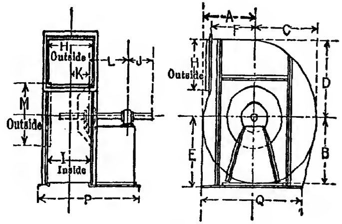


FIG. 15
Sizes 70–90

Table 11.—Dimensions of Steel Plate Exhaust Fans

(American Blower Corp., Detroit, Mich., 1934)

Fan No.	Diam. of Wheel, in.			Diam. of Inlet, in.	Dimensions, in. (see Figs. 14 and 15)																	Pulley			
																						Long Shaving and Cotton		Slow Speed	
	Slow Speed	Long Shaving	Cotton		A	B	C	D	E	F	G	H	I	J	JJ	K	L	M	P	PP	Q	R	S	R	S
25	....	17	19	11	11 1/2	12 3/8	13 1/4	14 1/8	15	13 1/2	10	10	9 1/2	13 3/4	....	10 3/16	7 1/4	11	33 1/8	.....	13 1/2	5	5 1/2	..	.....
30	22	20	22	13	13 3/8	14 7/16	15 9/16	16 5/8	18 5/8	15	11 1/2	11 1/2	11	15	16	11 3/16	8	13	36 3/8	52 1/4	16	6	6 1/2	7	7 1/2
35	25	23	25	15	15 1/4	16 5/8	18 1/16	19 7/16	21 1/2	16 3/8	13	13	12 1/2	16 13/16	17 3/8	12 1/8	9	15	39 3/4	56 1/2	17 1/2	7	7 1/2	8	8 1/2
40	28	26	28	17	17	18 5/8	20 3/16	21 13/16	24 1/4	17 13/16	14 5/8	14 5/8	14 1/8	18 7/16	19 7/16	13 1/4	10 1/16	17	44 5/8	62 1/4	19 1/2	8	8 1/2	9	9 1/2
45	31	29	31	19	19 1/8	21	22 7/8	24 3/4	28	19 5/8	16 5/8	16 5/8	16 1/8	20 11/16	21 11/16	14 7/16	11 5/16	19	48 5/8	69 3/4	23	9	9 1/2	10	10 1/2
50	34	33	34	21	21 1/4	23 3/8	25 1/2	27 5/8	30	21 3/8	18 3/8	18 3/8	17 7/8	22 9/16	23 5/16	15 3/8	12 3/16	21	54 7/8	71	25	10	10 1/2	11	11 1/2
55	38	36	....	23	22 11/16	25 3/16	27 3/4	30 1/4	33 3/4	22 1/4	20	20	19 1/2	23 9/16	24 3/8	14 7/8	13 1/4	23	54 15/16	75	28	11	11 1/2	12	12 1/2
60	42	40	....	25	24 3/8	27 1/4	30 3/16	33 1/8	39 3/8	23 3/4	21 7/8	21 7/8	21 3/8	25 7/16	27 7/16	15 7/8	14 3/16	25	60 1/8	83	33	12	12 1/2	14	12 1/2
70	46	46	....	29	27	30 1/4	33 1/2	37	33 1/2	24	25 3/8	25 3/8	24 1/2	15	....	11	22	29	56	.....	54 1/2	14	12 1/2	14	12 1/2
80	53	52	....	33	30 3/4	34 1/2	38 1/4	42	37 1/2	27	27 7/8	25 3/8	27	18	....	12 1/2	25	33	63	.....	62	16	14 1/2	16	14 1/2
90	60	58	....	37	34 1/2	38 5/8	42 5/8	46 5/8	42	30 3/4	31 3/8	31 3/8	30 1/2	19	....	13 1/2	27	37	68 3/4	.....	69 1/2	18	14 1/2	18	14 1/2

No. of Fan	1 in. S.P.			2 in. S.P.			3 in. S.P.			4 in. S.P.			5 in. S.P.			6 in. S.P.			7 in. S.P.		
	C.F.M.	R.p.m.	B.Hp.	C.F.M.	R.p.m.	B.Hp.	C.F.M.	R.p.m.	B.Hp.	C.F.M.	R.p.m.	B.Hp.	C.F.M.	R.p.m.	B.Hp.	C.F.M.	R.p.m.	B.Hp.	C.F.M.	R.p.m.	B.Hp.
000	40	2040	0.01	57	2880	0.04	69	3530	0.07	80	4080	0.11	90	4570	0.15	98	5000	0.20	......	......	......
	63	2150	.02	89	3030	.06	109	3720	.11	126	4300	.16	141	4820	.23	......	......	......	......	......	......
	97	2380	.04	137	3360	.10	168	4120	.18	194	4760	.26	......	......	......	......	......	......	......	......	......
	155	2760	.08	218	3900	.21	268	4775	.38	......	......	......	......	......	......	......	......	......	......	......	......
00	74	1510	.03	105	2135	.07	128	2615	.13	148	3020	.20	166	3380	.27	181	3700	0.36	196	4000	0.43
	116	1590	.04	164	2250	.11	201	2760	.19	232	3180	.29	260	3570	.41	284	3890	0.53	......	......	......
	178	1760	.07	251	2490	.18	308	3045	.33	356	3520	.50	395	3950	.70	......	......	......	......	......	......
	284	2040	.14	401	2980	.38	492	3535	.70	568	4080	1.10	......	......	......	......	......	......	......	......	......
0	108	1325	.04	152	1910	.10	187	2295	.18	216	2650	0.26	242	2970	.37	264	3220	0.49	286	3510	0.56
	174	1400	.06	246	1980	.15	301	2420	.28	348	2800	0.42	390	3140	.59	426	3430	0.77	460	3710	0.93
	258	1525	.09	364	2150	.25	446	2640	.44	516	3050	0.68	579	3420	.94	630	3720	1.24	......	......	......
	387	1800	.16	546	2540	.46	670	3120	.84	774	3600	1.28	869	4040	1.78	......	......	......	......	......	......
3	289	747	.08	410	1054	.23	502	1295	.42	578	1494	0.66	647	1673	0.92	709	1831	1.21	765	1975	1.53
	437	784	.12	617	1105	.34	753	1357	.63	871	1567	0.97	973	1751	1.36	1065	1918	1.78	1152	2072	2.25
	627	843	.19	888	1198	.54	1087	1467	.99	1253	1692	1.52	1401	1889	2.13	1535	2073	2.8	1660	2235	3.53
	912	969	.34	1287	1369	.97	1580	1681	1.79	1823	1940	2.76	2030	2167	3.86	2235	2380	5.09	2410	2560	6.41
4	369	662	.10	522	937	.29	638	1148	0.54	736	1325	0.84	825	1483	1.18	903	1624	1.54	975	1752	1.95
	556	695	.15	786	982	.43	960	1203	0.80	1110	1390	1.24	1241	1553	1.73	1358	1701	2.27	1468	1838	2.87
	798	749	.24	1131	1062	.68	1384	1300	1.26	1597	1500	1.94	1786	1676	2.72	1957	1839	3.58	2115	1983	4.49
	1162	859	.44	1640	1214	1.24	2013	1490	2.29	2325	1720	3.52	2592	1920	4.92	2850	2110	6.48	3070	2272	8.16
5	458	595	.13	647	842	0.36	792	1032	0.67	912	1190	1.04	1023	1332	1.46	1120	1460	1.92	1210	1573	2.42
	689	624	.19	975	882	0.54	1191	1080	0.99	1377	1248	1.54	1540	1395	2.15	1683	1527	2.82	1820	1650	3.56
	990	673	.30	1408	953	0.85	1716	1168	1.57	1980	1347	2.41	2213	1505	3.37	2425	1651	4.45	2625	1780	5.57
	1440	772	.54	2035	1090	1.54	2500	1339	2.84	2880	1546	4.37	3215	1725	6.10	3535	1895	8.05	3810	2040	10.13

6	556	539	.15	787	763	0.44	962	936	0.82	1107	1079	1.26	1243	1208	1.78	1361	1324	2.33	1468	1427	2.94
	837	567	.23	1183	800	0.66	1445	980	1.21	1672	1133	1.87	1871	1265	2.61	2048	1386	3.43	2210	1496	4.32
	1204	610	.36	1704	865	1.03	2085	1059	1.91	2408	1223	2.93	2690	1365	4.10	2950	1498	5.40	3190	1616	6.77
	1751	700	.66	2470	988	1.87	3035	1215	3.45	3500	1402	5.31	3900	1565	7.41	4290	1720	9.78	4625	1850	12.29
7	781	455	.22	1104	644	0.63	1352	790	1.16	1556	911	1.78	1746	1020	2.50	1913	1117	3.28	2065	1205	4.13
	1177	478	.32	1662	675	0.93	2030	827	1.70	2350	956	2.63	2630	1068	3.67	2875	1170	4.82	3110	1263	6.08
	1690	515	.51	2392	730	1.45	2930	894	2.68	3380	1031	4.12	3780	1153	5.76	4140	1264	7.58	4480	1364	9.52
	2460	591	.93	3470	835	2.63	4270	1025	4.85	4920	1183	7.46	5480	1320	10.4	6030	1450	13.73	6500	1560	17.26

No. of Fan	8 in. S.P.			10 in. S.P.			12 in. S.P.			14 in. S.P.			16 in. S.P.			18 in. S.P.			20 in. S.P.		
	C.F.M.	R.p.m.	B.Hp.	C.F.M.	R.p.m.	B.Hp.	C.F.M.	R.p.m.	B.Hp.	C.F.M.	R.p.m.	B.Hp.	C.F.M.	R.p.m.	B.Hp.	C.F.M.	R.p.m.	B.Hp.	C.F.M.	R.p.m.	B.Hp.
3	820	2113	1.87	916	2370	2.63	1000	2590	3.43	1085	2790	4.33	1156	2990	5.28	1229	3175	6.34	1296	3340	7.41
	1234	2215	2.76	1378	2480	3.85	1507	2710	5.05	1630	2932	6.38	1743	3135	7.78	1850	3330	9.31	1947	3500	10.87
	1775	2393	4.32	1984	2672	6.05	2170	2932	7.94	2347	3170	10.00	2510	3385	12.20	2665	3590	14.63	2805	3780	17.10
	2575	2740	7.81	2880	3068	10.95	3160	3360	14.37	3410	3620	18.17	3650	3880	22.10	......	......	......	......	......	......
4	1044	1874	2.38	1167	2100	3.34	1276	2296	4.37	1380	2475	5.52	1472	2650	6.73	1564	2815	8.08	1650	2966	9.44
	1572	1965	3.51	1757	2200	4.91	1920	2406	6.43	2075	2600	8.13	2220	2780	9.92	2355	2950	11.85	2482	3106	13.86
	2262	2124	5.50	2530	2370	7.71	2768	2600	10.12	2990	2810	12.74	3194	3000	15.56	3393	3185	18.64	3572	3352	21.76
	3280	2428	9.95	3670	2720	13.94	4026	2980	18.32	4340	3210	23.15	4650	3440	28.2	......	......	......	......	......	......
5	1295	1684	2.96	1446	1887	4.15	1570	2065	5.43	1710	2225	6.84	1825	2380	8.35	1940	2530	10.00	2050	2660	11.72
	1950	1766	4.36	2175	1976	6.08	2380	2160	7.97	2575	2335	10.08	2755	2495	12.3	2920	2650	14.7	3080	2787	17.18
	2807	1908	6.82	3135	2130	9.56	3430	2335	12.54	3710	2525	15.80	3960	2690	19.30	4210	2860	23.10	4425	3010	27.00
	4070	2180	12.34	4550	2445	17.30	4990	2680	22.70	5380	2885	28.70	5770	3090	35.00	......	......	......	......	......	......
6	1575	1527	3.59	1757	1710	5.04	1922	1872	6.59	2080	2015	8.32	2220	2160	10.15	2355	2292	12.15	2490	2415	14.23
	2370	1600	5.29	2647	1792	7.40	2890	1957	9.69	3130	2120	12.25	3345	2265	14.93	3550	2406	17.85	3740	2530	20.9
	3410	1730	8.29	3810	1930	11.6	4170	2120	15.25	4510	2290	19.20	4820	2445	23.45	5120	2590	28.10	5380	2730	32.8
	4945	1980	14.98	5530	2215	21.00	6060	2430	27.6	6550	2613	34.9	7000	2800	42.5	......	......	......	......	......	......
7	2210	1290	5.05	2470	1445	7.08	2700	1580	9.27	2920	1703	11.65	3115	1821	14.26	3310	1936	17.1	3490	2040	20.0
	3330	1350	7.43	3720	1513	10.38	4065	1653	13.62	4390	1788	17.2	4700	1910	20.95	4920	2030	25.1	5250	2130	29.3
	4790	1460	11.65	5350	1628	16.3	5850	1788	21.30	6330	1932	27.0	6770	2065	32.9	7180	2190	39.5	7560	2305	46.0
	6940	1670	21.10	7770	1870	29.5	8525	2050	38.80	9190	2210	49.0	9840	2365	59.7	......	......	......	......	......	......

COMPARISON OF FAN SIZES.—Manufacturers of fans designate the various sizes by a series of arbitrary numbers. To enable comparison of the same size of fan

Table 13.—Dimensions, inches, of Type P Blowers and Exhausters. (Figs. 16 and 17.)
(American Blower Corp., Detroit, Mich., 1934)

	Fan Size	Wheel Diam.	A	B	C	D	E	F	G	H
Cast-iron	000	7	4 21/32	5 19/32	5 1/8	....	5 1/4	2 13/16	....	6 9/16
	00	9 1/2	6 7/32	7 17/32	6 7/8	....	7	4	....	8 13/16
	0	10 1/4	7 3/4	9 7/16	8 19/32	....	8 13/16	5 1/16	....	11
Steel Plate	3	19 1/2	12 7/16	15 1/16	13 3/4	16 3/8	15 1/2	8	1 1/4	18
	4	22	14	17	15 1/2	18 1/2	17 1/2	9	1 3/8	21
	5	24 1/2	15 9/16	18 15/16	17 1/4	20 5/8	19 15/16	10	1 1/2	24
	6	27	17 1/8	20 7/8	19	22 3/4	23 1/4	11	1 5/8	27
	7	32	20 3/16	24 5/16	22 1/4	26 3/8	22 3/8	12 1/2	1 3/4	30

	Fan Size	Wheel Diam.	I	J	K	L	M	N	P	Q
Cast-iron	000	7	2 13/16	8 3/4	2 1/4	....	4 11/16	....	2 3/4	1 1/2
	00	9 1/2	4 11/16	11 3/4	3 1/16	....	5 3/8	....	3 3/4	1 1/2
	0	10 1/4	5 7/8	13 11/16	3 7/8	....	6 13/16	....	4	2 1/4
Steel Plate	3	19 1/2	10	28 9/16	6	6 1/2	12 5/16	12 11/16	5	4 1/4
	4	22	11	31 1/2	6 13/16	7 1/4	13 7/16	14 5/16	6	4 1/4
	5	24 1/2	12	35 1/2	7 1/2	8	15 1/2	15 15/16	7	5 1/4
	6	27	13	37 9/16	8 5/16	8 3/4	16	17 15/16	8	5 1/2
	7	32	15	42 9/16	9 1/4	10 1/2	18 1/16	20 1/2	9	6 1/2

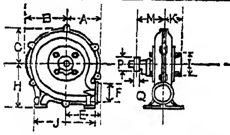


Fig. 16. Cast-iron Type P Blower and Exhauster

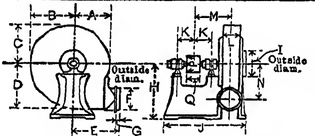


Fig. 17. Steel Plate Type P Blower and Exhauster

built by different makers, Table 14 has been prepared by the National Assoc. of Fan Manufacturers. Fans in the same line are fairly comparable as to size.

FAN CHARTS.—Fan charts perform the same function as multi-rating tables, and enable the proper size of fan for a given service to be easily selected. Two forms are illustrated below.

Fig. 18 is a form of chart used by the B. F. Sturtevant Co. to determine the capacity of steel plate, multivane, and turbovane fans under various conditions of operation, and for the purpose of enabling the proper fan to be selected for a given condition of operation. The lower portion of the chart comprises volume and pressure scales while the upper portion gives curves of total efficiency and of the ratio of total pressure divided by static pressure, together with curves showing the tip speed of the fans at various static pressures. To use the chart, locate point A to correspond with the desired volume in cubic feet per minute and the desired static pressure. A

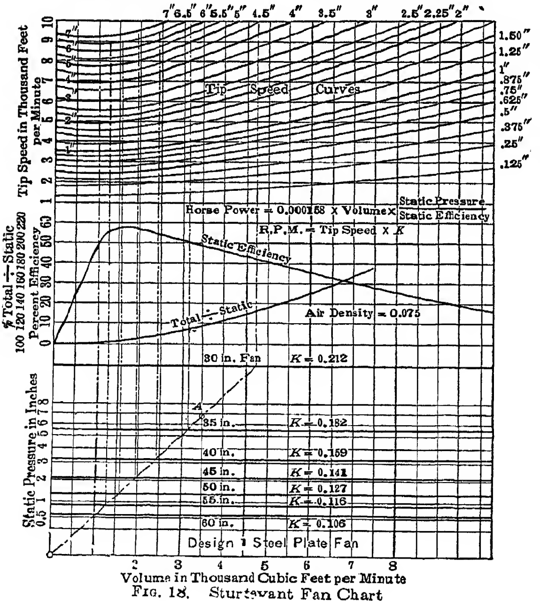


Fig. 18. Sturtevant Fan Chart

Table 14.—Comparison of Commercial Ventilating Fan Sizes. (National Assoc. of Fan Manufacturers, Chicago, 1935)

Figures in the table are the fan size numbers used by the several makers to designate their product. Fan sizes in the same line of each division of the table are fairly comparable.

Single Inlet, Single Width, Multiblade and Non-overloading Fans

Manufacturer *	1	2	3	4	4	5	6	7	8	9	10	10
Type or Design Class	HS / Sa	CL / LL[b]	HSV3 / HY	MV6	SV5	B	B	H / HB	Pc / Ad	ME	SL[e]	TR
	2	2	7/8 1	2	......	2	25	2	2 1/2	13 1/2	2	2 2 1/4
	2 1/2	2 1/2	1 1/8 1 1/4	3	57 1/2 60	2 1/2	30	2 1/2	3	17	2 1/2	2 1/2
	3	3	1 1/2	4	62 1/2 65	3	35	3	3 1/2	20 1/2	3	3
	3 1/2	3 1/2	1 3/4	5	67 1/2	4	40	3 1/2	4 4 1/2	......	3 1/2	3 1/2
	4 4 1/2 5 5 1/2 6 6 1/2 7	4 4 1/2 5 5 1/2 6 6 1/2 7	2 2 1/4 2 1/2 2 3/4 3 3 1/4 3 1/2	6 6 1/2 7 7 1/2 8 8 1/2 9	70 75 80 85 90 95 100	5 6 7 8 9	45 50 55 60 70 80	4 4 1/2 5 5 1/2 6 7	5 5 1/2 6 6 1/2 7 8 9	24 27 1/2 31 38 45	4 4 1/2 5 5 1/2 6 6 1/2 7 7 1/2	4 4 1/2 5 5 1/2 6 6 1/2 7 7 1/2
	8	8	4 4 1/4	10	105	10	90	8	10	52	8 8 1/2	8
	9	9	4 1/2	11	115	11	..	9	11	59	9	8 1/2 9
	10	10	5	12	125	12	..	10	12	66	10	10
	11	11	5 1/2	13	135	13	..	11	13	73	11	11
	12	12	6	14	145	14	..	12	14 15	80	12	12
	13 1/2	13	6 1/2	15	155	15	..	......	16	87	13	......
	15	14 15	7 7 1/2	16 1/2	165	16 17	..	......	17 18 19	94	14 15	13 14
	16 1/2 18	16 18	8 8 1/2	18† 20†	175	18	..	......	20 21 22	101 108	......	15 16 17

Double Inlet, Double Width, Multiblade Fans

Manufacturer *	1	2	3	4	5	6	7	8	9	10
Type or Design Class	Sa	CLD	HV	MV6	B	B	H	Pc	ME	TR
	2	2	7/8 1	2	2	..	2	2 1/2	13 1/2	2 2 1/4
	2 1/2	2 1/2	1 1/8 1 1/4	3	2 1/2	..	2 1/2	3	17	2 1/2
	3	3	1 1/2	4	3	..	3	3 1/2	20 1/2	3
	3 1/2	3 1/2	1 3/4	5	4	..	3 1/2	4 4 1/2	......	3 1/2
	4 4 1/2 5 5 1/2 6 6 1/2 7	4 4 1/2 5 5 1/2 6 6 1/2 7	2 2 1/4 2 1/2 2 3/4 3 3 1/4 3 1/2	6 6 1/2 7 7 1/2 8 8 1/2 9	5 6 7 8 9	50 55 60 70 80	4 4 1/2 5 5 1/2 6 7	5 5 1/2 6 6 1/2 7 8 9	24 27 1/2 31 38 45	4 4 1/2 5 5 1/2 6 6 1/2 7 7 1/2
	8	8	4	10	10	90	8	10	52	8
	9	9	4 1/4 4 1/2	11	11	..	9	11	59	8 1/2 9
	10	10	5	12	12	..	10	12	66	10
	11	11	5 1/2	13	13	..	11	13	73	11
	12	12	6	14	14	..	12	14 15	80	12
	13 1/2	13	6 1/2 7	15	15 16	..	......	16	87	13
	15	14 15	7 1/2	16 1/2	17	..	......	17 18 19	94	14
	16 1/2 18	16 18	8 8 1/2	18† 20†	18	..	......	20 21 22	101 108	15 16 17

Double Inlet, Double Width, Non-overloading Fans

Manufacturer *	1	2	3	4	7	8	10
Type or Design Class	HS	CLD	HSV3	SV5	HB	A[d]	SL[e]
	2	2	1	......	2	2 1/2	2
	2 1/2	2 1/2	1 1/4	57 1/2 60	2 1/2	3	2 1/2
	3	3	1 1/2	62 1/2 65	3	3 1/2	3
	3 1/2	3 1/2	1 3/4	67 1/2	3 1/2	4 4 1/2	3 1/2
	4 4 1/2 5 5 1/2 6 6 1/2 7	4 4 1/2 5 5 1/2 6 6 1/2 7	2 2 1/4 2 1/2 2 3/4 3 3 1/4 3 1/2	70 75 80 85 90 95 100	4 4 1/2 5 5 1/2 6 7	5 5 1/2 6 6 1/2 7 8 9	4 4 1/2 5 5 1/2 6 6 1/2 7
	8	8	4	105	8	10	7 1/2 8
	9	9	4 1/4 4 1/2	115	9	11	8 1/2 9
	10	10	5	125	10	12	10
	11	11	5 1/2	135	11	13	11
	12	12	6	145	12	14 15	12
	13 1/2	13	6 1/2 7	155	......	......	13
	15	14 15	7 1/2	165 175	......	......	14 15
	16 1/2 18	16 18	8 8 1/2	......	......	......	......

* Manufacturers are as follows: 1. American Blower Corp., Detroit. 2. Buffalo Forge Co., Buffalo, N. Y. 3. Clarage Fan Co., Kalamazoo, Mich. 4. B. F. Sturtevant Co., Hyde Park, Mass. 5. Garden City Fan Co., Chicago. 6. Ilg Elect. Ventilating Co., Chicago. 7. Autovent Fan & Blower Co., Chicago. 8. Bayley Blower Co., Milwaukee. 9. New York Blower Co., Chicago. 10. Western Blower Co., Seattle, Wash.

† Class MV3. [a] Sirocco; [b] Limit-load; [c] Plexiform; [d] Aeroplex; [e] Streamline.

line is drawn through this point and the origin as shown. The ordinate erected at the intersection of this line with the lines corresponding to the different sizes of fan will cut the static efficiency curve, the total divided by static pressure curve and the tip speed curve. The intersection of the ordinate with these curves will give efficiency and tip speed corresponding to volume and pressure assumed. Speed and horsepower are determined then by: R.p.m. = tip speed × K.

Horsepower = (0.000158 × volume × static pressure) ÷ static efficiency. Values of K are given on the lines corresponding to each size of fan.

Example.—Required the size of fan to deliver 3500 cu. ft. per min. against 6 ¾ in. static pressure. The point corresponding to this volume and pressure is shown at A, Fig. 18. Draw a line through A and the origin of co-ordinates, cutting the several fan size lines. Erect ordinates at the points of intersection and continue them to the 6.75 tip-speed curve, intersecting the static efficiency curve and the curve of total divided by static at the middle of the chart. The following data then may be read from the curves:

Fan No.	Volume	Pressure	Efficiency	Tip Speed
35	3500		50.5	Over 10,000
40	3500		54.5	9500
45	3500	6 3/4	56.5	9150
50	3500	6 3/4	57	9000
55	3500		55.5	8900
60	3500	6 3/4	47.5	8900

The tip speed of the 35-in. fan is too high and the 60-, 55-, and 50-in. fans have no apparent advantage in efficiency over the 45-in. fan. The choice, therefore, lies between the 40- and 45-in. fans. Since the difference in efficiency is small, the 40-in. fan is the natural selection. although for induced draft work the selection of the 45-in. fan probably will give better results.

Fig. 19 is a form of chart used by the American Blower Corp. It is a chart for double-inlet, 2/3 double-width fans, based on 690 r.p.m. Similar charts are used for other speeds. This chart will give the size of fan, brake-horsepower, mechanical efficiency and other data required for handling air or gas at temperatures from 70° to 1000° F. The chart is entered on the ordinate of the given static pressure. At the intersection of this ordinate with the line of given temperature on the left side of the chart a horizontal line is drawn to intersect the ordinate of the required volume. The curve of fan sizes nearest the intersection gives the size of fan. At the point where the volume ordinate intersects the brake-horsepower curve of the fan size selected, a horizontal is projected to intersect the given temperature line at the right of the chart. The intersection is on the ordinate of the required brake horsepower.

Example.—Required the size of fan, brake-horsepower and mechanical efficiency to move

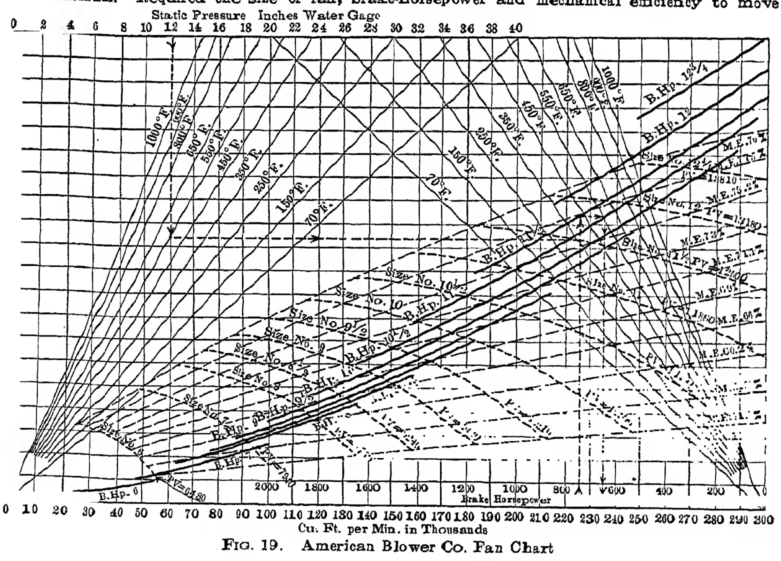


Fig. 19. American Blower Co. Fan Chart

225,000 cu. ft. of air at 550° F. and 12 in. water gage static pressure from entrance of inlet boxes to discharge of fan.

Solution.—Draw a line downward from 12 in. static pressure to intersect the 550° line. Draw the horizontal from the intersection to meet the ordinate 225,000 cu. ft. per min., and read size 11 1/2 fan at their intersection. Extend the ordinate vertically to the 11 1/2 B.Hp. curve, and draw a horizontal from the intersection to the 550° F. line. Follow vertically downward to read 653 B.Hp. The intersection of fan size curve and volume ordinate lies between the 75.2% and 77% mechanical efficiency curves, and may be read as 76% mechanical efficiency. Additional data shown are a cu. ft. per min. of 48.5% of the rated capacity of the fan with standard air (68° F.) and a value of pressure × velocity of 12,500.

2. DISC AND PROPELLER FANS

Mr. H. F. Hagen says that in disc or propeller type fans the air flows in lines substantially parallel to the axis of fan rotation. Such fans are built with a great variety of blade shapes, the different forms modifying the characteristics of performance.

This type of fan has the same change in performance with a change in speed as the centrifugal fan, against the same equivalent orifice. Volume increases directly as the speed, pressure as the square of the speed and horsepower as the cube of the speed; efficiency, ignoring the change in proportion of power consumed by bearing friction, remains constant. These relations have been proved so often by different investigators as to leave no doubt of their accuracy. Theoretically, they also hold for a constant kinematic viscosity. *Kinematic viscosity* is the relation between the viscosity of the air and the speed with which it will propagate a compression, that is, the speed of sound. Within the small range of pressure and temperature differences of air handled by fans, there is an entirely negligible change in kinematic viscosity. The square-cube law, therefore, holds theoretically and experimentally.

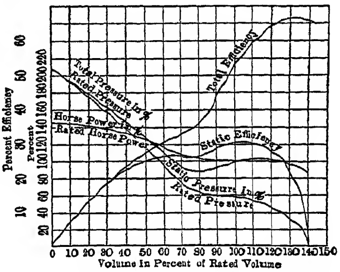


Fig. 20. Characteristics of Axial Flow Fan

The axial flow fan has peculiar characteristics. See Fig. 20. There is always a break in the pressure curves. The horsepower tends to increase from the lowest value at maximum opening and greatest volume to the highest value when the opening is blocked tight, that is, at zero volume. By various designs, the curves can be greatly modified. The reverse curvature in the pressure characteristics can be almost eliminated but the pressure

Table 15.—Capacity of Disc Ventilating Fans.—Free Delivery, Exhausting from Chamber

(American Blower Corp., Detroit, Mich., 1934)

Fan Diam., in.	Outlet Velocity, ft. per min.								
	300			600			900		
	C.f.m.	R.p.m.	B.Hp.	C.f.m.	R.p.m.	B.Hp.	C.f.m.	R.p.m.	B.Hp.
18	566	296	0.003	1132	592	0.025	1698	888	0.083
24	1006	222	.01	2012	444	.04	3019	666	.15
30	1572	178	.01	3145	356	.07	4717	534	.23
36	2264	148	.01	4528	296	.10	6792	444	.33
42	3083	127	.02	6166	254	.13	9249	381	.45
48	4025	111	.02	8050	222	.17	12076	333	.59
54	5094	99	.03	10188	198	.22	15282	298	.74
60	6289	89	.03	12578	178	.27	18868	267	.92
72	9056	74	.05	18112	148	.39	27168	222	1.32

Fan Diam., in.	Outlet Velocity, ft. per min.								
	1200			1500			1800		
	C.f.m.	R.p.m.	B.Hp.	C.f.m.	R.p.m.	B.Hp.	C.f.m.	R.p.m.	B.Hp.
18	2264	1184	0.197	2840	1480	0.38	3396	1776	0.663
24	4025	888	.35	5031	1110	.68	6038	1332	1.18
30	6289	712	.54	7861	890	1.06	9434	1068	1.84
36	9056	592	.79	11320	740	1.53	13584	888	2.65
42	12332	508	1.07	15415	635	2.08	18498	762	3.60
48	16101	444	1.39	20125	555	2.72	24151	666	4.70
54	20376	396	1.76	25470	496	3.44	30564	594	5.95
60	25157	356	2.17	31444	445	4.25	37735	534	7.35
72	36224	296	3.13	45280	370	6.12	54336	444	10.60

at this point then becomes unstable. By limiting leakage around the periphery, horsepower can be made nearly constant, but the general features of the curves are always present in greater or less degree.

The pressure curves are really a combination of two different flows of air. When the fan is operating against low resistance and delivering the greater volumes, the air can flow substantially straight through the fan. When the pressure increases to such an amount that at the speed of the portion of the blade near the hub sufficient pressure cannot be developed, all the air can no longer go straight through, but some of it must climb up the blade until it reaches a point where the speed is sufficient to create a pressure equal to that against which the fan is working. The fan then is to some extent working as a centrifugal fan. As the pressure increases, this action becomes more marked. The break in the pressure curves occurs at the point where the action is changing from that of a propeller to that of a centrifugal fan.

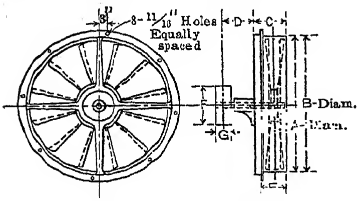


FIG. 21. Disc Ventilating Fan

Table 16.—Capacity of Disc Ventilating Fans.—Exhausting from Chamber Having Restricted Inlet

(American Blower Corp., Detroit, Mich., 1934)

Fan Diam., in.	Resistance Pressure, in. of Water								
	1/8 in.			1/4 in.			3/8 in.		
	C.f.m.	R.p.m.	B.Hp.	C.f.m.	R.p.m.	B.Hp.	C.f.m.	R.p.m.	B.Hp.
18	1181	872	0.11	1671	1236	0.31	2048	1516	0.56
24	2099	656	.19	2970	927	.54	3642	1136	1.00
30	3280	524	.30	4640	744	.85	5690	908	1.56
36	4723	436	.43	6683	618	1.22	8194	758	2.25
42	6429	375	.59	9094	532	1.66	11152	650	3.06
48	8397	328	.77	11880	464	2.18	14568	568	4.00
54	10627	290	.97	15037	412	2.75	18436	506	5.05
60	13120	262	1.20	18560	372	3.40	22760	454	6.24
72	18893	218	1.72	26732	309	4.88	32774	379	9.00

Fan Diam., in.	Resistance Pressure, in. of Water								
	1/2 in.			5/8 in.			3/4 in.		
	C.f.m.	R.p.m.	B.Hp.	C.f.m.	R.p.m.	B.Hp.	C.f.m.	R.p.m.	B.Hp.
18	2361	1714	0.86	2462	1952	1.21	2894	2144	1.59
24	4198	1312	1.54	4698	1464	2.15	5146	1608	2.82
30	6560	1048	2.40	7340	1171	3.36	8040	1286	4.41
36	9446	872	3.45	10570	976	4.84	11578	1072	6.35
42	12858	750	4.70	14386	840	6.59	15758	918	8.65
48	16794	656	6.16	18792	732	8.60	20584	804	11.28
54	21253	582	7.76	23782	650	10.89	26050	714	14.29
60	26240	524	9.60	29360	585	13.44	32160	643	17.64
72	37784	436	13.80	42282	488	19.36	46312	536	25.40

Table 17.—Dimension of Disc Ventilating Fans. (Fig. 21.)

(American Blower Corp., Detroit, Mich., 1934)

Dimension A, in. .	18	24	30	36	42	48	54	60	72
" B, in. .	19 1/8	25 1/8	31 1/8	37 1/4	43 1/16	49 7/16	55 1/2	61 1/2	73 1/4
" C, in. .	5 1/2	6 1/2	7 1/2	8	9	10 3/4	11	12	14
" D, in. .	6 5/8	8	8 1/2	9	9 1/2	10 1/2	11	11 1/2	13 1/2
" E, in. .	3 7/8	4 1/2	5 1/2	6	7	8 1/2	8 3/4	9 3/4	11 5/8
" F, in. .	4	6	8	10	12	13	14	16	18
" G, in. .	2 1/4	3	4	4	4	5	5 1/2	5	6

DUCTS FOR FANS AND BLOWERS.—The charts, Fig. 22, will enable the selection of the proper size of round or rectangular duct for a given fan installation, or permit the resistance of a given duct system to be determined, from which the proper fan selection can be made. The charts are based on 100 ft. of straight galvanized duct, with proper allowance for the friction of laps and rivets. To determine the size of duct for a given volume of air at a given friction loss, determine the friction loss per 100 ft., and then find on Chart 1 the intersection of the co-ordinate of this friction loss and of the given volume. Read diameter of duct on the duct-diameter line passing through the intersection, and the velocity on the velocity line passing through the same point. If the duct

is rectangular, follow the diameter line into Chart 2. At its intersection with the curve corresponding to one dimension of the duct, draw a horizontal line and read the other dimension on the scale at the right.

EXAMPLE 1.—Find the size of a round duct 175 ft. long, and of a rectangular duct 12 in. wide, that will convey 5000 cu. ft. per min. with a friction loss of $2^3/_4$ in. water gage. *Solution.*—Friction loss per 100 ft. = (100/175) × 2.75 = 1.57 in. Draw a horizontal line through 1.57 on the friction loss scale at left of Chart 1. It intersects the 5000 cu. ft. per min. ordinate just below the 16 in. diam. line, and between the 3500 and 3600 ft. per min. velocity lines. Hence, a 16 in. round duct, with an air velocity of 3550 ft. per min. will fulfill the given conditions. Draw a line

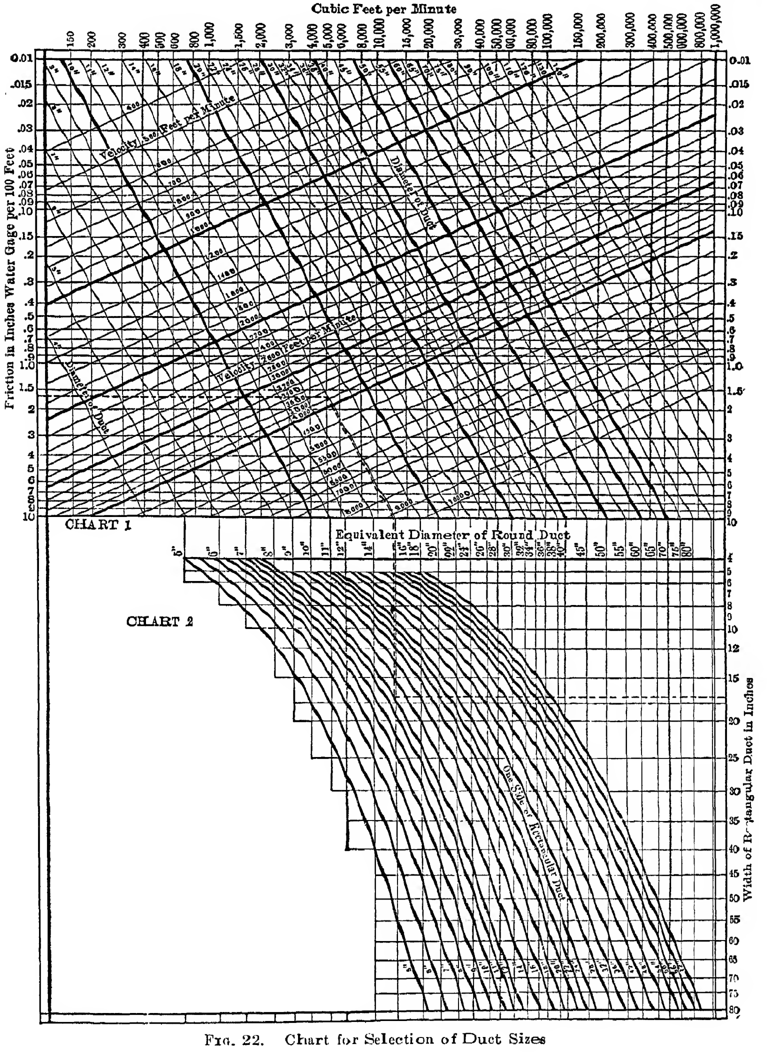


FIG. 22. Chart for Selection of Duct Sizes

parallel to the 16 in. diameter line to intersect the bottom line of Chart 1. Project a line vertically downwards from this point into Chart 2. At its intersection with the 12 in. curve, draw a horizontal line to the scale at the right and read the other dimension of the duct, 17 in.

EXAMPLE 2.—Find the resistance pressure in a rectangular duct system 250 ft. long, handling 15,000 cu. ft. of air per min., the ducts being 18 × 36 in. *Solution.*—The horizontal line through 18 in. width on Chart 2, intersects the 36 in. curve opposite the 28 in. diam. curve on Chart 1. Follow the 28 in. curve to its intersection with the 15,000 cu. ft. ordinate. Follow the horizontal line through this intersection to read a friction loss of 0.8 in. on the scale at the left. Total friction loss = (250/100) × 0.8 = 2.0 in.

Elbows should be considered as an equivalent length of straight pipe, added to the length of the duct. Friction loss should be computed on the total length thus found. Table 18 gives data for such corrections.

Table 18.—Resistance of 90-degree Elbows

D = pipe diameter, R = radius of throat of elbow, L = length of straight pipe of equivalent resistance.

R	L	R	L	R	L	R	L	R	L
$1/4\,D$	$35\,D$	D	$10\,D$	$1\,3/4\,D$	$5\,D$	$3\,D$	$4.8\,D$	$4\,1/2\,D$	$5.5\,D$
$1/2\,D$	$18\,D$	$1\,1/4\,D$	$7.5\,D$	$2\,D$	$4.3\,D$	$3\,1/2\,D$	$5\,D$	$5\,D$	$5.8\,D$
$3/4\,D$	$13\,D$	$1\,1/2\,D$	$6\,D$	$2\,1/2\,D$	$4.5\,D$	$4\,D$	$5.2\,D$	$5\,1/2\,D$	$6\,D$

3. METHODS OF TESTING FANS

ANEMOMETER METHOD.—Measurements by anemometers are liable to be very inaccurate (see page 1–21) and results obtained by them should be considered only as rough approximations.

WATER GAGE READINGS AT END OF TAPERED CONE.—This method is also far from accurate on account of variable eddies in the air column.

PITOT TUBE READINGS IN CENTER OF DISCHARGE PIPE give fairly accurate results when discharge pipe is the same size as fan outlet, when Pitot tube is placed at a distance of at least 15 diameters of pipe from fan outlet, when the tube is so made that it will give correct readings of static pressure, and when velocities computed from the readings are corrected by a coefficient (0.87 to 0.92 in different experiments) for the ratio between average velocity and velocity at the center of the tube.

PITOT TUBE READINGS IN ZONES OF EQUAL AREA.—The most accurate results are obtained if the tube is traversed across two diameters of the tube at right angles to each other, placing the nozzle successively at points which will divide the cross-sectional area into equal annular areas (with one central circular area). See Test Code, p. 1–83. Since velocity at any point is proportional to the square root of velocity head, it is necessary, for accurate results, to use the square of the average of the square roots of the readings as the mean velocity head of the whole area of the pipe. For low pressures, an inclined manometer should be used with the Pitot tube, containing gasoline instead of water; gasoline keeps the tubes clean, has a definite meniscus and almost no capillary attraction for the glass. Readings of the tube should be corrected for inclination and for specific gravity of gasoline to reduce them to equivalent in. of water.

For accurate scientific work it is well to check the static tube readings by manometer readings from a piezometer ring, which is a narrow annular air-tight channel encircling and soldered to the pipe. Six or more smooth holes bored into the pipe at right angles to its axis connect the interior of the pipe with the ring. The Pitot tube also may be calibrated by means of a Thomas electric gas meter.

THE THOMAS ELECTRIC METER for air and gas consists of an enlargement of section of the flow pipe into a chamber of a diameter equal to about two diameters of the pipe, with conical ends connecting it with the pipe. In the interior is placed an electric heater made of bare resistance wire mounted on a fiber frame and equally distributed over the section of the chamber, and also two electric resistance thermometers, one in front of and the other behind the heater. An electric current, measured by a wattmeter, is passed through the heater and the temperatures before and after the heating are measured by the thermometers. If T_1 and T_2 are the temperatures before and after the heating, H = the heat units corresponding to the watts delivered to the heater (1 watt = 3.145 B.t.u. per hour), and S = specific heat of the air, then the weight of air heated, in pounds per minute, is $W = H \div \{60\,S(T_2 - T_1)\}$.

ACCURACY OF PITOT TUBE MEASUREMENTS.—To obtain even approximately accurate results with Pitot tubes it is necessary both to have the tube properly made and to take great precautions in using it. W. C. Rowse, *Trans.* A.S.M.E. xxxv, p. 633, 1913, tested several forms of tubes, comparing their readings with those of a Thomas

electric gas meter. The best tube was one made of a $1/4$-in. outer and a $1/8$-in. inner thin brass tube, 4 or 5 in. long, soldered together at one end, which was tapered for $3/4$ in. down to the internal diameter of the inner tube, which was thus given a sharp edge. The outer tube was perforated with a smooth hole 0.02 in. diam. at each side at the middle of its length. The rear end of the small tube and each annular space between the two tubes was connected to $1/4$-in. upright tubes, from which rubber tubes led to two manometers. The inner tube received the impact pressure, and the annular space the static pressure. The difference between the two is the velocity pressure, a direct reading of which could be made by connecting the two rubber tubes to the two legs of a single manometer. The manometers were glass U-tubes about $1/2$ in. internal diameter, containing gasoline, inclined 1 vertical to 10 horizontal. The scale was graduated to read 0.01 in. of water column. The results of these tests showed that accuracy within 1% could be obtained when all readings were obtained with a sufficient degree of refinement and when the Pitot tube was preceded by a length of pipe 20 to 38 pipe diameters long. Tests of Pitot tubes with long narrow slots in the outer tube, instead of the small holes, gave results which were in error from 3.5 to 10%.

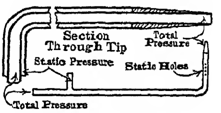


FIG. 23. Pitot Tube

4. TEST CODES FOR FANS

The following test codes for fans and blowers is abstracted from the standard test code of the Natl. Assoc. of Fan Mfrs. and the A.S.H.V.E., second edition, published 1932.

Centrifugal Fans

TEST DUCT is a discharge duct of area equal to fan discharge, straight, of uniform area and section, and not less than 10 diameters long. Readings are taken over a plane at right angles to the duct axis with a Pitot tube parallel to the duct axis, the impact end pointing upstream. Readings are taken at $3/4$ of duct length from fan end. An adjustable device for symmetrically throttling the air is at the outer end of the test duct. See Fig. 24.

STANDARD PRESSURE MEASURING INSTRUMENTS are the double Pitot tube and the manometer. The Pitot tube has not less than 4 static orifices not over 0.02 in. diam. No orifice is located at less than 8 tube diameters from upstream end of tube and at an equal distance from the elbow. The manometers or gages for measuring static pressure and velocity pressure are filled with a light liquid, kerosene, gasoline, or alcohol, and calibrated in position, with rubber tubing attached, by comparison with a calibrated water-filled hook gage whose screw is calibrated to a degree of accuracy commensurate with the accuracy of the proposed reading. The rubber tubing is not removed from the manometer after calibration. The manometer for measuring total pressure is vertical and water-filled.

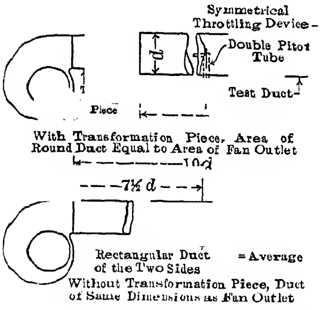


FIG. 24. Arrangement for Centrifugal Fan Test

DYNAMOMETER.—The cradle dynamometer is recommended for all power measurements. Calibrated direct-connected motors may be used. With indirect drive, friction losses are determined for each condition of the test.

NUMBER OF READINGS.—**Round Ducts.**—Not less than 2 traverses of 10 readings each, along diameters at right angles to each other at the centers of 5 equal-area concentric zones across the section of the duct, are made in each test. In small ducts the calibrated nozzle method of Sanford A. Moss may be used for determining volumes. See *Trans.* A.S.M.E. xxxviii, p. 761, 1916. Several nozzles with different size outlets are required to give range of capacity, but if the outlet duct is 20 diameters long with a throttling device midway, the number of nozzles may be reduced.

Rectangular Ducts.—Readings are taken in the center of not less than 16 nor more than 64 equal areas over the cross-section of the duct. With less than 64 readings the centers of the rectangular areas should be not more than 6 in. apart.

ANGLE OF TRANSFORMATION PIECE.—No angle between axis of test duct and any longitudinal element shall exceed 7 deg.

INLET OR OUTLET OBSTRUCTIONS.—Inlets and outlets must be unobstructed except for structural features of fan involved in bearings and their support. A fan provided with bearings is tested on its own shaft and in its own bearings, and run for several hours to wear in. Fans with bearings or bearing supports in the inlets are so mounted for test.

NUMBER OF DETERMINATIONS.—Eight or more determinations from approximately free delivery to no delivery, with approximately equal increments of capacity for discharge area.

OBSERVATIONS AND CALCULATIONS.—Simultaneous readings of total pressure, static pressure, velocity pressure, horsepower input and speed are made. Barometric pressure and wet and dry bulb temperatures in the room and in the discharge duct as close as practicable to point of traverse also are made. Speed is maintained as nearly constant as possible throughout test. Tachometers are always checked by revolution counter readings for periods of not less than 1 min. All air readings are corrected to standard air conditions at fan inlet, *i.e.*, air weighing 0.07488 lb. per cu. ft. These corrections are based on: At constant speed, pressure and horsepower input vary directly as weight of air; velocities corresponding to velocity pressure readings are averaged to obtain average velocity for each traverse, and average velocity pressures are calculated from

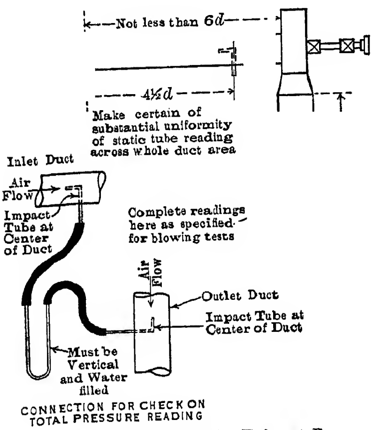


Fig. 25. Arrangement for Exhaust Fan Test

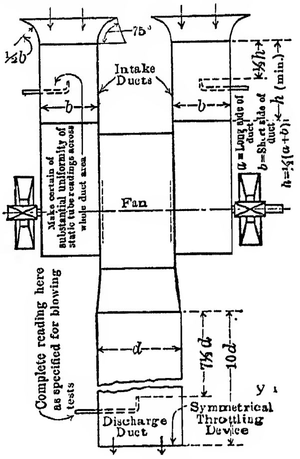


Fig. 26. Arrangement for Exhaust Fan Test with Inlet Boxes

average velocities thus determined. The formula for velocity is $V = 1096.2\sqrt{P_v/w}$, where V = velocity, ft. per min.; P_v = velocity pressure, in. of water; w = weight of air in duct, lb. per cu. ft., as taken from tables of the U. S. Weather Bureau. Capacity for each determination is product of average velocity and duct area, corrected for inlet conditions. Capacity, static pressure, total pressure and horsepower input are corrected to constant speed (see p. 1–58). Total pressure and static pressure also are corrected for duct friction to give conditions at fan discharge, by adding lost head due to duct friction between fan outlet and Pitot tube. This loss will be: For *round ducts*, $f = 0.0257(L/D)P_v$; for *rectangular ducts*, $f = 0.01285\,L\,\{(a + b)/ab\}P_v$, where f = pressure loss, in. of water; P_v = average velocity pressure, in. of water; L = distance from fan outlet to Pitot tube, ft.; D = diam. of discharge duct; a = long side of duct, ft.; b = short side of duct, ft.

RESULTS.—After performance has been calculated for each determination, results, as corrected for standard air conditions and constant speed, are plotted with capacity as abscissas and pressures and horsepowers as ordinates. Smooth curves drawn through these points give pressure and horsepower characteristics, from which data mechanical efficiency is plotted against capacity.

EXHAUSTING FANS.—Capacity is determined by readings in the discharge duct as above prescribed. To determine total pressure an inlet duct is used of area equal to fan inlet and 6 diameters long. See Fig. 25. Total head or total pressure is the difference between average absolute total pressure in discharge duct and average absolute

total pressure in inlet duct, with additions to cover friction of inlet and outlet ducts between points of measurement and the fan, in accordance with the methods previously prescribed. Average absolute total pressure in inlet duct is determined by adding to absolute total pressure in inlet duct the calculated average velocity pressure in inlet duct. Readings in inlet duct are made at 3/4 of inlet duct length from entering end. Static pressure developed by fan is the difference between total head so obtained and the velocity head in the duct.

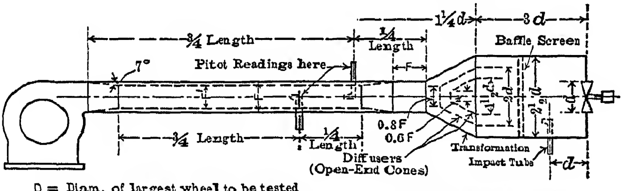
CENTRIFUGAL FANS WITH INLET BOXES are tested with inlet boxes in place. See Fig. 26. Connected to the inlet box is a duct, or ducts for double inlet fans, of the same size and form as the inlet box opening, of length not less than the mean of its two sides, with a nozzle inlet on the open end formed of cylindrical sides as in Fig. 26. Readings are made in test duct at a point 1/2 of a mean side from open end, exclusive of length of nozzle.

TOLERANCES.—A variation of 2 1/2% in determinations of capacity pressure or horsepower will not be considered excessive when applied to the mechanical efficiency.

Disc and Propeller Fans

The code assumes disc and propeller fans to exhaust from a chamber, and all air readings must be taken on the inlet side.

TEST SET-UP for disc and propeller fans consists of an air-tight chamber of rectangular or drum form and of minimum size in proportion to the fan to be tested. See Fig. 27. A duct 10 diameters long, including transformation piece, is on entrance side of chamber. Transformation piece is carried to full dimensions of the end of the chamber and provided with an effective diffuser. Volumes are varied by restriction at inlet end of test duct. Free delivery conditions are procured by an auxiliary blower for obtaining



D = Diam. of largest wheel to be tested
F = Diam. of duct to give velocity of 1800 ft. per min. when testing largest wheel
F' = Diam. of duct coned down to give velocity of 1800 ft. per min. when testing wheels of diameters less than D

Fig. 27. Arrangement for Disc and Propeller Fan Test

zero or above static pressures in the chamber. Capacity of fan is determined by readings in the test duct in the manner prescribed for centrifugal fan tests. In the large chamber, pressure is determined by readings of an impact tube.

DISC FANS TESTED BLOWING are to be tested with the same standard apparatus as shown in Fig. 27 with the addition of a collar on the discharge side of the fan, of the same diameter as fan opening and one diameter long.

MOUNTING.—Any obstruction to flow of air, as a driving motor, pulley, bearing, etc., that forms part of the unit, whether normally on either inlet or discharge side of fan, shall be in its designed position to so affect the test results as to show the true performance of fan as used in that unit.

5. POSITIVE ROTARY BLOWERS

ROTARY DISPLACEMENT BLOWERS ("Roots" type) of the single-stage lobed impeller type are used to move air and gases in all volumes against pressures up to 15 lb. per sq. in., or produce vacuums equivalent to 25 or 26 in. Hg at sea level. The compound type is efficient at pressures up to 25 or 30 lb. per sq. in. They also are used to handle gases and liquids simultaneously in varying proportions, particularly in vacuum processes. The essential parts are two impellers rotated by any suitable prime mover applied to either one or both shafts. Impeller shafts are geared together and impellers revolve in opposite directions. Each impeller traps during one revolution two pockets of the fluid being moved and forces it into the outlet pipe. The impellers can be made with any number of lobes, which are formed by gear tooth curves. A running clearance between the rotating pistons and around the casing prevents internal contact between working parts.

Volumetric efficiency depends on internal clearances, operating pressure and speed

Table 19.—Brake Horsepower Required for Rotary Displacement Blowers
(Connersville Blower Co., Connersville, Ind., 1933)

Cu. ft. per min. Free Air	Total Differential Pressure, lb. per sq. in									
	1	2	4	6	8	10	12	15	20	30
	Brake Hp., Single-stage						Brake Hp., Two-stage			
100	0.75	1.5	3.0	4.7	......	......	......	......	......	......
200	1.4	2.7	5.4	8.5	......	......	......	......	......	......
400	2.3	4.7	9.7	14.6	19.8	24	23	30	......	......
600	3.4	7.2	13.6	21.1	28.7	36	34	41	54	......
800	4.2	8.8	18.4	28.4	36.5	48	56	55	70	......
1,000	5.4	11.3	22.3	34.	45.5	58	69	69	88	......
1,500	7.7	15.9	33.	50.	65.	86	103	100	128	......
2,000	10.4	22.	44.	67.	90.	115	135	132	170	......
3,000	15.6	32.	66.	98.	130.	160	190	190	255	......
5,000	26.	54.	106.	158.	210.	250	295	300	410	570
10,000	50.	104.	207.	300.	385.	470	572	700	800	1110
15,000	74.	152.	303.	432.	575.	710	846	1065	1160	1650
20,000	99.	200.	397.	580.	740.	925	1100	1380	1530	2260
30,000	145.	295.	580.	870.	1120.	1380	1660	2070	2280	3380

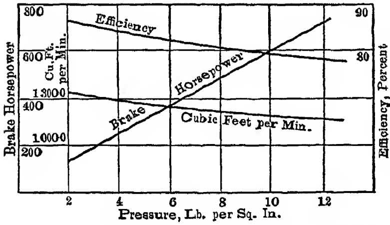
NOTE.—For all volumes and pressures of 10 lb. per sq. in. and higher, B.Hp. can be considerably reduced by compounding.

Table 20.—Sizes and Capacities of Pressure Blowers
(Connersville Blower Co., Connersville, Ind., 1933)

Size	Displacement, cu. ft. per rev.	Rev. per min.	Net Delivery of Free Air, cu. ft. per min.	Pressure, lb. per sq. in.	Brake Hp.	Pipe Connection, in.
47 AF	0.16	850	108	1.0	0.8	2 1/2
717 AF	1.17	570	570	1.0	3.3	5
10 × 24 SD	3.30	400	1,180	1.0	6.3	8
12 × 30 SD	6.00	375	2,040	1.0	10.7	10
14 × 36 SD	10.00	350	3,220	1.0	16.6	12
18 × 54 SD	24.0	230	5,000	1.0	26.0	16
22 × 66 SD	45.0	235	10,000	1.0	50.0	20
47 AF	0.16	850	96	2.0	1.5	2 1/2
717 AF	1.17	570	540	2.0	6.7	5
10 × 24 SD	3.3	400	1,160	2.0	12.5	8
12 × 30 SD	6.0	375	1,965	2.0	21.4	10
14 × 36 SD	10.0	350	3,100	2.0	33.0	12
18 × 54 SD	24.0	238	5,000	2.0	54.0	16
24 × 72 SD	57.0	192	10,000	2.0	104.0	20
55 AF	0.167	850	107	3.0	2.3	2 1/2
717 AFS	1.17	570	517	3.0	9.7	5
10 × 24 HD	3.3	440	1,214	3.0	22.0	8
12 × 30 HD	6.0	410	2,106	3.0	35.0	10
14 × 36 HD	10.0	385	3,360	3.0	55.0	12
16 × 42 HD	14.9	378	5,000	3.0	80.0	16
22 × 54 HD	36.8	298	10,000	3.0	155.0	20
55 AFS	0.167	850	96	5.0	3.9	2 1/2
8 × 12 HD	.9	660	469	5.0	14.3	6
10 × 15 HD	2.1	550	958	5.0	28.0	8
14 × 21 HD	5.83	440	2,198	5.0	62.0	10
16 × 24 HD	8.5	407	3,000	5.0	82.0	12
20 × 30 HD	16.5	340	5,000	5.0	132.0	16
24 × 36 HDH	28.5	380	10,000	5.0	250.0	20
65 AFS	0.222	750	118	7.5	6.4	3
10 × 10 HD	1.4	550	609	7.5	28.0	6
12 × 12 HD	2.4	495	965	7.5	43.0	8
16 × 16 HDH	5.7	418	2,000	7.5	82.0	10
18 × 18 HDH	8.0	432	3,000	7.5	119.0	12
24 × 24 HDH	19.0	300	5,000	7.5	192.0	16
30 × 30 HDH	35.7	300	10,000	7.5	360.0	20
12 × 6 HD	1.2	520	500	10.0	30.	6
14 × 10 HD	2.77	448	1,000	10.0	58.	8
18 × 14 HDH	6.2	382	2,000	10.0	109.	10
20 × 16 HDH	8.8	394	3,000	10.0	160.	12
26 × 18 HDH	16.3	340	5,000	10.0	250.	16
32 × 24 HDH	33.7	319	10,000	10.0	470.	20

In large, medium-pressure units operating at direct-connected motor speed it may be as high as 98%. When operating at reasonably high speeds there is considerable effect of adiabatic compression; hence, good efficiencies are possible in single-stage operation at pressures up to 12 or 15 lb. per sq. in.; in 2-stage operation up to 25 to 30 lb. per sq. in. Table 19 gives horsepower required for different volumes and pressures. Table 20 gives commercial sizes.

Fig. 28 gives characteristics of a blower operating at constant speed, working against pressures ranging from 2 to 12 lb. per sq. in. Fig. 29 shows characteristics of a blower operated at constant speed and equipped with a power-saving unloading valve to reduce horsepower input to the blower in proportion both to reduction in air pressure and reduction in volume. Fig. 30 gives horsepower and volume curves of a blower driven by a variable speed engine or motor.



28. Characteristics of Blower at Constant Speed and Volume

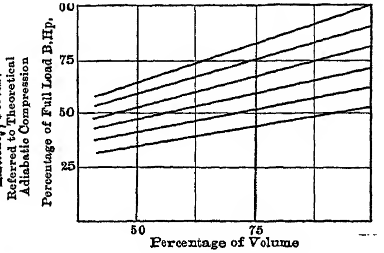
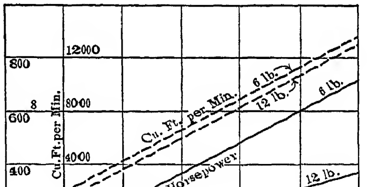


FIG. 29. Characteristics of Blower at Variable Speed

Roots type blowers are used for transmission of manufactured gas, pneumatic conveying systems, agitation and aeration of liquids as in the activated sludge process of sewage disposal, vacuum filtration, foundry cupola and smelting furnace blowing, ventilation of new tunnel workings, scavenging and supercharging Diesel engines, etc. In New York and Chicago they are used on pneumatic tube systems for conveying mail underground from sub-stations to the central post-office. In foundry applications they have been arranged to deliver a constant weight of air, either by manual control or automatically.

Modifications of impeller design permit these machines to be used with high efficiency for pumping heavy, viscous fluids which, due to absence of valve ports, can be handled effectively.



G. 30. Characteristics of Blower Equipped with Unloader, at Constant Speed

SIZE OF BLOWER REQUIRED.—Table 20 gives data on the capacities of blowers, from which the proper size may be selected when the volume of air required is known. An ordinary smith fire requires about 60 cu. ft. of air per min. An ordinary oil furnace burns about 2 gal. of oil per hr., and 1800 cu. ft. of air should be provided per gallon of oil. For quantity of air required by foundry cupolas, see Foundry Practice.

AN EXHAUSTER AND BLOWER is made by Schutte & Koerting Co., Philadelphia, on the principle of the steam-jet ejector. Capacities are given in Table 21. When used

Table 21.—Dimensions and Capacities of Steam-jet Blower and Exhauster

Diameter of Pipes, inches		Capacity per Hour,	Diameter of Pipes, inches		Capacity per Hour,	Diameter of Pipes, inches		Capacity per Hour,
Air	Steam	cu. ft.	Air	Steam	cu. ft.	Air	Steam	cu. ft.
1/2	1/4	300	2	3/4	4,000	5	2	27,000
3/4	3/8	600	2 1/2	1	6,000	6	2	35,000
1	3/8	1000	3	1 1/4	12,000	7	2 1/2	48,000
1 1/2	1/2	2000	4	1 1/2	18,000	8	3	60,000

as exhausters with a steam pressure of 45 lb., these machines will produce a vacuum of 24 in. of mercury, but they can be specially constructed to produce a 28-in. vacuum. When used as compressors, they will operate against a counter pressure equal to 1/7 of the steam pressure. Table 22 shows the time required to evacuate tanks and gives the performance of a 1 1/2-in. exhauster.

Another steam-jet blower is used for boiler firing, ventilation and similar purposes where a low counter pressure or rarefaction meets the requirements. The volumes as given in Table 23 are on the supposition of 60 lb. steam pressure, and a counter pressure of, say, from 0.5 to 2 in. of water.

Table 22.—Performance of 1 1/2-in. Steam-jet Exhauster

(Schutte & Koerting Co., Philadelphia, 1933)

Vacuum in tank, in.	Time, min., to evacuate 100 cu. ft.					Capacity, lb. per hr., of free dry air, 70° F.				
	Steam Pressure, lb. per sq. in.									
	100	125	150	175	200	100	125	150	175	200
5	0.63	0.60	0.59	0.58	0.56	2.85	3.0	3.06	3.12	3.21
10	1.36	1.30	1.27	1.25	1.21	6.16	6.5	6.63	6.76	6.96
15	2.41	2.60	2.54	2.50	2.42	12.35	13.0	13.26	13.52	13.91
20	3.78	3.70	3.63	3.55	3.44	16.63	17.5	17.85	18.20	18.73
24	5.67	5.40	5.29	5.18	5.02	25.65	27.0	27.54	28.08	28.89
27.8	12.60	12.00	11.76	11.52	11.16	57.0	60.0	60.2	62.4	64.2

Table 23.—Dimensions and Capacities of Steam-jet Blower for Boiler Furnaces and Ventilation

Diam., in.			Capacity	Diam., in.			Capacity	Diam., in.			Capacity
Air Inlet	Air Disch.	Steam Pipe	per hr., cu. ft.	Air Inlet	Air Disch.	Steam Pipe	per hr., cu. ft.	Air Inlet	Air Disch.	Steam Pipe	per hr., cu. ft.
4	3	3/8	10,000	11	7	3/4	60,000	18	14	1 1/4	240,000
5	4	1/2	20,000	12	8	3/4	90,000	24	18	1 1/2	500,000
8	5	1/2	30,000	14	10	1	120,000	32	24	2	1,000,000
9	6	3/4	45,000	16	12	1	180,000	42	32	2 1/2	2,000,000

Maximum coal burning capacity per hour = cu. ft. of air per hr. ÷ 200.

Section 2

WATER

PROPERTIES OF WATER

HYDRAULICS
By William P. Creager

HYDRAULIC TURBINES
By R. E. V. Sharp

PUMPS AND PUMPING ENGINES
By Robert Thurston Kent

CENTRIFUGAL PUMPS
By V. deP. Gerbereux

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WATER

PROPERTIES OF WATER

EXPANSION OF WATER.—Table 1 gives the relative volumes of water at different temperatures, compared with its volume at 4° C. according to Kopp, as corrected by Porter.

Table 1.—Relative Volume of Water at Different Temperatures

Cent.	Fahr.	Volume	Cent.	Fahr.	Volume	Cent.	Fahr.	Volume
4°	39.1°	1.00000	35°	95°	1.00586	70°	158°	1.02241
5	41	1.00001	40	104	1.00767	75	167	1.02548
10	50	1.00025	45	113	1.00967	80	176	1.02872
15	59	1.00083	50	122	1.01186	85	185	1.03213
20	68	1.00171	55	131	1.01423	90	194	1.03570
25	77	1.00286	60	140	1.01678	95	203	1.03943
30	86	1.00425	65	149	1.01951	100	212	1.04332

Weight of 1 cu. ft. at 39.1° F. = 62.4245 lb.; weight of 1 cu. ft. at 212° F. = 62.4245 ÷ 1.04332 = 59.833 lb.

WEIGHT OF WATER AT DIFFERENT TEMPERATURES.—The weight of water at maximum density, 39.1°, is generally taken at the figure given by Rankine, 62.425 lb. per cubic foot. Some authorities give as low as 62.379. The figure 62.5 commonly given is approximate. The highest authoritative figure is 62.428. At 62° F. the figures range from 62.291 to 62.360. At 32° F. figures given by different writers range from 62.379 to 62.418. Hamilton Smith, Jr. (from Rosetti) gives 62.416. The weights of water per cubic foot from 32° F. to and including 212° F., given in Table 2, are based on the International Critical Tables. The weights over 212° are based on Keenan's Steam Tables. Heat units above 32° F. are from Keenan's Steam Tables.

PRESSURE OF WATER DUE TO ITS WEIGHT.—The pressure of still water in pounds per square inch against the sides of any pipe, channel, or vessel of any shape whatever is due solely to the "head," or height of the level surface of the water above the point at which the pressure is considered, and is equal to 0.43302 lb. per square inch for every foot of head, or 62.355 lb. per square foot for every foot of head (at 62° F.). The pressure per square inch is equal in all directions, downwards, upwards, or sideways, and is independent of the shape or size of the containing vessel. See Table 3.

The pressure against a vertical surface, as a retaining-wall, at any point is in direct ratio to the head above that point, increasing from 0 at the level surface to a maximum at the bottom. The total pressure against a vertical strip of unit breadth increases as the area of a right-angled triangle whose perpendicular represents the height of the strip and whose base represents the pressure on a unit of surface at the bottom; that is, it increases as the square of the depth. The sum of all the horizontal pressures is represented by the area of the triangle, and the resultant of this sum is equal to this sum exerted at a point one-third of the height from the bottom. The horizontal pressure is the same if the surface is inclined instead of vertical.

For an elaboration of these principles see Merriman's Civil Engineers' Handbook or the chapter on Hydrostatics in any work on Physics. For dams, retaining-walls, etc., see Merriman.

The amount of pressure on the interior walls of a pipe has no appreciable effect upon the amount of flow.

BUOYANCY.—When a body is immersed in a liquid, whether it float or sink, it is buoyed up by a force equal to the weight of the bulk of the liquid displaced by the body. The weight of a floating body is equal to the weight of the bulk of the liquid that it displaces. The upward pressure or buoyancy of the liquid may be regarded as exerted at

the center of gravity of the displaced water, which is called the center of pressure or of buoyancy. A vertical line drawn through it is called the axis of buoyancy or of flotation. In a floating body at rest a line joining the center of gravity and the center of buoyancy is vertical, and is called the axis of equilibrium. When an external force causes the axis of equilibrium to lean, if a vertical line be drawn upward from the center of buoyancy to this axis, the point where it cuts the axis is called the *metacenter*. If the metacenter is above the center of gravity the distance between them is called the metacentric height, and the body is then said to be in stable equilibrium, tending to return to its original position when the external force is removed.

Table 2.—Weight of Water and Heat Units per Pound at Different Temperatures

Temp., deg. F.	Lb. per cu. ft.	B.t.u. per lb.	Temp., deg. F.	Lb. per cu. ft.	B.t.u. per lb.	Temp., deg. F.	Lb. per cu. ft.	B.t.u. per lb.	Temp., deg. F.	Lb. per cu. ft.	B.t.u. per lb.
32	62.41	0.	91	62.10	58.99	150	61.19	117.84	208	59.92	175.98
33	62.41	1.01	92	62.08	59.98	151	61.17	118.85	209	59.90	176.99
34	62.42	2.01	93	62.07	60.98	152	61.15	119.85	210	59.87	177.99
35	62.42	3.02	94	62.06	61.97	153	61.13	120.85	211	59.85	179.00
36	62.42	4.03	95	62.05	62.96	154	61.11	121.85	212	59.82	180.00
37	62.42	5.03	96	62.04	63.96	155	61.09	122.85	214	59.81	182.02
38	62.42	6.04	97	62.02	64.95	156	61.07	123.85	216	59.77	184.03
39	62.42	7.04	98	62.01	65.94	157	61.05	124.85	218	59.70	186.04
40	62.42	8.05	99	62.00	66.94	158	61.03	125.85	220	59.67	188.06
41	62.42	9.05	100	61.99	67.93	159	61.01	126.85	230	59.42	198.15
42	62.42	10.05	101	61.98	68.92	160	60.99	127.85	240	59.17	208.26
43	62.42	11.05	102	61.96	69.92	161	60.97	128.85	250	58.89	218.39
44	62.42	12.05	103	61.95	70.91	162	60.95	129.85	260	58.62	228.55
45	62.42	13.05	104	61.94	71.91	163	60.93	130.85	270	58.34	238.74
46	62.41	14.06	105	61.93	72.91	164	60.91	131.85	280	58.04	248.95
47	62.41	15.06	106	61.91	73.90	165	60.89	132.85	290	57.74	259.20
48	62.41	16.06	107	61.90	74.90	166	60.87	133.85	300	57.41	269.48
49	62.41	17.06	108	61.89	75.90	167	60.85	134.85	310	57.08	279.80
50	62.40	18.06	109	61.87	76.89	168	60.83	135.85	320	56.75	290.17
51	62.40	19.06	110	61.86	77.89	169	60.81	136.85	330	56.40	300.59
52	62.40	20.06	111	61.84	78.89	170	60.79	137.85	340	56.02	311.05
53	62.39	21.06	112	61.83	79.89	171	60.77	138.85	350	55.65	321.55
54	62.39	22.06	113	61.81	80.89	172	60.75	139.85	360	55.25	332.10
55	62.38	23.06	114	61.80	81.89	173	60.73	140.85	370	54.85	342.71
56	62.38	24.05	115	61.78	82.89	174	60.71	141.85	380	54.47	353.39
57	62.38	25.05	116	61.77	83.88	175	60.68	142.86	390	54.05	364.14
58	62.37	26.05	117	61.75	84.88	176	60.66	143.86	400	53.62	374.96
59	62.37	27.05	118	61.74	85.88	177	60.64	144.86	410	53.19	385.86
60	62.36	28.05	119	61.72	86.88	178	60.62	145.86	420	52.74	396.84
61	62.35	29.05	120	61.71	87.88	179	60.60	146.87	430	52.33	407.91
62	62.35	30.05	121	61.69	88.88	180	60.57	147.87	440	51.87	419.07
63	62.34	31.05	122	61.68	89.88	181	60.55	148.87	450	51.28	430.3
64	62.34	32.04	123	61.66	90.88	182	60.53	149.87	460	51.02	441.7
65	62.33	33.04	124	61.64	91.88	183	60.51	150.87	470	50.51	453.2
66	62.32	34.04	125	61.63	92.87	184	60.49	151.87	480	50.00	465.0
67	62.32	35.04	126	61.61	93.87	185	60.46	152.87	490	49.50	477.0
68	62.31	36.03	127	61.60	94.87	186	60.44	153.88	500	48.78	489.1
69	62.30	37.03	128	61.58	95.87	187	60.42	154.88	510	48.31	501.6
70	62.30	38.03	129	61.56	96.86	188	60.40	155.88	520	47.62	514.2
71	62.29	39.03	130	61.55	97.86	189	60.37	156.89	530	46.95	527.0
72	62.28	40.02	131	61.53	98.86	190	60.35	157.89	540	46.30	540.0
73	62.27	41.02	132	61.51	99.86	191	60.33	158.90	550	45.66	553.2
74	62.26	42.02	133	61.50	100.86	192	60.30	159.90	560	44.84	566.7
75	62.25	43.01	134	61.48	101.85	193	60.28	160.90	570	44.05	580.4
76	62.25	44.01	135	61.46	102.85	194	60.26	161.91	580	43.29	594.4
77	62.24	45.01	136	61.44	103.85	195	60.23	162.91	590	42.37	608.7
78	62.23	46.00	137	61.43	104.85	196	60.21	163.92	600	41.49	623.2
79	62.22	47.00	138	61.41	105.84	197	60.19	164.92	610	40.49	638.0
80	62.21	48.00	139	61.39	106.84	198	60.16	165.93	620	39.37	653.4
81	62.20	49.00	140	61.37	107.84	199	60.14	166.93	630	38.31	669.5
82	62.19	50.00	141	61.36	108.84	200	60.11	167.94	640	37.17	686.6
83	62.18	51.00	142	61.34	109.84	201	60.09	168.95	650	35.97	705.2
84	62.17	52.00	143	61.32	110.84	202	60.07	169.95	660	34.48	725.3
85	62.16	53.00	144	61.30	111.84	203	60.04	170.96	670	32.89	747.5
86	62.15	54.00	145	61.28	112.84	204	60.02	171.96	680	31.06	772.6
87	62.14	55.00	146	61.26	113.84	205	59.99	172.97	690	28.82	803.0
88	61.13	56.00	147	61.25	114.84	206	59.97	173.97	700	25.38	846.3
89	62.12	57.00	148	61.23	115.84	207	59.95	174.98	706.1	19.16	925.0
90	62.11	58.00	149	61.21	116.84	.....	.....	......	......	......	......

Table 3.—Comparison of Heads of Water in Feet with Pressures in Various Units

One foot of water at 39 1° Fahr. = 62.425 lb. on the square foot;
" " " = 0.4335 lb. on the square inch;
" " " = 0.0295 atmosphere;
" " " = 0.8826 inch of mercury at 32°;
" " " = 773.3 feet of air at 32° and atmospheric pressure;

One lb. per square foot, at 39.1° Fahr.	=	0.01602	foot of water;
One lb. per square inch, at 39.1° Fahr.	=	2.307	feet of water;
One atmosphere of 29.922 in. of mercury	=	33.9	feet of water;
One inch of mercury at 32°	=	1.133	feet of water;
One foot of air at 32°, and 1 atmosphere	=	0.001293	foot of water;
One foot of average sea-water	=	1.026	feet of pure water;
One foot of water at 62° F.	=	62.355	lb. per sq. foot;
One foot of water at 62° F.	=	0.43302	lb. per sq. inch;
One inch of water at 62° F. = 0.5774 ounce.	=	0.036085	lb. per sq. inch;
One lb. of water per square inch at 62° F.	=	2.3094	feet of water;
One ounce of water per square inch at 62° F.	=	1.732	inches of water.

Table 4.—Feet or Inches of Water Expressed in Pounds per Square Inch

Depth in Inches

In.	Lb. per sq. in.	In.	Lb. per sq. in.	In.	Lb. per sq. in.	In.	Lb. per sq. in.
0.01	0.00036	0.1	0.00361	1.0	0.036085	10.0	0.36085
.02	.00072	.2	.00722	2.	.07217	20.	.72170
.03	.00108	.3	.01083	3.	.10826	30.	1.08255
.04	.00144	.4	.01443	4.	.14434	40.	1.44340
.05	.00180	.5	.01804	5.	.18043	50.	1.80425
.06	.00217	.6	.02165	6.	.21651	60.	2.16510
.07	.00253	.7	.02526	7.	.25260	70.	2.52595
.08	.00289	.8	.02887	8.	.28863	80.	2.88680
.09	.00325	.9	.03248	9.	.32477	90.	3.24765

Depth in Feet

Ft.	Lb. per sq. in.	Ft.	Lb. per sq. in.	Ft.	Lb. per sq. in.	Ft.	Lb. per sq. in.
0.01	0.00433	0.1	0.04330	1.0	0.43302	10	4.3302
.02	.00866	.2	.08660	2.	.86604	20	8.6604
.03	.01299	.3	.12991	3.	1.29906	30	12.9906
.04	.01732	.4	.17321	4.	1.73208	40	17.3208
.05	.02165	.5	.21651	5.	2.16510	50	21.6510
.06	.02598	.6	.25981	6.	2.59812	60	25.9812
.07	.03031	.7	.30311	7.	3.03114	70	30.3114
.08	.03464	.8	.34642	8.	3.46416	80	34.6416
.09	.03897	.9	.38972	9.	3.89718	90	38.9718

The pressure corresponding to any depth can be found by addition.

EXAMPLE.—Pressure corresponding to a depth of 37.42 in. equals

$$1.08255 + 0.25260 + 0.01433 + 0.00072 = 1.3502 \text{ lb. per sq. in.}$$

Table 5.—Pounds per Square Inch Equivalent to Inches of Water

Lb. per sq. in.	In. of Water	Lb. per sq. in.	In. of Water	Lb. per sq. in.	In. of Water	Lb. per sq. in.	In. of Water
0.01	0.27712	0.1	2.77123	1.0	27.7123	10.0	277.123
.02	.55425	.2	5.54246	2.0	55.4246	20.0	554.246
.03	.83137	.3	8.31369	3.0	83.1369	30.0	831.369
.04	1.10849	.4	11.0849	4.0	110.849	40.0	1108.49
.05	1.38562	.5	13.8562	5.0	138.562	50.0	1385.62
.06	1.66274	.6	16.6274	6.0	166.274	60.0	1662.74
.07	1.93986	.7	19.3986	7.0	193.986	70.0	1939.86
.08	2.21698	.8	22.1698	8.0	221.698	80.0	2216.98
.09	2.49411	.9	24.9411	9.0	249.411	90.0	2494.11

BOILING-POINT.—Water boils at 212° F. (100° C.) at mean atmospheric pressure at the sea-level, 14.696 lb. per square inch. The temperature at which water boils at any given pressure is the same as the temperature of saturated steam at the same pressure. For boiling-point of water at other pressures than 14.696 lb. per square inch, see table of the Properties of Saturated Steam, p. 5–04.

The Boiling-Point of Water May Be Raised.—When water is entirely freed of air, which may be accomplished by freezing or boiling, the cohesion of its atoms is greatly increased, so that its temperature may be raised over 50° above the ordinary boiling-point before ebullition takes place.

It was found by Faraday that when such air-freed water did boil the rupture of the liquid was like an explosion. When water is surrounded by a film of oil, its boiling temperature may be raised considerably above its normal standard. This has been applied as a theoretical explanation of some boiler explosions.

The freezing point also may be lowered, if the water is perfectly quiet, to −10° C., or 18° Fahrenheit below the normal freezing-point. (Hamilton Smith, Jr., Hydraulics, p. 13.)

FREEZING-POINT.—Water freezes at 32° F. at the ordinary atmospheric pressure, and ice melts at the same temperature. In the melting of 1 pound of ice into water at 32° F. about 143.6 heat-units are absorbed, or become latent; and in freezing 1 lb. of water into ice a like quantity of heat is given out to the surrounding medium.

Sea-Water freezes at 27° F. The ice is fresh.

ICE AND SNOW. (From Clark.)—1 cu. ft. of ice at 32° F. weighs 57.50 lb.; 1 lb. of ice at 32° F. has a volume of 0.0174 cu. ft. = 30.067 cu. in.

Relative volume of ice to water at 32° F., 1.0855, the expansion in passing into the solid state being 8.55%. Specific gravity of ice = 0.922, water at 62° F. being 1.

At high pressures the melting-point of ice is lower than 32° F., being at the rate of 0.0133° F. for each additional atmosphere of pressure.

The specific heat of ice is 0.504, that of water being 1. The latent heat of fusion of ice is 143.6 B.t.u. per lb.

1 cu. ft. of fresh snow, according to humidity of atmosphere, weighs from 5 lb. to 12 lb. 1 cu. ft. of snow moistened and compacted by rain weighs from 15 lb. to 50 lb. (Trautwine.)

COMPRESSIBILITY OF WATER.—Water is very slightly compressible. Its compressibility is from 0.000040 to 0.000051 for one atmosphere, decreasing with increase of temperature. For each foot of pressure, distilled water will be diminished in volume 0.0000015 to 0.0000013. Water is so incompressible that even at a depth of a mile a cubic foot of water will weigh only about half a pound more than at the surface.

Table 6.—Solubility of Oxygen in Distilled Water at Different Pressures and Temperatures

(Geo. C. Whipple in paper before Am. Water Works Assoc., *Power*, June 25, 1912)

Temp., deg. Fahr.	Water Pressure, lb. per sq. in.					
	0	20	40	60	80	100
	Parts per Million					
32	14.7	34.7	54.7	74.7	94.7	114.7
50	11.3	26.7	42.2	57.5	73.0	88.1
68	9.2	21.7	34.2	46.8	59.2	71.7
86	7.6	17.9	28.3	38.6	48.9	59.2
104	6.2	14.6	23.1	31.4	39.8	48.2
122	4.8	11.4	17.9	24.5	31.1	37.6
140	3.6	8.4	13.3	18.1	22.9	27.7
176	1.7	3.9	6.2	8.4	10.7	12.9
194	1.0	2.2	3.5	4.8	6.1	7.4
212	0					

IMPURITIES OF WATER.—For a discussion of the impurities in water and of water purification, see Boiler Feedwater, under Steam Boilers, p. 6–63.

HYDRAULICS

By William P. Creager

1. FLOW THROUGH ORIFICES AND SHORT TUBES

ORIFICES AND SHORT TUBES.—Fig. 1 shows typical examples of orifices and short tubes. A knowledge of the laws of the flow of water through them is necessary to determine the discharge through sluiceways and the entrances to conduits. If the entrance is not properly shaped, a contraction of the jet occurs as in *a*, *c* and *h*, Fig. 1, and the area of the jet is not as great as that of the orifice or tube. For properly rounded approaches to orifices, as in *b* and *e*, and in the constant-diameter short tubes in *d*, *f* and *g*, the diameter of the jet equals the area of the orifice or tube. In short tubes without rounded entrances, the contraction does occur, but the jet, with certain exceptions as explained below, re-expands as indicated, a partial vacuum occurring just inside the entrance.

Let H = head of water, ft., on the center line of a freely flowing orifice or tube, or the difference in water level for a submerged orifice or tube; a = area, sq. ft., of the orifice or tube; v = theoretical velocity, ft. per sec., corresponding to head H; g = acceleration of gravity = 32.2 ft. per sec.; Q = discharge, cu. ft. per sec.; C_2 = coefficient of contraction, or ratio of area of jet to area of orifice or tube; C_1 = coefficient of friction; C = coefficient of discharge.

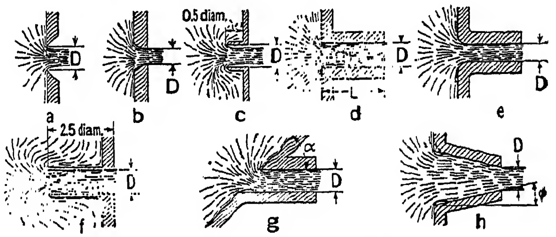


FIG. 1. Types of Orifices

The general equation for the velocity of spouting water is $v = \sqrt{2gH}$ . . . . [1]

Considering friction, the actual velocity due to head H is $v = C_1\sqrt{2gH}$.

The discharge is equal to the product of the actual velocity and the area of the jet, or, since the area of the jet is C_2a, $Q = vC_2a$, or $Q = C_1C_2a\sqrt{2gH}$.

In experiments conducted to determine the discharge through orifices and tubes, the coefficient of friction, C_1, and the coefficient of contraction, C_2, are combined and the general equation for the discharge is $Q = Ca\sqrt{2gH}$ . . . . [2].

The value of C_1 varies with the shape of the orifice or tube. Table 1 gives average experimental values of C, for use in Equation [2] for several types of orifices and tubes.*

These orifices and tubes are circular unless otherwise noted. According to Bovey and other authorities on orifices in thin plates (Fig. 1*a*), the value of C for circular orifices is about 2% less than for square orifices; from 3 to 4% less than for rectangular orifices having a ratio of length to height of 4; and from 5 to 7% less than for rectangular orifices having a ratio of length to height of 10. A similar relation probably exists for tubes having square-cornered entrances. The coefficient C is *not greatly* affected by submergence.

Coefficients for short tubes apply only to heads less than about 40 ft. For higher heads the expansion heretofore explained does not occur and the coefficients C approach those for orifices of similar type.

The expansion of the jet within the short tubes may not occur if they are not submerged and if sufficient friction is lacking, *i.e.*, the jet, after contracting, may pass through the tube without touching its sides, if under a high head; even if the trajectory is such that the jet strikes the bottom of the tube, expansion will not occur if the friction along the bottom is insufficient.

* Where the head is large in comparison with size of orifice or tube. For head or orifice equal to (1.5 × height of orifice), results are about 1% too large.

The shape of the jet from a sharp-cornered round orifice, Fig. 1*a*, between the orifice and the vena contracta, or section at which the area of the jet becomes constant, is indicated in Fig. 2 * and should be used for rounded entrances as in Fig. 1*e*.

Values of C for vertical orifices with full contraction, Fig. 1*a*, with a free discharge into the air, with the inner face of the plate, in which the orifice is pierced, plane, and with sharp inner corners so that the escaping vein only touches these inner edges are given in Table 2, which is an abridgment of tables compiled by Hamilton Smith, Jr., from many experimental data.

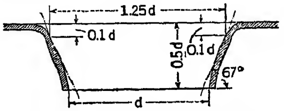


Fig. 2. Standard Mouthpiece

The coefficients in Tables 1 and 2 have been obtained from experiments under ideal conditions. As an indication of the smaller discharge coefficients obtained for practical cases, Fig. 3 gives the results of Stewart's experiments. (Investigation of Flow through Large Submerged Orifices and Tubes, C. B. Stewart, Univ. of Wis. *Bull.* 216, 1908, on the discharge through

* A. H. Gibson, after Weisbach.

Table 1.—Coefficients of Discharge, C, through Orifices and Tubes for Equation [2]
(Circular Except as Noted)

No.	Type of Orifice							
1 *a*	In thin plate, C =							0.60
1 *b*	Rounded, C =							0.97
1 *c*	Inwardly projecting, C =							0.50
1 *d*	Short tube, sharp-cornered entrance *:							
	L/D = 0	0.25	0.50	0.75	1.00	1.50	2.50	3.50
	C = 0.60	0.63	0.67	0.72	0.76	0.79	0.80	0.80
1 *e*	Short tube, rounded entrance, C =							0.97
1 *f*	Inwardly projecting, sharp-cornered entrance, C =							0.72–0.80
1 *g*	Inclined short tube sharp-cornered entrance †:							
	90°	80°	70°	60°	50°	40°		
	C = 0.82	0.80	0.78	0.76	0.75	0.73		0.72
1 *h*	Convergent short tube ‡:							
	ϕ	0°	5.75°	11.25°	22.5°			45°
	C (sharp-cornered entrance)	0.82	0.94	0.92	0.85			....
	C (rounded entrance) =	0.97	0.95	0.92	0.88			0.75

* From experiments by Rogers and Smith on submerged tubes (*Eng. News*, vol. lxxvi, p. 827, 1916). The coefficient for L/D = 2.5 and more has been found by other experimenters to be 0.82. † According to Weisbach. ‡ H. W. King after Unwin.

Table 2.—Values of Coefficient C for Vertical Orifices with Sharp Edges, Full Contraction, and Free Discharge into Air

Head from Center of Orifice H	Square Orifices. Length of the Side of the Square in Feet												
	.02	.03	.04	.05	.07	.10	.12	.15	.20	.40	.60	.80	1.0
0.4	.....	.....	.643	.637	.628	.621	.616	.611	.....	.....	.....	.....	.....
0.6	.660	.645	.636	.630	.623	.617	.613	.610	.605	.601	.598	.596	.....
1.0	.648	.636	.628	.622	.618	.613	.610	.608	.605	.603	.601	.600	.599
3.0	.632	.622	.616	.612	.609	.607	.606	.606	.605	.605	.604	.603	.603
6.0	.623	.616	.612	.609	.607	.605	.605	.605	.604	.604	.603	.602	.602
10.	.616	.611	.608	.606	.605	.604	.604	.603	.603	.603	.602	.602	.601
20.	.606	.605	.604	.603	.602	.602	.602	.602	.602	.601	.601	.601	.600
100.(?)	.599	.598	.598	.598	.598	.598	.598	.598	.598	.598	.598	.598	.598
	Circular Orifices. Diameters in Feet												
H	.02	.03	.04	.05	.07	.10	.12	.15	.20	.40	.60	.80	1.0
0.4	.....	.....	.....	.637	.628	.618	.612	.606	.....	.....	.....	.....	.....
0.6	.655	.640	.630	.624	.618	.613	.609	.605	.601	.596	.593	.590	.....
1.0	.644	.631	.623	.617	.612	.608	.605	.603	.600	.598	.595	.593	.591
2.	.632	.621	.614	.610	.607	.604	.601	.600	.599	.599	.597	.596	.595
4.	.623	.614	.609	.605	.603	.602	.600	.599	.599	.598	.597	.597	.596
6.	.618	.611	.607	.604	.602	.600	.599	.599	.598	.598	.597	.596	.596
10.	.611	.606	.603	.601	.599	.598	.598	.597	.597	.597	.596	.596	.595
20.	.601	.600	.599	.598	.597	.596	.596	.596	.596	.596	.596	.595	.594
50.(?)	.596	.596	.595	.595	.594	.594	.594	.594	.594	.594	.594	.593	.593
100.(?)	.593	.593	.592	.592	.592	.592	.592	.592	.592	.592	.592	.592	.592

4 ft. square submerged sluices of different types). The minimum value of C for each set of experiments is shown. The forms of the entrances to the sluices, Fig. 4, are given as follows: Series A, Square entrance; Series a, Contraction suppressed on bottom; Series b, Contraction suppressed on bottom and one side; Series c, Contraction suppressed on bottom and two sides; Series d, Contraction suppressed on bottom, two sides and top.

The peculiar shape of the curve for Series c is probably an error in the experiments. The experiments covered heads ranging from 0.05 to 0.30 ft. It was found that C was least for heads of 0.15 ft., and these losses are the ones indicated in Fig. 3.

In these experiments, Series A corresponds to Item 1d of Table 1, in which C varied from 0.60 to 0.80, as compared with Stewart's coefficients of 0.60 to 0.784, which is a reasonably close agreement; but Series d corresponds to Item 1e of Table 1 in which C is given as 0.97, as compared with Stewart's variation of 0.925 to 0.882. In the latter case, the considerably lower values of Stewart's coefficient probably were caused in part by an imperfect mouthpiece.

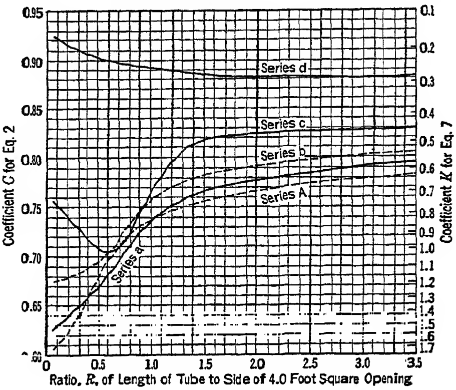


Fig. 3. Values of C and K

VELOCITY OF APPROACH.—If the velocity in the channel of the approach were uniform, the actual head on the orifice or tube would, in reality, be the measured head plus the head corresponding to the velocity of approach. The velocity of approach, however, is not uniform through any section of the channel, being less than the average at the sides and bottom. The velocity directly opposite the orifice therefore is greater than the average, and the effective head on the orifice is the measured head plus a head somewhat greater than that corresponding to the velocity of approach.

Equation [2], corrected for velocity of approach, may be written

$$Q = Ca\sqrt{2g}\,(H + \beta h_v), \quad . \quad . \quad [3]$$

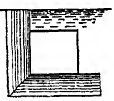


Fig. 4. Stewart's Sluice, Showing Contraction Suppressed on Bottom and Two Sides

where v = average velocity of approach, h_v = head corresponding to this velocity, and β a coefficient which must be determined experimentally. Unfortunately, β is not well known for many types of orifices, and may vary between 1.0 and 2.0, depending upon the location and relative size of the orifice.

RECTANGULAR ORIFICES AND TUBES UNDER LOW HEAD.—If the area of the freely flowing orifice or tube is large in comparison with the head, the equation [1] should be written

$$Q = (2LC/3)\sqrt{2g}\,(H_1^{3/2} - H_2^{3/2}), \quad . \quad . \quad . \quad . \quad [4]$$

where H_1 and H_2 = heads on the bottom and top of the orifice, respectively. If the orifice is completely submerged, the head is the difference in level between the upper and lower water surfaces, and Equation [2] applies.

If, in Equation [4] the top of the orifice is at the water surface, $H_2 = 0$ and the equation reduces to:

$$Q = (2LC/3)\sqrt{2g}\,H^{3/2}. \quad . \quad . \quad . \quad . \quad [5]$$

This is the basic theoretical equation for discharge over weirs. A. H. Gibson says that Equation [2] can be used with an error not greater than 1% if the depth of water to the top of the freely flowing orifice is greater than twice the height of the orifice.

DISCHARGE THROUGH SLUICE GATES.—Sluice gates are made in a variety of forms. Many types of sluices, controlled by sluice gates, are in reality short conduits in which skin friction is a large percentage of the total loss. A discussion of the losses through conduits is given later. See p. 2–10. If the sluice is short, the discharge may be considered as that through a short tube (Figs. 1d to 1h inclusive), or, if the sluice gate is in a thin wall (Figs. 1a to 1c inclusive), the discharge may be obtained from Equa-

tions [2] or [3] with the proper coefficient of discharge selected from Table 1 or Fig. 3 according to the details of the sluice gate opening.

2. FLOW OF WATER IN CONDUITS

HEAD BY BERNOULLI'S THEOREM.—The velocity head, or head required to produce a given velocity, is,

$$\text{Velocity head} = h = v^2/2g \quad . \quad . \quad . \quad . \quad . \quad . \quad . \quad . \quad [5a]$$

According to Bernoulli's Theorem, the general law governing steady flow of water in conduits is as follows: For steady flow in a conduit, the sum of the velocity head, the pressure head, and the potential head at any point, A, is equal to the sum of the corresponding heads at any upstream point, B, less the frictional resistance between the points A and B, Fig. 5.

Expressed mathematically, we have:

$$(v^2/2g) + h_p + h_e = (v_1^2/2g) + h_p' + h_e' - h_f. \quad . \quad . \quad . \quad . \quad . \quad [5b]$$

where v and v_1 = the velocities, ft. per sec., at points A and B respectively; h_p and h_p' = corresponding pressure heads, ft.; h_e and h_e' = the corresponding potential heads above a common datum plane, ft.; h_f = total frictional resistance or lost head, ft., between points A and B.

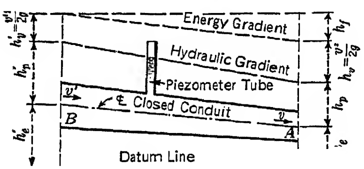


FIG. 5. Closed Conduits

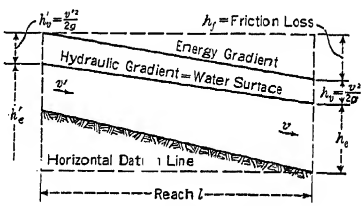


FIG. 6. Open Conduits

If a vertical or oblique tube, Fig. 5, be inserted in a pipe containing water under pressure, the water will rise in the tube, and the vertical height to which it rises will be the head, h_p, producing the pressure at the point where the tube is attached. Such a tube is called a piezometer or pressure measure. If the water in the piezometer falls below its proper level, it shows that the pressure in the main pipe has been reduced by an obstruction between the piezometer and the reservoir. If the water rises above its proper level, it indicates that the pressure there has been increased by an obstruction beyond the piezometer.

If we imagine a pipe full of water to be provided with a number of piezometers, then a line joining the tops of the columns of water in them is the hydraulic gradient, and $h_p + (v^2/2g)$ defines the corresponding elevation of the energy gradient.

In an open conduit, h_e and h_e' are measured to the water surface, as in Fig. 6, and h_p and h_p', measured to the same place, are equal to zero. Therefore, the hydraulic gradient for open conduits is at water surface.

The head lost by friction and eddies between any two points in a closed or open conduit is equal to the drop in level of the energy gradient between the two points. This in turn is equal to the drop in level of the hydraulic gradient, as shown by piezometer readings (or water surface in open conduits), plus the velocity head at the upper point and less the velocity head at the lower point.

CRITICAL VELOCITY.—In capillary tubes, between plates close together, or in large pipes at very low velocity, the friction head varies directly as the velocity. The velocity below which this takes place is the *critical velocity*. Below the critical velocity, the water flows in straight-line filaments, but above it the flow is turbulent and the exponent of v is increased. The critical velocity extends over a considerable range, depending on whether the velocity is increasing or decreasing. For an increasing velocity the critical velocity will be higher than for a decreasing one. King's Hydraulics gives the following values of critical velocities in feet per second, in pipes at a temperature of 68° F:

Diam., in.	1/2	1	2	4	6	12
Lower value	0.53	0.26	0.13	0.07	0.04	0.02
Higher value	3.35	1.68	0.84	0.42	0.28	0.14

FLOW OF WATER IN CLOSED CONDUITS.—The quantity of water discharged through a pipe depends on the *head, i.e.*, the vertical distance between the level surface of still water in the chamber at the entrance end of the pipe and the level of the center of the discharge end of the pipe; also upon the length of the pipe, upon the character of its interior surface as to smoothness, and upon the number and sharpness of the bends. It is independent of the position of the pipe, as horizontal, or inclined upwards or downwards.

The head, instead of being an actual distance between levels, may be caused by pressure, as by a pump, in which case the head is calculated as a vertical distance corresponding to the pressure, 1 lb. per sq. in. = 2.309 ft. head, or 1 ft. head = 0.433 lb. per sq. in.

The total head operating to cause flow is divided into four parts: 1. The *velocity-head*, which is the height through which a body must fall *in vacuo* to acquire the velocity with which the water flows into the pipe $= v^2 \div 2g$, in which v is the velocity in ft. per sec. and $2g = 64.32$; 2. The *entry-head*, that required to overcome the resistance to entrance to the pipe. With sharp-edged entrance the entry-head = about 1/2 the velocity-head; with smooth rounded entrance the entry-head is inappreciable; 3. The *friction-head*, due to the frictional resistance to flow within the pipe; 4. The *eddy losses* due to bends, valves and sudden changes in area of the pipe. All of these heads are lost except the velocity head, a large part of which may be regained at any point by gradual enlargement of the pipe.

Table 3.—Fall in Feet per Mile, the Distance on Slope Corresponding to 1 Ft. Fall, the Fall in 1000 Ft., the Equivalent Loss in Pressure in Pipes per 1000 Ft. Length

Fall in Feet per Mile	Slope, 1 ft. in	Slope, Feet per 1000	Loss of Pressure per 1000 Feet, lb. per sq. in.	Fall in Feet per Mile	Slope, 1 ft. in	Slope, Feet per 1000	Loss of Pressure per 1000 Feet, lb. per sq. in.
0.25	21120 ft.	0.0473	0.02048	20	264 ft.	3.7879	1.640
.30	17600	.0568	.02459	21.12	250	4.0000	1.732
.40	13200	.0758	.03282	22	240	4.1667	1.804
.50	10560	.0947	.04101	24	220	4.5455	1.968
.60	8800	.1136	.04919	26.4	200	5.0000	2.165
.80	6600	.1515	.06560	28	188.6	5.3030	2.296
1	5280	.1894	.08201	31.68	166.7	6.0000	2.598
1.056	5000	.2000	.08660	35.20	150	6.6667	2.887
1.25	4224	.2367	.1025	42.24	125	8.0000	3.464
1.5	3520	.2841	.1230	44	120	8.3333	3.608
1.75	3017	.3314	.1435	48	110	9.0909	3.936
2	2640	.3788	.1640	52.8	100	10.000	4.330
2.5	2112	.4735	.2050	60	88	11.364	4.913
2.64	2000	.5000	.2165	63.36	83.3	12.000	5.196
3	1760	.5682	.2460	66	80	12.500	5.413
3.5	1508	.6631	.2871	70.4	75	13.333	5.773
4	1320	.7576	.3280	79.20	66.7	15.000	6.495
5	1056	.9470	.4101	88	60	16.667	7.217
5.28	1000	1.0000	.4330	105.6	50	20.000	8.660
6	880	1.1364	.4921	120	44	22.727	9.841
7	754.3	1.3257	.5740	132	40	25.000	10.83
8	660	1.5152	.6561	160	33	30.303	13.12
9	586.6	1.7044	.7380	220	24	41.667	18.04
10	528	1.8939	.8201	264	20	50.000	21.65
10.56	500	2.0000	.8660	330	16	62.500	27.06
12	440	2.2727	.9841	440	12	83.333	36.08
13	406.1	2.4621	1.066	528	10	100.00	43.30
14	377.1	2.6515	1.148	660	8	125.00	54.13
15	352	2.8409	1.230	880	6	166.67	72.17
16	330	3.0303	1.312	1056	5	200	86.60
18	293.3	3.4091	1.476	1320	4	250	108.25

Flow of Water in Open Conduits.—The previously explained theory of the flow of water in closed conduits applies directly to the flow of water in open conduits. In the case of open conduits, however, the elevation of the surface of the water is the piezometer or pressure measure.

Loss of Head in Conduits.—The loss of head in conduits may be divided into two general groups: 1. *Eddy losses*, caused by sudden changes in the direction of flow, as at bends, branches, etc., or by sudden changes in velocity due to sudden changes in area, as at the entrance, sudden enlargements, valves, etc.; 2. *Skin friction* in straight, uniform conduits.

EDDY LOSSES IN CONDUITS.—It is convenient to measure eddy losses in terms of the velocity head of the flowing water. The velocity head, or head required to produce a given velocity, may be obtained by transposing Equation [1]:

$$\text{Velocity head} = h_v = v^2/2g \quad . \quad . \quad . \quad . \quad . \quad . \quad . \quad . \quad [6]$$

Then h_f, the head lost at any point in the conduit due to eddies, is

$$\text{Eddy loss} = h_f = Kh_v = K(v^2/2g) \quad . \quad . \quad . \quad . \quad . \quad . \quad . \quad . \quad [7]$$

where K = coefficient of eddy loss; v = highest velocity at the point under consideration.

LOSSES AT CONDUIT ENTRANCES.—A direct relation exists between the coefficient of discharge for short tubes and the coefficient of eddy loss K, Equation [7], which can be derived as follows: A drop in pressure or head at the entrance to a conduit is required for two purposes: *a*. Velocity head to provide the necessary velocity; *b*. Head to overcome friction due to eddies. Or,

$$H = h_v + h_f = (v^2/2g) + K(v^2/2g) = (v^2/2g)\ (1 + K).$$

As $Q = av$, $H = (Q^2/2ga^2)\ (1 + K)$. Also from Equation [2],

$$H = (Q^2/2ga^2) \times (1/C^2), \quad \text{and} \quad K = (1/C^2) - 1 \quad . \quad . \quad . \quad . \quad . \quad . \quad [8]$$

The loss of head in short tubes where there is no residual contraction in the jet, as in Figs. 1*d*, 1*e*, 1*f* and 1*g*, and in the sluices used in Stewart's experiments, may be assumed to be the same for similar entrances to closed conduits. Values of K derived from the experimental values of C are given in Table 4.

Table 4.—Coefficients for Eddy Loss for Equation [7]

Fig. No.	Type of Conduit						
1 *d*	Short tube with sharp-cornered entrance, K = ... 0.56						
1 *e*	Short tube with rounded entrance, K = ... 0.06						
1 *f*	Inwardly projecting tube with sharp-cornered entrance,* K = ... 0.56–0.93						
	Inclined tube with sharp-cornered entrance:						
	α = 90°	80°	70°	60°	50°	40°	30°
	K = 0.49	0.56	0.65	0.73	0.78	0.88	0.93

* Depending upon distance of projection.

Values of K for the sluices used in Stewart's experiments, which also are applicable to losses at entrances of similar type, are given in Fig. 3.

LOSSES AT CONDUIT INTAKES.—Losses at the intake racks are usually very small because the velocity must be low to facilitate raking. For clear racks, a safe approximate value of K for use in Equation [7] may be obtained from

$$K = 1.45 - 0.45R - R^2 \quad . \quad . \quad . \quad . \quad . \quad . \quad . \quad . \quad . \quad [9]$$

where R = ratio of net to gross area of racks and supports; v in Equation [7] = velocity in the net area of racks and supports. For a usual value of R = 0.65, the resulting value of K is 0.74, and for a usual velocity of 2.5 ft. per sec. in net area, the loss is 0.072 ft. However, allowance should be made for partial closure of the racks with trash. From 25 to 50% of the area of hand-raked racks frequently is obstructed in practical operations if the amount of debris in the water is large. This would increase the above loss to from 0.13 to 0.29 ft. The head losses in well-designed intakes, excepting the loss at racks and gates, are usually negligible, as changes in area are gradual.

Losses at head gates correspond to the losses previously given for conduit entrances. The worst practical condition is that in which the gate is at the head of an entrance similar to Fig. 1*d*. It is possible usually to provide a flare at the upper end of the intake, to produce a gradually increasing velocity between racks and gates, with sides and bottom of the intake nearly flush with the gate opening. In such cases there is a decided eddy only at the top of the gate, and the coefficient K is no greater than about 0.5 in Equation [7], where v is the velocity through the gate. The gradual increase in velocity between the gate and the normal section of the conduit is made without appreciable loss.

Intakes to open flumes vary greatly in type according to the conditions to be met and the ideas of the designer. Losses at the gates must be estimated by comparison with the nearest type of test model indicated in Fig. 3.

EDDY LOSSES AT CONDUIT BENDS.—In the present discussion, values of K used in Equation [7] to derive the losses in bends of conduits, are such as to give a loss in excess of that occurring in a straight pipe of equal length; that is, normal skin friction is not included. The conclusion of a number of experimenters concerning the value of K for 90° bends in closed conduits show an extreme lack of agreement.

Experiments on closed conduits (*Trans.* A.S.C.E., vol. lxii, p. 97; Univ. of Wis. *Bull.* No. 578) tend to show that bends where R_b/d = 5 (approx.) correspond to the minimum

loss in the bend, and hence to the minimum value of K as indicated in Fig. 7. R_b = radius of bend and d = diam. of conduit. Fig. 7 gives recommended values of K for 90° bends in closed conduits, based on available data. The curve representing safe values is an enveloping curve for all experiments. The curve representing probable values is the mean of some of the more recent experiments. Values of K from Fig. 7 should be increased by 50% for screwed pipe elbows, due to the sudden enlargement and contraction of such fittings. It is recommended that values of K for bends other than 90° be taken from Fig. 8.

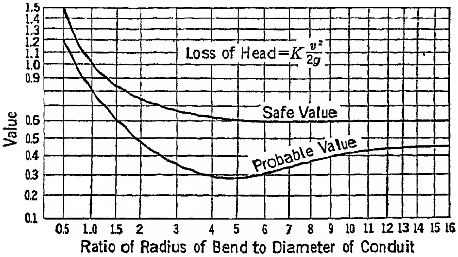


Fig. 7. Values of K for 90-deg. Bends

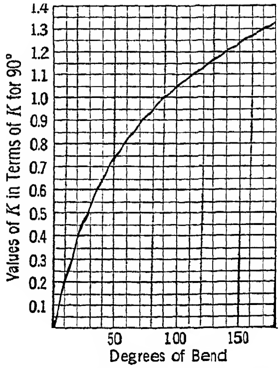


Fig. 8. Values of K for Bends

Few experiments have been made on the losses in bends of open conduits. Fortunately, sharp bends in open conduits usually are not required. Writers often allow for loss at bends of open conduits by adjusting the coefficient C or n (see p. 2–18) of the flow equation. A coefficient is given for a straight open conduit and another coefficient for the same type of conduit "with moderate curvature" or other equally indefinite description. The author believes that no allowance for bends should be made in the flow coefficient C or n, but that the slope for straight alignment should be determined first, and increased slopes provided at bends, corresponding to the best judgment as to additional losses at bends. The few experiments that have been made seem to indicate that the loss due to bends in open conduits is much less than for closed conduits, and values equal to one-half of those in Fig. 7 are recommended. The author has compared such values with experimental data and has found them to agree very closely when one-half the "probable curve" value is used.

For reverse bends in both open and closed conduits, the loss is probably somewhat greater than if the two bends were separated by a considerable length of straight pipe, but the amount of such loss has not been determined.

Recommended safe values for K for miscellaneous fittings are given in Fig. 9.

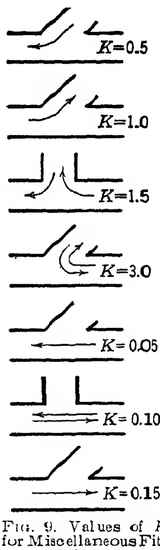


Fig. 9. Values of K for Miscellaneous Fittings

EDDY LOSSES AT CONDUIT VALVES.—The value of K for use in Equation [7] for wide-open gate valves is probably less than 0.1, although few experiments are available. E. A. Dow's experiments on the disc-arm type of butterfly valves indicate a value of

$$K = t/d \quad . \quad . \quad . \quad . \quad . \quad . \quad . \quad . \quad [10]$$

where t = thickness of valve disc; d = valve diameter.

The value of v for use in Equation [7] is that used for the normal section of the conduit.

The author found only three experiments on valves of the Johnson plunger type. On the basis of these experiments he has devised the equation,

$$K = 0.183/\sqrt[3]{d} \quad . \quad . \quad . \quad . \quad . \quad . \quad . \quad . \quad . \quad . \quad [11]$$

where d = diam. at small end, ft. The value of v for use in Equation [7] is that for the small end of the valve.

MISCELLANEOUS CONDUIT EDDY LOSSES.—As all losses, other than those due to skin friction and bends, are due to sudden changes in section of the conduit, knowledge

of the laws governing losses due to sudden contractions and enlargements will assist materially in determining losses caused by various irregularities in conduits. Archer proposes (*Trans.* A.S.C.E., vol. lxxvi, pp. 999-1026, 1913) for the loss due to sudden enlargement in a closed conduit, the equation,

$$h_f = 1.098\ (v_1 - v_2)^{1.919}/2g = 0.01705\ (v_1 - v_2)^{1.919}, \quad . \quad . \quad . \quad . \quad [12]$$

where v_1 and v_2 = velocities in the smaller and larger sections respectively.

An approximate equation giving losses which are correct within 10% for values of $(v_1 - v_2)$, between 1 and 13 ft. per sec. is

$$h_f = (v_1 - v_2)^2/2g. \quad . \quad . \quad . \quad . \quad . \quad . \quad [13]$$

Losses due to gradual enlargements in a closed conduit are not known exactly. Etcheverry gives the value of

$$K = \{1 - (a_1/a_2)\}^2 \sin\theta, \quad . \quad . \quad . \quad . \quad . \quad . \quad [14]$$

where θ = angle formed by intersection of one side of the taper with the center line; a_1 and a_2 = areas of the smaller and larger sections respectively.

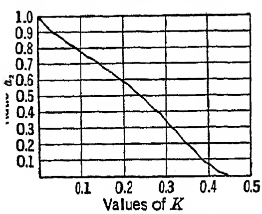


FIG. 10. Values of K for Sudden Contractions in Closed Conduits

Equations [12], [13] and [14] often are used to determine the value of K for sudden enlargements in open conduits and for the junction of a closed and open conduit. This is not correct, as the fundamental principles used in their derivation do not apply to open-water conditions. If the maximum possible loss is to be determined, the value of K should be taken from the equation *

$$K = 1 - (a_1^2/a_2^2). \quad . \quad . \quad . \quad . \quad [15]$$

The value for v for use in Equation [7] is that in the smaller section. This loss is reduced for gradual enlargements in open conduits. Unfortunately, the ratio between the rate of enlargement and the loss is not known exactly, but the loss probably will not exceed 25%, *i.e.*, $K = 0.25$, if the angle of the flare of each side of the conduit makes an angle with the center line not in excess of 2° or 3°.

For sudden contractions in closed conduits, the values of K are given in Fig. 10, which was derived in a manner similar to that employed by Merriman, except that the coefficient of contraction for a sharp-edged orifice, equal to 0.60, was used instead of 0.62 as employed by him in his basic equation. The value of v for use in Equation [7] is that in the smaller section. Losses due to gradual contraction are very small. The loss in gradual contractions in open conduits is negligible.

SKIN FRICTION CONDUIT LOSSES.—Most of the many empirical equations for the uniform flow of water in conduits are based on the equation developed by Chezy in 1775.

$$\text{or in its more general form, } v = CR^nS^m, \quad . \quad . \quad . \quad . \quad [16]$$

where v = average velocity, ft. per sec.; R = hydraulic mean radius, ft., being the cross-sectional area divided by the wetted perimeter; S = sine of the angle of slope of the energy gradient; C, n and m = coefficients depending upon the type of the conduit and the condition of the wetted perimeter, all of which must be determined by experiment.

None of the empirical equations agree very closely with one another and, moreover, they all embody the coefficient C, which must be assumed according to judgment as indicated by available experiments, depending upon the nature and condition of the wetted perimeter. In some equations, n and m are constant for all types of conduits. Other writers have suggested different values for each type of conduit.

A sufficient number of experiments have not been made to enable the engineer to adopt precise coefficients C. The character of the wetted perimeter is liable to considerable change during the life of the conduit, due to corrosion, tuberculation, growth of fungi, weeds, and other vegetation, silt deposits, scour, ice, etc. The chief difficulty in the use of existing experimental data is the lack of the power of definite description of the condition of the wetted perimeter of the experimental conduit. For these reasons, a great refinement in flow equations seems unwarranted at present, and a considerable factor of safety must be adopted to compensate for possible errors in the adopted equation and in the choice of the coefficient C.

At present (1935), Kutter's equation or its approximation, Manning's equation, is mostly used for flow in open channels and the Williams and Hazen equation for flow in closed conduits.

* The exception is where the hydraulic jump is introduced.

Kutter's equation is

$$\frac{\dfrac{1.811}{n} + 41.66 + \dfrac{0.00281}{S}}{0.00281} \sqrt{RS}, \quad . \quad . \quad . \quad [17]$$

where n = coefficient of roughness; R = hydraulic mean radius, ft.; S = sine of angle of slope of the energy gradient.

Manning's equation is

$$v = (1.486R^{2/3}S^{1/2})/n \quad . \quad . \quad . \quad [18]$$

Manning's equation is a simplified approximation of Kutter's equation, and the same values of n that would be adopted for Kutter's equation are to be used.

The Manning and Kutter equations agree fairly closely for conditions which usually obtain in practice, but all published values of n were derived from experiments by use of Kutter's equation.

Williams and Hazen's equation as published in their Hydraulic Tables (John Wiley & Sons, 1920) is

$$v = CR^{0.63}S^{0.54}\,0.001^{-0.04} \quad . \quad . \quad . \quad [19]$$

This may be reduced to $\quad v = 1.32\,CR^{0.63}S^{0.54}. \quad . \quad . \quad . \quad [20]$

Table 5 gives values of R^n and S^m for use in the foregoing flow equations.

As the condition of the surface after a few years, and hence the coefficients n and C, are known only approximately, it is necessary to adopt such values as will result in an error, if any, on the side of safety.

The tabulations given hereinafter have been made from a study of all available sources of information regarding experimental determination of the friction coefficient.* The values corresponding to best and worst conditions embrace practically all variations in the data studied. Isolated values, outside the limits given, have been excluded where there was reasonable doubt as to their accuracy. It is believed that the values given cover the range of practical considerations. The *probable* values are those ordinarily used in practice.

Unless otherwise stated, the friction coefficients in the tables are based on straight or slightly sinuous conduits free from the following influences: *a.* Curvature, other than slightly sinuous; *b.* Settlement of open conduits or other defects of construction; *c.* Sediments, rocks, or other deposits washed or fallen in; *d.* Plant growth, moss, etc.; *e.* Ice covering; *f.* Wind movement.

All experimental data were adjusted as closely as possible to correct for curvature, so that all coefficients are practically for straight conduits. A correction to the calculated slope should be made for appreciable curvature, as indicated heretofore under Eddy Losses at Conduit Bends, p. 2–12. Obviously, the results of defects in construction, settlement, and obstructions are indeterminate, although an attempt has been made in the tabulation to give approximate coefficients for certain conduits which should cover the range of reasonable maintenance.

FRICTION OF ICE COVERING.—It is not known that any experiments have been made to determine the friction coefficient of the under side of a sheet of ice. If the ice had the same effect on friction as the bottom and sides of the conduit, the determination of velocity would be made in the same way as for open-water conditions, except that the width of the ice sheet would be included in the wetted perimeter and the area would be the area under the ice. Although experimental data are not available, it is known that the retarding effect of the under side of an ice sheet is much less than that of the ordinary form of earth and rock canals, and probably equal to that of the smoothest conduits In the absence of a better method, it is recommended that, for ice conditions, the coefficient n for the conduit under open-water conditions be used, and a factor W which is a percentage of the width of the ice sheet, depending upon the roughness of the conduit, as indicated below, be included in the wetted perimeter. In the following tabulation n = value of the coefficient for conduit under open-water conditions, and W = percentage of width of ice sheet to be included in the wetted perimeter.

n =	0.010	0.015	0.020	0.025	0.030	0.035	0.040	0.045	0.050	0.055	0.060
W =	100	87	76	66	58	50	43	37	32	28	25

The probable maximum thickness of the ice sheet depends upon climatic conditions

(Continued on p. 2–18)

* Compiled by Fred C. Scobey, Mem. A. S. C. E., in charge of experiments and compilation of data on the carrying capacity of water conduits for the U. S. Dept. of Agriculture, Bureau of Agricultural Engineering, for the past 20 years. Author of U. S. Dept. Agr. *Bul.* 194, The Flow of Water in Irrigation Channels. U. S. Dept. Agr. *Bul.* 376, The Flow of Water in Wood-Stave Pipe. U. S. Dept. Agr. *Bul.* 852, The Flow of Water in Concrete Pipe. U. S. Dept. Agr. *Bul.* 150. The Flow of Water in Riveted Steel and Analogous Pipe.

Table 5.—Powers of Numbers in Flow Formulas

(Maximum error less than 1 percent)

Number	0.5	0.526	0.54	0.555	0.58	0.63	0.65	2/3
	Number to Given Power							
0.00010	0.0100	0.0079	0.0069	0.00602	...	...	...	...
.00011	.0105	.0083	.0073	.00635	...	...	...	...
.00012	.0110	.0087	.0076	.00667	...	...	...	...
.00013	.0114	.0091	.0080	.00697	...	...	...	...
.00014	.0118	.0094	.0083	.00727	...	...	...	...
.00015	.0122	.0098	.0086	.00755	...	...	...	...
.00016	.0126	.0101	.0089	.00782	...	...	...	...
.00017	.0130	.0104	.0092	.00809	...	...	...	...
.00018	.0134	.0108	.0095	.00835	...	...	...	...
.00019	.0138	.0111	.0098	.00860	...	...	...	...
.00020	.0141	.0113	.0101	.00885	...	...	...	...
.00022	.0148	.0119	.0106	.00933	...	...	...	...
.00024	.0155	.0125	.0111	.00980	...	...	...	...
.00026	.0161	.0130	.0116	.0103	...	...	...	...
.00028	.0167	.0136	.0121	.0107	...	...	...	...
.00030	.0173	.0141	.0125	.0111	...	...	...	...
.00032	.0179	.0145	.0130	.0115	...	...	...	...
.00034	.0184	.0150	.0134	.0119	...	...	...	...
.00036	.0190	.0155	.0138	.0122	...	...	...	...
.00038	.0195	.0159	.0143	.0126	...	...	...	...
.00040	.0200	.0163	.0147	.0130	...	...	...	...
.00042	.0205	.0168	.0150	.0134	...	...	...	...
.00044	.0210	.0172	.0154	.0137	...	...	...	...
.00046	.0214	.0176	.0158	.0141	...	...	...	...
.00048	.0219	.0180	.0162	.0144	...	...	...	...
.00050	.0224	.0184	.0165	.0147	...	...	...	...
.00055	.0235	.0193	.0174	.0155	...	...	...	...
.00060	.0245	.0202	.0183	.0163	...	...	...	...
.00065	.0255	.0210	.0191	.0170	...	...	...	...
.00070	.0265	.0219	.0198	.0178	...	...	...	...
.00075	.0274	.0227	.0206	.0185	...	...	...	...
.00080	.0283	.0234	.0213	.0191	...	...	...	...
.00085	.0292	.0242	.0220	.0198	...	...	...	...
.00090	.0300	.0250	.0227	.0204	...	...	...	...
.00095	.0308	.0257	.0233	.0210	...	...	...	...
.0010	.0317	.0264	.0240	.0216	...	...	...	...
.0011	.0332	.0278	.0253	.0228	...	...	...	...
.0012	.0346	.0291	.0265	.0239	...	...	...	...
.0013	.0360	.0304	.0277	.0250	...	...	...	...
.0014	.0374	.0317	.0289	.0260	...	...	...	...
.0015	.0387	.0328	.0299	.0270	...	...	...	...
.0016	.0400	.0339	.0310	.0280	...	...	...	...
.0017	.0412	.0350	.0320	.0290	...	...	...	...
.0018	.0424	.0361	.0330	.0299	...	...	...	...
.0019	.0436	.0371	.0340	.0308	...	...	...	...
.0020	.0447	.0381	.0350	.0317	...	...	...	...
.0022	.0469	.0400	.0368	.0334	...	...	...	...
.0024	.0490	.0420	.0385	.0350	...	...	...	...
.0026	.0510	.0438	.0402	.0366	...	...	...	...
.0028	.0529	.0455	.0419	.0382	...	...	...	...
.0030	.0548	.0472	.0435	.0398	...	...	...	...
.0032	.0566	.0489	.0450	.0412	...	...	...	...
.0034	.0583	.0505	.0464	.0426	...	...	...	...
.0036	.0600	.0520	.0479	.0440	...	...	...	...
.0038	.0616	.0534	.0493	.0454	...	...	...	...
.0040	.0632	.0549	.0507	.0468	...	...	...	...
.0042	.0648	.0563	.0520	.0480	...	...	...	...
.0044	.0664	.0578	.0534	.0492	...	...	...	...
.0046	.0679	.0592	.0548	.0505	...	...	...	...
.0048	.0693	.0606	.0560	.0518	...	...	...	...
.0050	.0707	.0619	.0572	.0530	0.0463	0.0354	0.0320	0.0294
.0055	.0741	.0650	.0601	.0558	.0490	.0377	.0340	.0312
.0060	.0774	.0680	.0630	.0585	.0516	.0399	.0360	.0331
.0065	.0806	.0710	.0659	.0611	.0540	.0419	.0380	.0349
.0070	.0836	.0738	.0688	.0638	.0564	.0439	.0399	.0366
.0075	.0866	.0764	.0714	.0662	.0588	.0458	.0417	.0383

(*Table continued on following page*)

Table 5.—Powers of Numbers Used as Constants in Flow Formulas—*Continued*

(Maximum error less than 1 percent)

Number	0.5	0.526	0.54	0.555	0.58	0.63	0.65	2/3
	Number to Given Power							
.0080	.0894	.0790	.0738	.0685	.0610	.0477	.0435	.0400
.0085	.0922	.0816	.0763	.0708	.0632	.0496	.0451	.0417
.0090	.0948	.0840	.0786	.0730	.0652	.0514	.0468	.0434
.0095	.0974	.0864	.0809	.0752	.0674	.0532	.0485	.0450
.010	.100	.088	.083	.0776	.069	.055	.0501	.047
.011	.105	.093	.087	.0816	.073	.058	.0533	.049
.012	.110	.097	.092	.0856	.077	.061	.0566	.052
.013	.114	.102	.096	.0895	.081	.065	.0596	.055
.014	.118	.106	.099	.0930	.084	.068	.0625	.058
.015	.122	.110	.103	.0965	.088	.071	.0653	.061
.016	.126	.113	.107	.1000	.091	.074	.0681	.063
.017	.130	.117	.111	.104	.094	.076	.0710	.066
.018	.134	.121	.114	.107	.097	.079	.0737	.069
.019	.138	.124	.117	.110	.100	.082	.0762	.071
.020	.141	.128	.121	.113	.104	.085	.0789	.074
.022	.148	.134	.127	.120	.110	.090	.0840	.078
.024	.155	.141	.133	.126	.115	.095	.0888	.083
.026	.161	.147	.139	.132	.121	.100	.0935	.088
.028	.167	.152	.145	.137	.126	.105	.0982	.093
.030	.173	.158	.151	.142	.131	.109	.103	.097
.032	.179	.164	.156	.147	.136	.114	.107	.101
.034	.184	.169	.161	.152	.141	.118	.111	.105
.036	.190	.174	.166	.157	.146	.122	.115	.109
.038	.195	.179	.171	.162	.150	.127	.120	.113
.040	.200	.184	.176	.167	.155	.131	.124	.117
.042	.205	.189	.180	.172	.159	.135	.128	.121
.044	.210	.193	.186	.176	.164	.139	.131	.124
.046	.214	.197	.190	.180	.168	.143	.135	.128
.048	.219	.202	.194	.185	.172	.147	.139	.132
.050	.224	.206	.198	.189	.177	.151	.143	.136
.055	.235	.217	.209	.199	.187	.160	.152	.145
.060	.245	.228	.219	.209	.196	.170	.161	.154
.065	.255	.238	.228	.219	.205	.179	.169	.162
.070	.265	.247	.238	.228	.214	.187	.178	.170
.075	.274	.256	.247	.237	.223	.195	.186	.179
.080	.283	.265	.256	.246	.232	.203	.194	.187
.085	.292	.274	.265	.254	.240	.211	.201	.194
.090	.300	.282	.273	.262	.249	.219	.209	.201
.095	.308	.290	.281	.270	.257	.227	.217	.209
.100	.316	.298	.289	.278	.264	.234	.224	.216
.11	.332	.313	.304	.293	.279	.249	.239	.230
.12	.346	.328	.318	.307	.294	.262	.252	.243
.13	.360	.342	.332	.321	.308	.276	.267	.257
.14	.374	.357	.346	.334	.321	.290	.280	.270
.15	.387	.370	.359	.348	.334	.303	.293	.282
.16	.400	.382	.372	.361	.347	.316	.306	.295
.17	.412	.395	.384	.373	.360	.328	.319	.307
.18	.424	.407	.397	.386	.372	.340	.330	.319
.19	.436	.419	.409	.398	.383	.351	.341	.331
.20	.447	.430	.420	.409	.394	.363	.353	.343
.22	.469	.451	.441	.430	.417	.386	.377	.365
.24	.490	.472	.461	.451	.439	.408	.399	.387
.26	.510	.493	.482	.472	.460	.430	.420	.408
.28	.529	.513	.502	.492	.480	.450	.440	.429
.30	.548	.532	.522	.511	.500	.470	.460	.450
.32	.566	.550	.541	.531	.519	.490	.480	.470
.34	.583	.568	.559	.550	.538	.508	.500	.490
.36	.600	.586	.576	.569	.556	.526	.519	.509
.38	.616	.603	.593	.586	.573	.544	.538	.526
.40	.632	.620	.610	.603	.591	.563	.556	.544
.42	.648	.636	.626	.620	.607	.580	.572	.562
.44	.664	.651	.642	.636	.624	.596	.590	.580
.46	.679	.666	.658	.650	.640	.616	.609	.599
.48	.693	.681	.674	.665	.655	.633	.624	.616
.50	.707	.697	.690	.680	.671	.649	.640	.632
.55	.741	.734	.725	.718	.709	.688	.682	.675

(*Table continued on following page*)

Table 5.—Powers of Numbers Used as Constants in Flow Formulas—*Continued*

(Maximum error less than 1 percent)

Number	0.5	0.526	0.54	0.555	0.58	0.63	0.65	2/3
	Number to Given Power							
.60	.774	.768	.760	.754	.746	.727	.720	.715
.65	.806	.800	.794	.787	.782	.765	.759	.754
.70	.836	.832	.827	.821	.818	.803	.797	.792
.75	.866	.863	.860	.856	.850	.838	.834	.829
.80	.894	.892	.889	.884	.880	.871	.870	.865
.85	.922	.920	.918	.914	.910	.904	.902	.900
.90	.948	.947	.945	.943	.940	.936	.935	.935
.95	.974	.974	.973	.972	.970	.968	.968	.968
1.0	1.00	...	...	...	1.00	1.00	1.00	1.00
1.1	1.05	...	...	...	1.06	1.06	1.07	1.07
1.2	1.10	...	...	...	1.12	1.12	1.13	1.13
1.3	1.14	...	...	...	1.17	1.18	1.19	1.20
1.4	1.18	...	...	...	1.22	1.24	1.25	1.26
1.5	1.22	...	...	...	1.27	1.29	1.30	1.31
1.6	1.26	...	...	...	1.32	1.35	1.36	1.37
1.7	1.30	...	...	...	1.36	1.40	1.42	1.43
1.8	1.34	...	...	...	1.41	1.45	1.47	1.49
1.9	1.38	...	...	...	1.45	1.50	1.52	1.54
2.0	1.41	...	...	...	1.50	1.55	1.58	1.60
2.2	1.48	...	...	...	1.58	1.64	1.67	1.70
2.4	1.55	...	...	...	1.66	1.73	1.77	1.80
2.6	1.61	...	...	...	1.74	1.82	1.86	1.90
2.8	1.67	...	...	...	1.81	1.91	1.95	1.99
3.0	1.73	...	...	...	1.89	1.99	2.04	2.09
3.2	1.79	...	...	...	1.96	2.07	2.13	2.18
3.4	1.84	...	...	...	2.03	2.15	2.22	2.27
3.6	1.90	...	...	...	2.10	2.23	2.30	2.36
3.8	1.95	...	...	...	2.17	2.30	2.39	2.44
4.0	2.00	...	...	...	2.23	2.38	2.47	2.52
4.2	2.05	...	...	...	2.30	2.46	2.54	2.60
4.4	2.10	...	...	...	2.36	2.53	2.62	2.69
4.6	2.14	...	...	...	2.42	2.60	2.70	2.77
4.8	2.19	...	...	...	2.48	2.67	2.77	2.85
5.0	2.24	...	...	...	2.54	2.74	2.84	2.93
5.5	2.35	...	...	...	2.69	2.91	3.02	3.12
6.0	2.45	...	...	...	2.82	3.08	3.20	3.30
6.5	2.55	...	...	...	2.95	3.24	3.38	3.49
7.0	2.65	...	...	...	3.08	3.39	3.55	3.67
7.5	2.74	...	...	...	3.21	3.54	3.71	3.84
8.0	2.83	...	...	...	3.33	3.69	3.87	4.00
8.5	2.92	...	...	...	3.45	3.83	4.02	4.17
9.0	3.00	...	...	...	3.57	3.97	4.18	4.33
9.5	3.08	...	...	...	3.69	4.10	4.33	4.49
10.0	3.16	...	...	...	3.80	4.24	4.48	4.65
11.0	3.32	...	...	...	4.00	4.50	4.75	4.93
12.0	3.46	...	...	...	4.21	4.75	5.02	5.22
13.0	3.60	...	...	...	4.42	5.00	5.30	5.51
14.0	3.74	...	...	...	4.62	5.25	5.57	5.79
15.0	3.87	...	...	...	4.81	5.48	5.83	6.06
16.0	4.00	...	...	...	4.99	5.70	6.07	6.33
17.0	4.12	...	...	...	5.17	5.92	6.30	6.59
18.0	4.24	...	...	...	5.34	6.15	6.55	6.85
19.0	4.36	...	...	...	5.50	6.36	6.78	7.11
20.0	4.47	...	...	...	5.67	6.57	7.01	7.37

at the site, the length, width, and depth of the conduit, and the velocity. It can be estimated only by comparison with similar conduits in the vicinity.

Values of Kutter's Friction Coefficient *n* for Straight Open Conduits

DRY EXCAVATED EARTH CANALS.—In general, the value of n in earth canals increases with the life of the canal unless constantly maintained. Slightly silted waters will " slick " over an original rough surface so that the value of n becomes less. Heavily silted water will decrease n, but also will decrease the area of the water prism. Best conditions are found in tough silt or clay soils, with velocities below scouring limits. $n = 0.016$ may be acquired by silt deposit free from growths.

New canals in sandy loam to clay loam range from class above to one next below, and $n = 0.020$.

Medium to Large Canals.—The accepted value for *medium to large canals* in firm earth or gravelly loam with silty water, operated by organization that will give reasonable maintenance, is $n = 0.0225$.

Small ditches, easily influenced by slight roughness, and *larger canals poorly maintained*, $n = 0.025$.

Mountain Power Canals, with cobble bottoms but without finer materials for a graded bedding, $n = 0.028$.

DREDGED EARTH CANALS.—A dredged channel is rougher than one excavated by hand and teams. Likewise, a dipper dredge leaves a rougher bottom than does a drag line. Differences in value of n are largely brought about by adaptability of soil types and silt in the water to smooth over the original roughness. For best conditions $n = 0.0225$; average conditions, $n = 0.030$; worst conditions, without neglect of maintenance, $n = 0.040$.

CANALS EXCAVATED IN ROCK.—It is possible for excavation to be done in horizontally stratified rock, resulting in a very smooth bottom. Such canals, if very wide, will have a very low value of n as the rough sides have relatively little influence and, if the canal is of ordinary size and no attempt is made to smooth the sides, it is considered that the minimum value of $n = 0.020$ approx.

For usual or average conditions, with care in smoothing the rock cut by breaking off projections, $n = 0.033$. Under worst possible conditions, there is no limit, but it is seldom above $n = 0.045$. Silt and gravel deposits in rock canals may lower n by filling in the holes in the bottom.

NATURAL CHANNELS.—It is impossible to describe accurately the conditions of a natural channel that correspond to any given value of n. The best natural channels have a value of n seldom below $n = 0.025$. Average natural channels have a value of n probably in the neighborhood of $n = 0.030$. For the worst possible conditions, there is no limit. Judgment and experience are necessary to fix the value of n accurately for natural channels.

CONCRETE LININGS.—The value of n depends upon the specifications for the concrete surface and the workmanship. Considerable variation in n may result under the same specifications but with different workmen. It should be borne in mind that more favorable values may be attained in construction than should be anticipated in design. The following are the general characteristics:

Best possible, with neat cement, extremely well-troweled surface, $n = 0.010$. This value is seldom realized in practical construction.

The highest grade of practical concrete linings in best condition, with surface troweled as smooth as hand troweled sidewalks, and expansion joints perfectly smooth: Best, $n = 0.011$; probable, $n = 0.012$; worst, $n = 0.013$.

Surface as left by smooth jointed forms, or roughly troweled, and expansion joints fair. The probable value usually is adopted for concrete lining: Best, $n = 0.013$; probable, $n = 0.014$; worst, $n = 0.015$.

Concrete having prominent form marks, or previous types subject to deposits of stones on the bottom: Best, $n = 0.015$; probable, $n = 0.016$; worst, or probable maximum value not subject to rejection because of bad workmanship, $n = 0.018$. If liable to a growth of moss, the foregoing values should be increased by adding 0.002.

Values of n in excess of 0.017 indicate very poor concrete, which may be either badly spalled, owing to frost or to great difference in mixtures of concrete and finish coat, or broken down by the action of alkali. Similar values hold where the channel is losing its identity as a concrete surface, by deposits of sand and gravel or by moss or larval growths. Both moss and larvae appear to thrive in high velocities, even those of 30 or 40 ft. per sec. Covering a channel to exclude sunlight is effective in preventing moss growths and also would tend to prevent larval growth.

GUNNITE LININGS.—Concrete linings deposited by a cement gun, from the inside: Best, if scrubbed with wire brush, $n = 0.016$; average, if not scrubbed, $n = 0.019$; worst, for poor workmanship, $n = 0.021$.

Following a cement gun with a trowel will improve the surface from a capacity standpoint but may induce seepage loss by impairing the original density attained by the process. It is, however, recommended that the surface "rebound" be scrubbed off with a wire broom before it hardens and sticks to the canal bed.

MISCELLANEOUS MASONRY LININGS.—Glazed brickwork: Best, $n = 0.011$; probable, $n = 0.013$; worst, $n = 0.015$.

Brick in cement mortar: Best, $n = 0.012$; probable, $n = 0.015$; worst, $n = 0.017$.

Dressed ashlar surface: Best, $n = 0.013$; probable, $n = 0.015$; worst, $n = 0.017$.

For bench flume, consisting of natural rock surface for the uphill side, a smooth con-

crete retaining wall on the downhill side, and with a floor between, lined with concrete and clean, the uphill side being without projecting points, $n = 0.020$.

For the same construction, but with the floor covered with sand or gravel, or left as excavated without projections, uphill side with a few projecting points, such as obtained with careful excavation in hard rock, $n = 0.025$.

Cement-rubble surface: Best, $n = 0.017$; probable, $n = 0.025$; worst, $n = 0.030$.

Dry-rubble surface: Best, $n = 0.025$; probable, $n = 0.033$; worst, $n = 0.035$.

WOODEN BOX FLUMES.—The following applies to well-constructed flumes, carefully maintained, without projecting calking or other imperfections. Battens where used to be included in wetted perimeter.

Planed lumber, longitudinal boards sides and bottom: Best, $n = 0.011$; probable, $n = 0.014$; worst, after years of service, $n = 0.018$.

Unsurfaced lumber, longitudinal boards sides and bottom: Best, $n = 0.012$; probable, $n = 0.015$; worst, after years of service, $n = 0.018$.

Roofing paper lining varies with the type, generally from $n = 0.010$ to $n = 0.017$.

In general, the best and worst values correspond to new and very old flumes respectively, the latter with patches here and there in place of complete renewal of rotted members and corresponding to rather faulty work.

Wood-stave flumes. Creosoted: Best, $n = 0.011$; probable, $n = 0.012$; worst, $n = 0.014$; Untreated: Best, $n = 0.010$; probable, $n = 0.0115$; worst, $n = 0.014$.

SMOOTH INTERIOR STEEL FLUMES.—For smooth-interior flumes, as manufactured and erected under various trade names. When unpainted: Best, $n = 0.0105$; probable, $n = 0.012$; worst, $n = 0.014$. When painted: Best, $n = 0.012$; probable, $n = 0.013$; worst, $n = 0.017$.

The condition of these patent-joint flumes is largely a function of size and the number of carrying rods. Small flumes (2 to 5 ft. diam.) maintain their catenary shape quite well, if carrying rods are installed midway of each sheet, as well as at the ends, or if the side girders are set close to the sheets. Either method prevents excessive "scalloping." Large flumes require very frequent carrying rods and heavy gage metal.

ROUGH INTERIOR STEEL FLUMES.—Many of the older installations in the U. S. include steel flumes with projecting band joints or steel flumes of corrugated sheet metal. Both of these types are now obsolete and will not be discussed further. A fair value of n for this first type was 0.017, and for the second type 0.022.

Recommended Values of Friction Coefficients for Straight Closed Conduits

CAST-IRON PIPE.—On account of the growth of tubercles on the inside of the pipe, which decreases its area as well as increases its roughness, the value of C for a given age decreases as the diameter, but variation of C with age depends largely upon the composition of the water flowing in the pipe. Values are, therefore, rough approximations. Williams and Hazen recommend the average values given in Table 6. Cleaning old pipe increases the coefficient materially. Experimental values of $C = 150$ have been obtained for best new cast-iron pipe.

STEEL PIPE.—Scobey's equation for flow in steel pipe (U. S. Dept. of Agriculture, *Bul.* 150, 1930) 4 in. size and larger is

$$V = C_1 R^{0.58} S^{0.526} \qquad [21]$$

Values of $R^{0.58}$ and $S^{0.526}$ are given in Table 5.

Scobey divides steel pipe into the following classes, values of C_1 for which are given in Table 7:

Class 1. Full-riveted pipe, having both longitudinal and girth seams held by one or more lines of rivets with projecting heads. (Countersunk rivets belong in Class 3.) 1-*a*. Sheet metal up to 3/16 in. thick; 1-*b*. Plate metal from 3/16 in. to 7/16 in. thick, with either taper or cylinder joints; 1-*c*. Plate metal from 1/2 in. up, with either taper or cylin-

Table 6.—Average Value of C for Cast-iron Pipe

Diam. of Pipe, in.	Age in Years						
	0	5	10	20	30	40	50
4	130	118	107	89	75	64	55
8	130	119	109	93	83	73	65
12	130	120	111	96	86	77	70
16	130	120	112	98	87	80	72
24	130	120	113	100	89	81	74
30	130	120	113	100	90	83	76
36	130	120	113	100	90	83	76
40	130	120	113	100	90	83	77
60	130	120	113	100	90	83	77

der joints, and for plate from 1/4 in. to 7/16 in. thick when butt jointed; 1-*d*. Butt-strap pipe of plate from 1/2 in. up.

Class **2.** Girth-riveted pipe, having no retarding rivet heads in the longitudinal seams, but having the same girth seams as full-riveted pipe.

Class **3.** Continuous-interior pipe, having the interior surface unmarred by plate offsets or by projecting rivet heads in either longitudinal or girth seams. Not necessarily described as "smooth."

Recommended conservative values of C_1, based on experience with eastern waters are given in Table 7. Some engineers claim that western waters are less active. For waters known to be non-aggressive, Scobey recommends that the values of C_1 in Table 7 be multiplied by 1.0026^t, where t = age in years.

SMALL SMOOTH PIPE.—For smooth pipe of brass, lead, tin, glass and drawn copper use Williams and Hazen's coefficients: New and in good condition, $C = 140$; average, $C = 130$; bad, $C = 120$. The same values apply to new small wrought-iron and steel pipe. Falling off of C with age depends upon indeterminate conditions which are accentuated in small pipe. Therefore, an ample factor of safety should be used.

WOOD-STAVE PIPE.—Scobey's equation (U. S. Dept. of Agriculture *Bull.* 376) for flow in wood-stave pipe is

$$v = C_3 R^{0.65} S^{0.555} \qquad [21a]$$

Best information seems to indicate that C_3 does not vary materially with age for wood-stave pipe. Experimental values differ, ranging as follows: Best, $C_3 = 224$; probable average, $C_3 = 185$; worst, $C_3 = 170$.

UNLINED TUNNELS IN ROCK.—No experiments, based on other equations than that of Kutter, have been made. Unless more than usual information as to detail of the surface can be anticipated, use values of $n = 0.035$ to $n = 0.040$, depending on the importance of definite capacity, and compute with the theoretical neat section. In construction, overbreak will run from 10 to 30%, making a larger perimeter and greater area than the theoretical one but giving a correspondingly lower velocity. Two sets of tests disclosed values of n slightly less than 0.035, using a measured discharge but a theoretical neat-section hydraulic radius.

CONCRETE PIPES AND CONCRETE-LINED PRESSURE TUNNELS.—Scobey's equation (U. S. Dept. of Agriculture *Bull.* 852) for flow in concrete lined closed conduits is

$$v = C_2 R^{0.625} S^{0.5} \qquad [22]$$

Without appreciable error this equation may be written

$$v = C_2 R^{0.63} S^{0.5} \qquad [23]$$

Values of $R^{0.63}$ and $S^{0.5}$ are given in Table 5.

Recommended values of C_2 are as follows:

Concrete pipes, adequate for conveyance of city water supply are made in precast units from 8 to 20 ft. long in rigid oiled forms, or centrifugally spun in units 8 ft. long. Steel cylinders, with concrete interior and exterior, yield essentially the same surface. Many experimental data indicate that a value C_2 between 142 and 146 can be expected, but a value of 131 is recommended for design purposes. Large tunnels, concrete lined with oiled steel forms by means of a concrete gun, yield about the same surfaces. Ordinary form work in large bulk will yield results to conform to $C_2 = 121$ or better, depending on the care used in chipping down all offsets and fins remaining from construction. Wood form work now is but little used. Lack of rigidity causes the forms to squeeze under the pressure head of wet concrete. The resulting offsets, crack fins, cavities, etc., conform to a value of $C_2 = 128$ for the best, but may be $C_2 = 89$ for the worst. Concrete is a synthetic product, made by all degrees of workmanship, with all degrees of form excellence. Obviously the best results are obtained by experienced, careful organizations. Usually the difference in cost is more than made up in the relative capacity of the product. Data now available indicate that concrete pipes do not deteriorate in capacity like cast-iron or other metal pipes, by tuberculation. Certain waters may coat concrete and reduce capacity, or algae growths may accomplish this reduction. Reduction from such causes comes quickly or not at all.

Table 7.—Recommended Values of C_1 for Eastern Waters

Class of Pipe	Age in Years					
	0	10	20	30	40	50
3	154	142	131	121	112	104
2	149	138	127	118	109	100
1 *a*	140	130	120	111	103	95
1 *b*	131	120	111	103	95	88
1 *c*	125	115	106	98	90	84
1 *d*	120	110	102	94	87	80

3. MISCELLANEOUS DATA ON CONDUITS

PERMISSIBLE VELOCITIES IN CANALS.—Conditions of economy usually require as small a section and hence as high a velocity as the material will stand. Therefore, if the character of the bed is such as to require low velocities, sedimentation and plant growth are necessary evils, unless the alternative of a smaller lined section is adopted.

A mean velocity of 2 or 3 ft. per sec. generally will be sufficient to prevent the deposit of silt. Sand and gravel entering the canal will not be deposited with velocities somewhat smaller than the maximum given hereinafter to prevent scour of beds of like materials.

Plant growth has seriously affected the capacity of some canals. A temperature below 65° F., turbid or deep water, or a velocity greater than 2.5 ft. per sec. usually prevents serious growth.

The Special Committee on Irrigation Structures of the A.S.C.E. recommends the values given in Table 8 for permissible canal mean velocities to prevent scour. (Permissible Canal Velocities by Fortier and Scobey, *Trans.* A.S.C.E., vol. lxxxix, p. 940, 1926.) This table is for canals on tangents and for depths not exceeding 3 ft. For sinuous alignment reduce velocities 25%. For greater depths use velocities not exceeding 0.5 ft. per sec. greater. Column 3 recognizes that colloidal silts will precipitate and eventually form a plastic, highly cohesive mass, provided the canal is not fully loaded until seasoned. Column 4 recognizes that waters conveying abrasive sand or gravel will furnish a graded bedding and more resistance in some cases, but may assist scour in shales and clays.

AIR-BOUND PIPES.—A pipe is said to be air-bound when, in consequence of air being entrapped at the high points of vertical curves in the line, water will not flow out of the pipe, although the supply is higher than the outlet. The remedy is to provide cocks or valves at the high points, through which the air may be discharged.

WATER-HAMMER.—When selecting valves and fittings, the possibility of shock or strain due to water-hammer, in excess of the average working pressure of the line or system, should be considered. Many valves and fittings, installed where the working pressure under normal conditions would be low, have failed because of pressure due to water-hammer. This danger can be avoided by proper cushioning of the line by air chambers, or by relief valves.

When a valve in a pipe is closed while the water is flowing, the velocity of the water behind the valve is retarded and a dynamic pressure is produced. When the valve is closed quickly this dynamic pressure may be very great. It is then called "water-hammer" or "water-ram," and it causes in many cases fracture of the pipe. It is provided against by arrangements which prevent the rapid closing of the valve.

The excess pressure in feet of water produced by the instantaneous closure of a valve in a pipe is

$$h = av/g \qquad [24]$$

where a = velocity, ft. per sec. of propagation up the pipe; v = reduction of velocity, ft. per sec. The velocity of wave propagation is given by Joukowsky's equation

$$a = 4660/\sqrt{1 + KB} \qquad [25]$$

where K = ratio of the elastic moduli of water to the material of the pipe shell (0.01 for steel pipe); B = ratio of pipe diameter to shell thickness.

Table 8.—Permissible Canal Velocities

Original material excavated for canal	Velocity, ft. per sec., after Aging in Canals Carrying: Clear water, no detritus	Water transporting colloidal silts	Water transporting non-colloidal silts, sands, gravels, or rock fragments
Fine sand (non-colloidal)	1.50	2.50	1.50
Sandy loam "	1.75	2.50	2.00
Silt loam "	2.00	3.00	2.00
Alluvial silts when non-colloidal	2.00	3.50	2.00
Ordinary firm loam	2.50	3.50	2.25
Volcanic ash	2.50	3.50	2.00
Fine gravel	2.50	5.00	3.75
Stiff clay (very colloidal)	3.75	5.00	3.00
Graded, loam to cobbles, when non-colloidal	3.75	5.00	5.00
Alluvial silts when colloidal	3.75	5.00	3.00
Graded, silt to cobbles, when colloidal	4.00	5.50	5.00
Coarse gravel (non-colloidal)	4.00	6.00	6.50
Cobbles and shingles	5.00	5.50	6.50
Shales and hard-pan	6.00	6.00	5.00

Equation [24] applies to any time in seconds of valve closure less than

$$t = 2L/a \quad [26]$$

where L = length of pipe, ft.

The pressure produced by a closure in time greater than $2L/a$ is less than given by Equation [24], but its determination is quite complex. (See Hydro-Electric Handbook, by Creager and Justin, John Wiley & Sons, 1927.)

4. FLOW OVER DAMS

The basic theoretical expression for flow over weirs is given in Equation [5], which, if all constants are combined, may be written

$$Q = Clh^{3/2} \quad [27]$$

where Q = total discharge, cu. ft. per sec.; C = coefficient of discharge, which depends on the shape of the crest and the head on the crest; l = net or effective length of crest, ft., *i.e.*, the total length of crest corrected for end contractions due to piers and sharp-cornered abutments; h = actual or measured head on the crest, ft., taken at a point sufficiently remote from the dam to avoid the surface curve.

Francis has determined, that, to allow for the effect of the velocity of approach, this equation should be written

$$Q = Cl(h + h_v)^{3/2} - h_v^{3/2} \quad [28]$$

where h_v = head corresponding to velocity of approach. An approximate form of Francis's equation is

$$Q = Cl(h + h_v)^{3/2} \quad [29]$$

Equation [29] gives values of Q in excess of that from Equation [28]. The error for a depth of channel approach greater than twice the head on the crest is less than 1%.

Francis's equation for the necessary correction due to complete sharp-cornered end contractions is

$$l = l_t - 0.1nh \quad [30]$$

where l_t = total or gross length of crest between abutments and piers; n = number of complete contractions.

If the crest is obstructed by wide rectangular piers, n represents the number of corners that deflect the water, there being two for each pier and one for each abutment. However, if the piers are very thin or pointed upstream, or if the abutments are well rounded or continuous upstream, the effective reduction in crest will be much less.

For weirs of short length, the contraction can never be greater than about $(1.0 - \sqrt{0.62})/2 = 0.106$ times the clear distance between two piers or abutments, as this corresponds to the side contraction through sharp-cornered orifices having a contraction coefficient of 0.62. Therefore, the contraction at each corner is (0.106 × distance between piers) for all cases where the depth is greater than (1.06 × distance between piers or abutments).

The value of the discharge coefficient C for use in Equations [28] and [29] has been determined experimentally for spillways of many different types. These experiments have been carefully tabulated in Weir Experiments, Coefficients and Formulas, by R. E. Horton (U. S. Geol. Survey, Water Supply Paper 200, 1907). Coefficients adaptable to rounded dam crests, scientifically designed to fit the bottom contraction is given in Hydro-Electric Handbook by Creager and Justin (John Wiley & Sons, 1927). For sharp-crested weirs, as in Fig. 11, the coefficient C is approximately 3.33.

SUBMERGED SPILLWAYS.—If the crest of the spillway is submerged, the discharge coefficients for use in Equations [28] and [29] should be modified according to the degree of submergence, as indicated in Table 9. In this table C is the coefficient for *free* discharge over a similar crest under the same head, and C' is the modified coefficient due to submergence. The head h is the aforementioned head on the dam, and h_s is the corresponding superelevation of tail water above the crest.

Table 9.—Relative Coefficients, Submerged Crest and Free Crest

h_s/h =	0.0	0.1	0.2	0.3	0.4	0.5	0.6	0.7	0.8	0.9	1.0
C'/C =	1.000	0.991	0.983	0.972	0.956	0.937	0.907	0.856	0.778	0.621	0.000

5. MEASUREMENT OF FLOWING WATER *

MEASURING WEIRS.—The measuring weir consists of a dam which may extend the full width of the channel or which may have a crest consisting of a notch, rectangular or otherwise, cut in the dam, in which case it is called a weir with end contractions.

* See also A.S.M.E. Test Code for Hydraulic Power Plants and Their Equipment.

RECTANGULAR WEIRS.—Fig. 11 shows a rectangular weir. The general formula for discharge of water over a rectangular weir is

$$Q = Clh^{3/2} \qquad [31]$$

where Q = discharge, cu. ft. per sec.; l = effective length of crest, ft.; h = head on weir, ft., measured by hook gage or stilling boxes at a distance from the crest between $5h$ and $10h$; C = a coefficient depending upon the type of crest, the head on the weir and the depth of the channel of approach.

Where l_t = total length of crest, ft., the usual expression for the effective length of crest, with two complete sharp-edged end contractions is from Equation [30]

$$l = l_t - 0.2h \qquad [32]$$

Experiments on many different types of crests have been made, which indicate that C varies from about 2.6 for flat-crested dams to about 4.0 for rounded crests scientifically designed to fit the bottom contraction. (See Hydro-Electric Handbook by Creager and Justin, John Wiley & Sons, 1927.) These experiments, with the resulting values of C, are summarized in Weir Experiments, Coefficients and Formulas by R. E. Horton (U. S. Geol. Survey, Water Supply Paper 200, 1907). Measurement of discharge over existing dams of similar types of crest can be made by comparison with these experiments within only a fair degree of accuracy.

Unless duplicating exactly a given set of accurate experiments, measuring weirs, constructed for that purpose, should have the following characteristics: 1. Straight uniform channel with uniform velocity, provided by stilling racks if required; 2. Suppressed end contractions, provided by making sides of channel form the end of the weir. This results in $l = l_t$ in Equation [32]. 3. Weir to have free overfall with complete aeration of the nappe. Aeration usually requires air passages leading to the space between the nappe and the dam; 4. A metal crest free from rust, with sharp right-angle corner on the upstream edge, a crest width of 1/8 in. and beveled to an angle of 45° on the downstream face.

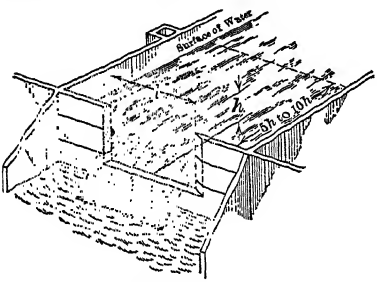


Fig. 11. Weir and End Contractions

A number of equations, based on experiments with sharp-crested rectangular weirs of the foregoing specifications, have been proposed. The following apply to such weirs with suppressed end contractions:

Francis Weir Equation.—The best known equation is that of Francis (Lowell Hydraulic Experiments, by J. B. Francis), which is

$$Q = 3.31l\left[(h + h_v)^{3/2} - h_v^{3/2}\right] \qquad [33]$$

where h_v is the head due to the velocity of approach. With no velocity of approach, Equation [33] reduces to $Q = 3.31lh^{3/2}$. [34]

Francis's equations are subject to considerable error, particularly for high velocities of approach and small values of h. The equation should not be used for accurate weir measurements and is given here only because Equation [34] is a convenient one to remember for rough approximations.

Fteley and Stearns Weir Equation.—(Description of Some Experiments on the Flow of Water, by Fteley and Stearns, *Trans.* A.S.C.E., xii, p. 1, 1883.) This equation is:

$$Q = 3.31\,l\,(h + 1.5h_v)^{3/2} + 0.007l. \qquad [35]$$

Equation [35] is the result of a study of their own experiments and those of Francis. The range of conditions covered by these experiments is given in Table 10. Fteley and Stearns's experiments indicate discharges under low heads about 3% less than those of Bazin.

Table 10.—Range of Conditions in Fteley and Stearns' Experiments

	Francis	Fteley and Stearns	
		With End Contractions	Without End Contractions
Limit of head	0.6 to 1.6 ft.	1.0 ft.	0.83 and 1.63 ft.
Height of weir	2.0 and 5.0 ft.	3.56 ft.	3.17 and 6.55 ft.
Length of weir	8.0 and 10.0 ft.	2.30 to 4.00 ft.	5.00 and 19.00 ft.
Width of channel approach	10.0 and 14.0 ft.	....	....
Maximum velocity of approach	0.2 to 1.0 ft. per sec.	0.54 ft. per sec.	0.6 and 0.8 ft. per sec

Bazin Weir Equation.—(Translation by Marichal and Trautwine, *Proc.*, Engrs. Club of Philadelphia, 1890.) Bazin derived his equation from his own experiments. It is written for weirs without end contractions as

$$Q = \left(0.405 + \frac{0.00984}{h}\right)\left(1 + 0.55\frac{h^2}{D^2}\right) lh\sqrt{2gh}. \quad . \quad . \quad . \quad . \quad . \quad [36]$$

where D is the depth, ft., of water upstream from the weir.

The experiments covered the following range of conditions: Range of head 0.3 to 1.78 ft.; height of weir 1.16 to 3.72 ft.; length of weir 1.64 to 6.56 ft.

King Weir Equation.—King (Handbook of Hydraulics, by H. W. King, McGraw-Hill Book Co., 1929) proposes the following equation, which, after considerable study, he finds to agree more closely with various experimental data than any of the others:

$$Q = 3.34lh^{1.47}\{1 + (0.56h^2/D^2)\}. \quad . \quad . \quad . \quad . \quad . \quad . \quad . \quad [37]$$

Rehbock's Weir Equation.—Perhaps the most accurate of all weir equation is that of Rehbock. (See discussion of Precise Weir Measurements, by Schoder and Turner, *Trans.* A.S.C.E., vol. xciii, 1929.) If P = height of weir, ft., this equation is

. . . . . . [38]

CHOICE OF TYPE AND METHOD.—As no equations have been derived which will agree exactly with experimental data, precise measurements should be made under conditions identical with those under which some one set of original experiments were made. Bazin's experiments are by far the most complete, but they have not been proved to be more accurate than those of Fteley and Stearns. As the latter indicate discharges for low heads somewhat less than those of Bazin, they have been used extensively for measurements where an error, if any, would result in a determined flow less than the actual.

On account of the disagreement between the various proposed equations, the preliminary draft of the A.S.M.E. Test Code for Hydraulic Power Plants and Their Equipment specifies the use of Equation [31] for sharp-crested rectangular weirs without end contractions. Coefficient C is taken from Table 11, which gives values including the correction for the velocity of approach.

The coefficients in Table 11 are the average of values computed by the equations of Fteley and Stearns, of Bazin, and of Rehbock.

TRIANGULAR NOTCH WEIRS.—Triangular or V-notch weirs, in which the apex is down, are adaptable to small discharges. In this type of weir, the head for extremely small discharges is proportionally greater due to the reduction of crest length near the apex. This results in greater accuracy.

From his own experiments and those of Barr, King (Handbook of Hydraulics, Second edition, p. 93) gives the following equations for discharge: For a sharp-edged right-angle notch cut in a large sheet of commercial steel plate, $Q = 2.52\,h^{2.47}$; . . . . [39]
For a similar notch cut in a polished brass plate, $Q = 2.48\,h^{2.48}$. . . . . [40]

D. R. Yarnall (*Mech. Engg.*, Jan., 1927) gives the following coefficients for the equation

$$Q = Ch^{5/2} \quad . \quad . \quad . \quad . \quad . \quad . \quad . \quad . \quad . \quad . \quad . \quad . \quad [41]$$

Head, h	0.4	0.6	0.8	1.0	1.2
Coefficient, C	2.511	2.492	2.484	2.481	2.480

These coefficients apply to a right-angle notch cut in a smooth brass plate, with edges very carefully finished but not highly polished. An accuracy within 1/2 of 1 percent is claimed.

THE CIPPOLETI, OR TRAPEZOIDAL WEIR.—Cippoleti found that by using a trapezoidal weir with the sides inclined 1 horizontal to 4 vertical, with end contraction, the discharge is equal to that of a rectangular weir without end contraction (that is with the width of the weir equal to the width of the channel) and is represented by the simple formula $Q = 3.367\,lh^{3/2}$. A. D. Flinn and C. W. D. Dyer (*Trans.* A.S.C.E., 1894), in

Table 11.—Test Code Values of C for Various Heads and Heights of Crest

Head, h	Height of Crest, P										
	4	5	6	7	8	9	10	12	14	16	20
1.0	3.376	3.356	3.344	3.335	3.329	3.325	3.322	3.317	3.314	3.311	3.308
1.2	3.391	3.366	3.350	3.339	3.332	3.326	3.322	3.316	3.311	3.308	3.305
1.4	3.409	3.378	3.359	3.346	3.336	3.330	3.324	3.316	3.311	3.307	3.303
1.6	3.429	3.392	3.370	3.354	3.343	3.334	3.328	3.319	3.312	3.308	3.302
1.8	3.450	3.408	3.382	3.363	3.350	3.340	3.333	3.322	3.315	3.309	3.303
2.0	......	3.425	3.394	3.373	3.358	3.347	3.338	3.325	3.317	3.311	3.304

experiments with a trapezoidal weir, with values of l from 3 to 9 ft. and of h from 0.24 to 1.40 ft. found the value of the coefficient to average 3.334, the water being measured by a rectangular weir and the results being computed by Francis's formula, and 3.354 when Smith's formula was used. They conclude that Cippoleti's formula when applied to a properly constructed trapezoidal weir will give the discharge with an error due to combined inaccuracies, not greater than 1%.

CURRENT METERS.—The current meter is a mechanism consisting of an arrangement of cups or vanes, on vertical or horizontal axes, respectively, held stationary in a stream of flowing water. The vanes are revolved by the motion of the water. The number of revolutions in a given time is proportional to the velocity of flow. The number of revolutions is usually made known to the observer by the making and breaking of an electric circuit as indicated by a buzzer or on a recording device, or by sound conveyed through a tube used to hold the meter. Current meters are rated by drawing them through still water at several velocities. The meter should be rated before and after each test.

Current meters are not extremely accurate means of testing, on account of the vertical and horizontal components of flow in streams caused by eddies. Some meters are retarded and others accelerated by turbulent flow. The amount can be gaged by holding the meter obliquely during rating. For greater accuracy, two types of meters should be used which have opposite characteristics due to turbulence, and the weighted average of the readings used.

The section of the stream used for the test should be as uniform in area and as smooth as possible for some distance up and down stream. Observations should be made at a sufficient number of points in the gaging cross-section to accurately determine the average velocity, the spacing of observations being ordinarily closer towards the sides, top and bottom, where the change in velocities is greatest. Vertical and horizontal velocity curves should be plotted from which to calculate the mean velocity.

For a discussion of different types of meters, see Effect of Turbulence on the Registration of Current Meters, by Yarnell and Nagler, *Trans.* A.S.C.E., vol. xcv, 1931, p. 766.

FLOAT MEASUREMENTS.—The velocity of the stream can be found by laying off 100 ft. of the bank and throwing a float into the middle, noting the time taken in passing over the 100 ft. By doing this a number of times and taking the average, the velocity at the surface is determined by dividing the average by the distance. As the top of a stream flows faster than the bottom or sides, the average velocity being about 83% of the surface velocity at the middle, it is convenient to measure a distance of 120 ft. for the float and reckon it as 100.

PITOT TUBES.—The Pitot tube is used for measuring the velocity of flowing fluids and gases. Its essential feature is a thin-edged orifice placed at right angles to the flow at the end of a tube. The impact of the fluid causes an excess pressure in the tube equal to the velocity head. In general, the Pitot tube consists of a tube passing through a stuffing box in the shell of the pipe, the end of which is bent at right angles upstream and tapered to form the orifice. The orifice is made to traverse the pipe and at least two diameters mutually perpendicular should be traversed. The pressure in the tube is

$$H = h_p + h_v = h_p + (C^2 v^2/2g), \quad . \quad . \quad . \quad . \quad . \quad . \quad . \quad [42]$$

where h_p and h_v = the pressure and velocity heads respectively, and C = a constant.

Since the pressure in the tube is the pressure head plus the velocity head, the Pitot tube reading must be compared with the average of at least four piezometers placed around the pipe to measure the pressure head. The piezometers consist of tubes connected to $^1/_8$- or $^1/_4$-in. openings flush with the inside wall of the pipe. The wall of the pipe near the piezometers should be particularly smooth and should be free from burrs.

According to Allen and Hooper (Piezometer Investigation, *Trans.* A.S.M.E., Hyd. 54–1, 1932) square-edge piezometer openings are very sensitive to slight defects, and they recommend $^1/_8$- to $^1/_4$-in. holes with inner edges rounded to $^1/_{32}$ to $^1/_{16}$ in. radius respectively. The tube connected to the piezometer will read exactly the pressure head h_p.

The Pitot and piezometer tubes usually are joined through a manometer in which the differential head is directly measured. The differential head is, from Equation [41],

$$h_v = H - h_p = C^2v^2/2g, \text{ from which } v = C\sqrt{2gh}\,, \quad . \quad . \quad . \quad . \quad [43]$$

where C = about 0.98, but should be determined experimentally.

Pitot tubes are not extremely accurate unless the flow in the pipe is extremely smooth and straight, because whorls and eddies greatly affect the readings.

Many forms of tubes are used. In the pitometer, the Pitot tube and the piezometer tube encased in a larger tube, pass through the stuffing box and traverse the pipe diameter simultaneously.

THE VENTURI METER.—The Venturi meter, as shown in Fig. 12 consists of a contraction in a pipe or other closed conduit for the purpose of accelerating the fluid and lowering its static pressure. Piezometers are placed at A, the upstream end of the contraction and at B, the lower end. The equation for the discharge past the meter is

. . . . [44]

where Q = discharge, cu. ft. per sec.; A = area at A, sq. ft.; a = area at B, sq. ft.; h = difference between the pressure heads at B and A, as shown by the piezometers; C = a coefficient which varies between 0.97 and 1.0.

From data in Part 1 of the A.S.M.E. Fluid Meters Report, 1924, the author has devised the following empirical formula for the value of C for water:

$$0.051 \qquad [45]$$

where S = velocity at the throat, ft. per sec.; t = temperature of the water deg. F.; d = diameter at the throat, in.

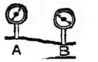


FIG. 12. Venturi Meter

King (Handbook of Hydraulics, 2nd edition, p. 381) claims that more recent experiments by Ledoux (Venturi Tube Characteristics, *Trans.* A.S.C.E., lxi, 1927) indicate values of the coefficients to be about 2% smaller for throat velocities of 5 ft. per sec., and about 1% smaller for the higher velocities. However, the exact determination of the coefficient can be obtained only by test.

Equation [45] applies for meters having the following usual dimensions, where D is the normal diameter of the pipe: Diameter of the throat at B from $D/4$ to $D/2$; length of the throat at B from $D/4$ to $D/2$; entrance cone to have a total angle of about 21°; exit cone to have a total angle of about 5° to 7°; throat to be accurately machined to exact diameter; diameter at A to be smooth and to accurate dimension; angles in entrance cone at B to be rounded off to an easy tangential curve; length of straight pipe before meter at least $5D$, and preferably more.

The tubes from the piezometers A and B are joined through a manometer and the differential pressure h_v read directly. Special attachments may be obtained for indicating, recording and integrating the flow.

The loss of head in passing through the meter can be calculated from Equation [14], practically the entire loss being confined to the enlargement below the throat.

There is no limit to the sizes of the meters nor the quantities of water that may be measured. Three Venturi tubes, approximately 17 ft. diameter, are installed in the Catskill Aqueduct supply to New York. While the Venturi meter originally was applied to the measurement of water it has since been used extensively for the measurement of sewage, gases, steam and many other fluids.

Any sudden reduction in pipe diameter may be used as a Venturi meter if properly calibrated by other means of measuring the flow. Examples of such uses are at the reduction of penstock diameter at entrance to hydraulic turbines and in the scroll cases of such turbines for permanent meters for recording changes in discharge. See Mechanical Features Affecting Hydro Plants by E. A. Dow, *Mech. Engg.*, Oct. 1925.

Many other modifications of the Venturi meter are in use, including contractions in open conduits. (See The Improved Venturi Flume, by R. L. Parshall, *Trans.* A.S.C.E., 1926, p. 840.)

Thin plate or sharp-edged orifices in pipes used for measuring flow are merely a rounded hole in a flat diaphragm clamped between the pipe flanges at a joint in the pipe line, with the hole concentric with the pipe. The theory of measurement is the same as that of the Venturi meter.

MEASUREMENT OF DISCHARGE OF PUMPING-ENGINES BY MEANS OF NOZZLES.—(*Trans.* A.S.M.E., xii, 575)—The measurement of water by computation from its discharge through orifices, or through the nozzles of fire-hose, furnishes a means of determining the quantity of water delivered by a pumping-engine which can be applied without much difficulty. John R. Freeman (*Trans.* A.S.C.E., Nov., 1889), describes a series of experiments covering a wide range of pressures and sizes, and the results showed that the coefficient of discharge for a smooth nozzle of ordinary good form was within one-half of 1%, either way, of 0.977; the diameter of the nozzle being accurately calipered, and the pressures being determined by means of an accurate gage attached to a suitable piezometer at the base of the play pipe.

In order to use this method for determining the quantity of water discharged by a pumping-engine, it would be necessary to provide a pressure-box, to which the water would be conducted, and attach to the box as many nozzles as would be required to carry

off the water. According to Mr. Freeman's estimate, four 1¼-in. nozzles, thus connected, with a pressure of 80 lb. per sq. in., would discharge the full capacity of a 2,500,000-gal. engine. He also suggests the use of a portable apparatus with a single opening for discharge, consisting essentially of a Siamese nozzle, so-called, the water being carried to it by three or more lines of fire-hose. Table 14 gives the discharge of fire-hose nozzles.

SALT VELOCITY METHOD OF MEASURING FLOW.—The salt velocity method of measurement is based on the fact that salt in solution increases the electrical conductivity of water. Brine (salt solution) is injected through a system of piping and pop valves at any point in the conduit, usually at the upper end. The introduction of this brine and its passage past one or more pairs of electrodes at other points in the conduit, together with the elapsed time between points, are recorded graphically. The electrodes are so designed as to give equal weight electrically to unit areas of the pipe cross-section. The discharge $Q = V/t$ cu. ft. per sec., where V = volume of the conduit test section, cu. ft., t = time of passage of the salt solution, sec. This method, in common with all other methods of water measurement, should be used only by trained and experienced men. This salt velocity method was developed by C. M. Allen, Professor of Hydraulic Engineering, Worcester Polytechnic Institute. For a complete description of the method, see paper by Allen and Taylor, *Trans.* A.S.M.E., vol. xlv, 1923. It has been used extensively in measuring water in both closed and open conduits, and particularly in connection with field efficiency tests of water-wheels, and has been developed and improved so that it is now accepted as one of the standard methods.

THE GIBSON METHOD OF MEASURING FLOW.—The Gibson method is based on the equation of impulse and momentum applied to an enclosed column of water in motion. It is applicable in testing hydraulic power plants where the turbine is supplied with water through a closed conduit and means are available, such as turbine gates, for interrupting the flow of water. To apply the method it is necessary to measure the physical dimensions of the conduit and to obtain pressure-time diagrams, which show the changes of pressure with respect to time that occur in the conduit during and after the closing of the turbine gates. There are two kinds of such diagrams: *a. Simple Diagrams*, in which the changes of pressure at one point in the conduit are recorded; *b. Differential Diagrams*, in which the difference between the changes of pressure at two points in the conduit are recorded.

The length of conduit upstream from the point of measurement for simple diagrams or the length between the two points of measurement for differential diagrams, should, preferably, be not less than 25 ft. The conduit measurements should be made as precisely as possible. The method may be used whether the conduit is of uniform or variable cross-sectional area but, for simplicity, a section of greatest regularity and suitable length should be selected. For complete description of the Gibson method, see *Trans.* A.S.M.E., vol. xlv, p. 343, 1923. This method is applicable only to closed conduits.

SALT TITRATION METHOD OF MEASURING FLOW.—In cases where the flow is too turbulent for other methods, the salt titration method has been used. In this salt in solution is introduced at the inlet at a uniform known rate and its concentration measured at the outlet. See Salt Solution Test of Holtwood Plant, *Eng. Record*, Mar. 20, 1915, p. 358.

6. FIRE-STREAMS

FIRE-STREAM TABLES.—Table 12 is condensed from one contained in the pamphlet of Fire-Stream Tables of the Associated Factory Mutual Fire Ins. Cos., based on the experiments of John R. Freeman, *Trans.* A.S.C.E., 1889.

The pressure in the first column is that indicated by a gage attached at the base of the play pipe and set level with the end of the nozzle. The vertical and horizontal distances, in columns 2 and 3 are those of effective fire-streams with moderate wind. The maximum limit of a "fair stream" is about 10% greater for a vertical stream; 12% greater for a horizontal stream. In still air much greater distances are reached by the extreme drops. The pressures given are for the best quality of rubber-lined hose, smooth inside. The hose friction varies greatly in different kinds of hose, according to smoothness of inside surface, and pressures as much as 50% greater are required for the same delivery in long lengths of inferior rubber-lined or linen hose. The pressures at the hydrant are those while the stream is flowing, and are those required with smooth nozzles. Ring nozzles require greater pressures. With the same pressures at the base of the play pipe, the discharge of a ¾-in. smooth nozzle is the same as that of a ⅞-in. ring nozzle; of a ⅞-in. smooth nozzle, the same as that of a 1-in. ring nozzle.

The figures for hydrant pressure in the body of the table are derived by adding to the nozzle or play-pipe pressure the friction loss in the hose, and also the friction loss of a

Chapman 4-way independent-gate hydrant, ranging from 0.86 lb. for 200 gal per min. flowing to 2.31 lb. for 600 gal.

The following notes are taken from the pamphlet referred to: The discharge as stated in Ellis's tables and in their numerous copies in trade catalogs is from 15 to 20% in error.

Table 12.—Hydrant Pressures Required with Different Sizes and Lengths of Hose

(J. R. Freeman, *Trans.* A.S.C.E., 1889)

Nozzle	Play-pipe Pressure, lb.	Fire-Stream Distance, ft. Vert.	Fire-Stream Distance, ft. Hor.	Gal. per min.	Length of Hose, ft. 50	100	200	300	400	500	600	800	1000
3/4-inch Smooth Nozzle	10	17	19	52	10	11	11	12	13	13	14	15	16
	20	33	29	73	21	22	23	24	25	26	28	30	32
	30	48	37	90	31	32	34	36	38	40	41	45	49
	40	60	44	104	42	43	46	48	50	53	55	60	65
	50	67	50	116	52	54	57	60	63	66	69	75	81
	60	72	54	127	63	65	68	72	76	79	83	90	97
	70	76	58	137	73	75	80	84	88	92	97	105	114
	80	79	62	147	84	86	91	96	101	106	111	120	130
	90	81	65	156	94	97	102	108	113	119	124	135	146
	100	83	68	164	105	108	114	120	126	132	138	150	163
7/8-inch Smooth Nozzle	10	18	21	71	11	11	13	14	15	16	17	19	22
	20	34	33	100	22	23	25	27	30	32	34	39	43
	30	49	42	123	33	34	38	41	45	48	51	58	65
	40	62	49	142	43	46	50	55	59	64	68	78	87
	50	71	55	159	54	57	63	69	74	80	86	97	108
	60	77	61	174	65	69	75	82	89	96	103	116	130
	70	81	66	188	76	80	88	96	104	112	120	136	152
	80	85	70	201	87	91	101	110	119	128	137	155	173
	90	88	74	213	98	103	113	123	134	144	154	174	195
	100	90	76	224	109	114	126	137	148	160	171	194	216
1-inch Smooth Nozzle	10	18	21	93	12	12	14	16	18	20	22	26	30
	20	35	37	132	23	25	29	33	37	41	45	52	60
	30	51	47	161	34	37	43	49	55	61	67	79	90
	40	64	55	186	46	50	58	66	73	81	89	105	120
	50	73	61	208	57	62	72	82	92	102	111	131	151
	60	79	67	228	69	75	87	98	110	122	134	157	181
	70	85	72	246	80	87	101	115	128	142	156	183	211
	80	89	76	263	92	100	115	131	147	162	178	209	241
	90	92	80	279	103	112	130	147	165	183	200	236	.....
	100	96	83	295	115	125	144	164	183	203	223	.....	.....
1 1/8-inch Smooth Nozzle	10	18	22	119	12	14	17	20	24	27	30	36	43
	20	36	38	168	25	28	34	41	47	54	60	73	85
	30	52	50	206	37	42	52	61	71	80	90	109	128
	40	65	59	238	50	56	69	81	94	107	120	145	171
	50	75	66	266	62	70	86	102	118	134	150	181	213
	60	83	72	291	74	84	103	122	141	160	180	218	256
	70	88	77	314	87	98	120	143	165	187	209	254	.....
	80	92	81	336	99	112	138	163	188	214	239	.....	.....
	90	96	85	356	112	126	155	183	212	241	.....	.....	.....
	100	99	89	376	124	140	172	204	236	.....	.....	.....	.....
1 1/4-inch Smooth Nozzle	10	19	22	148	14	16	21	26	31	36	41	51	61
	20	37	40	209	27	32	42	52	62	72	82	101	121
	30	53	54	256	41	49	63	78	93	108	123	152	182
	40	67	63	296	55	65	84	104	124	144	164	203	243
	50	77	70	331	68	81	106	130	155	180	204	254	.....
	60	85	76	363	82	97	127	156	186	216	245	.....	.....
	70	91	81	392	96	113	148	182	217	252	.....	.....	.....
	80	95	85	419	110	129	169	208	248	.....	.....	.....	.....
	90	99	90	444	123	145	190	234	.....	.....	.....	.....	.....
	100	101	93	468	137	162	211	261	.....	.....	.....	.....	.....
1 3/8-inch Smooth Nozzle	10	20	23	182	16	19	27	34	42	49	56	71	86
	20	38	42	257	31	39	53	68	83	98	113	143	173
	30	55	56	315	47	58	80	103	125	147	169	214	259
	40	69	66	363	62	77	107	137	166	196	226	.....	.....
	50	79	73	406	78	96	134	171	208	245	.....	.....	.....
	60	87	79	445	93	116	160	205	250	.....	.....	.....	.....
	70	92	84	480	109	135	187	239	.....	.....	.....	.....	.....
	80	97	88	514	124	154	214	.....	.....	.....	.....	.....	.....
	90	100	92	545	140	173	240	.....	.....	.....	.....	.....	.....
	100	103	96	574	156	193	.....	.....	.....	.....	.....	.....	.....

In the best rubber-lined hose, $2\,^1/_2$-in. diam., the loss of head due to friction, for a discharge of 240 gal. per minute, is 14.1 lb. per 100 ft. length; in inferior rubber-lined mill hose, 25.5 lb., and in unlined linen hose, 33.2 lb.

Less than a $1\,^1/_8$-in. smooth-nozzle stream with 40 lb. pressure at the base of the play pipe, discharging about 240 gal. per min., cannot be called a first-class stream for a factory fire. 80 lb. per sq. in. is considered the best hydrant pressure for general use; 100 lb. should not be exceeded, except for very high buildings, or lengths of hose over 300 ft. Table 13 is computed on the basis of 14 lb. per 100 ft. of $2\,^1/_2$-in. hose with

Table 13.—Friction Loss in Rubber-lined Cotton Hose with Smoothest Lining

Diam. of Hose, in.	Gallons per Minute Flowing									Velocity, ft. per sec.	Velocity Head, $V^2 \div 2g$	
	100	200	300	400	500	600	700	800	1000		ft.	lb.
	Friction Loss, Pounds per 100 ft. Length											
2	6.836	27.3	61.5	109.	171.	......	......	......	......	5	0.39	0.17
2 1/8	5.170	20.7	46.5	82.7	129.	189.	......	......	......	10	1.6	0.69
2 1/4	3.790	15.2	34.1	60.6	94.7	136.	186.	......	......	15	3.5	1.5
2 3/8	2.895	11.6	26.1	46.3	72.4	104.	138.	185.	......	20	6.2	2.7
2 1/2	2.240	9.0	20.2	35.8	56.0	80.6	110.	143.	224.	25	9.7	4.2
2 5/8	1.748	7.0	15.7	28.0	43.7	62.9	85.7	112.	175.	30	14.0	6.1
2 3/4	1.391	5.6	12.5	22.3	34.8	50.1	68.2	89.0	139.	35	19.0	8.2
2 7/8	1.097	4.4	9.9	17.6	27.4	39.5	53.8	70.2	110.	40	24.8	10.7
3	0.900	3.6	8.1	14.4	22.5	32.4	44.1	57.6	90.	45	31.4	13.6
3 1/2	0.416	1.7	3.7	6.7	10.4	15.0	20.4	26.6	41.6	50	38.8	16.7
4	0.214	0.9	1.9	3.4	5.4	7.7	10.5	13.7	21.4	......	......	......

Table 14.—Discharge of Nozzles Attached to 50 Ft. of Fire-hose

Quantities are in Gallons per Minute

Hydrant Pressure	Diameter of Smooth Nozzle, in.								Ring Nozzle		
	1 3/4	1 1/2	1 3/8	1 1/4	1 1/8	1	7/8	3/4	1 3/8	1 1/4	1 1/8
10 lb.	193	163	146	127	107	87	68	51	118	101	84
20 "	274	232	206	179	151	123	96	72	167	143	119
30 "	335	283	251	219	184	150	118	88	205	175	145
40 "	387	327	291	253	213	173	136	101	237	202	168
50 "	432	366	325	283	238	194	152	113	264	226	188
60 "	473	400	357	309	261	213	167	124	289	247	205
70 "	510	432	385	334	281	230	180	134	313	267	222
80 "	546	461	412	357	301	246	192	144	334	285	237
90 "	579	490	437	379	319	261	204	152	355	303	252
100 "	610	515	461	400	337	275	215	161	374	319	266

Table 15.—Pipe Sizes for Fire Streams

(Cast Iron Pipe Handbook, 1932)

Flow given in cu. ft. per min. Figures are based on $1\,^1/_8$-in. smooth bore nozzles, playing simultaneously, and attached to 200 ft. of best quality rubber lined hose; pressures measured at hose connections. Velocity of water in pipe, approximately 3 ft. per sec. To convert to gallons, multiply figures in table by 7.4805.

No. of 1 1/8-in. Hose Nozzles	Pressure, lb. per sq. in.																	
	60		80		90		100		110		120		130		140		150	
	Pipe, in.	Flow	Pipe, in.	Flow	Pipe, in.	Flow	Pipe, in.	Flow	Pipe, in.	Flow	Pipe, in.	Flow	Pipe, in.	Flow	Pipe, in.	Flow	Pipe, in.	Flow
1	6	25	6	29	6	30	6	32	6	33	6	35	6	36	6	38	6	39
2	8	50	8	57	8	61	8	64	8	67	10	70	10	73	10	75	10	78
3	10	74	10	86	10	91	10	96	10	100	10	105	10	109	12	113	12	117
4	10	99	12	114	12	121	12	128	12	134	12	140	12	145	12	151	14	156
5	12	124	12	143	12	152	14	160	14	167	14	174	14	181	14	188	14	195
6	12	149	14	172	14	182	14	191	14	200	14	209	16	218	16	226	16	234
7	14	174	14	200	16	212	16	223	16	234	16	244	16	254	16	264	18	273
8	14	199	16	229	16	242	16	255	16	267	18	279	18	290	18	301	18	313
9	16	223	16	257	18	273	18	287	18	301	18	314	18	326	18	339	20	352
10	16	248	18	286	18	303	18	319	18	334	20	349	20	363	20	377	20	391
11	18	273	18	314	18	333	20	351	20	367	20	384	20	399	20	414	20	430
12	18	298	20	343	20	364	20	383	20	401	20	419	20	435	24	452	24	469
13	18	323	20	372	20	394	20	415	20	424	24	454	24	472	24	490	24	508
14	20	348	20	400	20	424	24	447	24	467	24	488	24	508	24	528	24	547
15	20	372	20	429	24	455	24	479	24	501	24	523	24	544	24	565	24	586

250 gal. per min. flowing, as found in Freeman's tests, assuming that the loss varies as the square of the quantity, and for different diameters and the same quantity inversely as the 5th power of the diameter.

7. THE SIPHON

The siphon is a bent tube of unequal branches, open at both ends, and is used to convey a liquid from a higher to a lower level, over an intermediate point higher than either. Its parallel branches being in a vertical plane and plunged into two bodies of liquid whose upper surfaces are at different levels, the fluid will stand at the same level both within and without each branch of the tube when a vent or small opening is made at the bend. If the air be withdrawn from the siphon through this vent, the water will rise in the branches due to the atmospheric pressure without. When the two columns unite and the vent is closed, the liquid will flow from the upper reservoir as long as the end of the shorter branch of the siphon is below the surface of the liquid in the reservoir. If the water is free from air the height of the bend above the supply level might be as great as 33 ft. at sea level, and less at higher levels, corresponding approximately to the atmospheric pressure. In practice it is difficult to maintain more than about 80% of this.

If A = cross-section of the tube, sq. ft.; H = the difference in level between the two reservoirs, ft.; D = density of the liquid, lb. per cu. ft., then ADH measures the intensity of the force which causes the movement of the fluid, and $V = \sqrt{2gH} = 8.02\sqrt{H}$ is the theoretical velocity, ft. per sec., which is reduced by the loss of head for entry and friction, as in other cases of flow of liquids through pipes. In the case of the difference of level between the summit and the lower level being greater than 33 ft., however, the velocity of water in the shorter leg is limited to that due to a height of 33 ft.

8. TIDAL POWER

THE POWER OF OCEAN WAVES.—Albert W. Stahl, U. S. N. (*Trans.* A.S.M.E., xiii, 438), gives the following formulas based upon a theoretical discussion of wave motion: The total energy of one whole wave-length of a wave H ft. high, L ft. long, and 1 ft. wide, the length being the distance between successive crests, and the height the vertical distance between the crest and the trough, is

$$E = 8LH^2\{1 - 4.935(H^2/L^2)\} \text{ foot-pounds.}$$

The time required for each wave to travel through a distance equal to its own length is $P = \sqrt{L/5.123}$ seconds, and the number of waves passing any given point in one minute is $N = 60/P = 60\sqrt{5.123/L}$. Hence the total energy of an indefinite series of such waves, expressed in horsepower per foot of breadth, is

$$\frac{E \times N}{33{,}000} \quad 0.0329\frac{H^2L}{\sqrt{L}}\left(1 - 4.935\frac{H^2}{L^2}\right).$$

The utilization of the energy in ocean waves divides itself into:

1. The various motions of the water which may be utilized for power.
2. The wave-motor proper. That is, the portion of the apparatus in direct contact with the water, and receiving and transmitting the energy thereof; together with the mechanism for transmitting this energy to the machinery for utilizing the same.
3. Regulating devices, for obtaining a uniform motion from the irregular and more or less spasmodic action of the waves, as well as for adjusting the apparatus to the state of the tide and condition of the sea.
4. Storage arrangements for insuring a continuous and uniform output of power during a calm, or when the waves are comparatively small.

The motions that may be utilized for power purposes are the following: 1. Vertical rise and fall of particles at and near the surface. 2. Horizontal to-and-fro motion of particles at and near the surface. 3. Varying slope of surface of wave. 4. Impetus of waves rolling up the beach in the form of breakers. 5. Motion of distorted verticals. All of these motions have at various times been proposed to be utilized for power purposes.

The motion of distorted verticals is thus defined: A set of particles, originally in the same vertical straight line when the water is at rest, does not remain in a vertical line during the passage of the wave; so that the line connecting a set of such particles, while vertical and straight in still water, becomes distorted, as well as displaced, during the passage of the wave, its upper portion moving farther and more rapidly than its lower portion.

9. PIPE DATA *

STEEL AND WROUGHT-IRON PIPE.—For data on steel and wrought-iron pipe see pp. 5–23 to 5–68.

THICKNESS OF CAST-IRON WATER PIPES.—The specifications of the American Waterworks Assoc. (May, 1908) base the thickness of cast-iron water pipes on Brackett's formula

$$t = \{(P + P')r/3300\} + 0.25,$$

where t = thickness of pipe, in.; P = maximum static pressure, lb. per sq. in. for which the pipe is designed; P' = allowance for water ram; r = radius of pipe, in. The formula includes a large factor of safety to cover casting inequalities, water ram, and external stresses. The value of the allowance P' depends on the service. For ordinary waterworks service for pipes 42 to 60 in. diameter, a value of 70 lb. per sq. in. is sufficient. For smaller pipe the following values of P' are given by Brackett:

Diameter of pipe, in..............	36	30	24	20	16	12	10 to 3
P', lb. per sq. in..................	75	80	85	90	100	110	120

The values obtained by Brackett's formula are for ordinary conditions. For city service, the pipe should be made somewhat heavier.

Cast-iron pipe should be made of a soft and tough quality of iron, and should be tested to a pressure of twice the working pressure. It always should be coated by dipping in coal tar at a temperature of about 300° F. Table 16 gives thicknesses of cast-iron bell-and-spigot pipe for various classes of service. Table 17 gives the amount of lead required for joints in these pipes. For dimensions of cast-iron flanged fittings, see page 5–44.

LEAD AND TIN PIPE.—Weight of lead is taken 0.4106 lb. per cu. in. The safe working strength of lead is about 1/4 the elastic limit, or 225 lb. per sq. in.

Thickness of Lead Pipe, in Inches, Required for Given Head of Water is the product of the head in feet by size of pipe, expressed decimally, divided by 750.

EXAMPLE: Thickness of 1/2-in. pipe for a head of 25 ft. = 25 × 0.5/750 = 0.017. This rule corresponds to a safe working stress of 165 lb. per sq. in.

Lead Waste Pipe is made in the following sizes and weights: 2 1/2 in., 5 lb. and 10 lb. 10 oz. per ft.; 3 in., 6 lb. and 12 lb. 8 oz. per ft.; 4 in., 7 lb. 14 oz. and 16 lb. 6 oz. per ft.; 5 in., 9 lb. 14 oz. and 20 lb. 4 oz. per ft.; 6 in., 11 lb. 13 oz. and 24 lb. 2 oz. per ft.

Lead-lined Pipe is particularly adapted for use in contact with acids, mine water, salt water, or any liquid which has a corrosive action on iron pipe. See Tables 20 and 22.

Table 16.—Standard Thicknesses and Weights of Cast-Iron Pipe

(U. S. Pipe and Foundry Co., Burlington, N. J.)

Class..	A		B		C		D		E		F		G		H	
Head.	100 ft.		200 ft.		300 ft.		400 ft.		500 ft.		600 ft.		700 ft.		800 ft.	
Press..	43 lb.		86 lb.		130 lb.		173 lb.		217 lb.		260 lb.		304 lb.		347 lb.	
	Ordinary Service										Fire Lines and High-pressure Service					
Nominal Inside Diam., in.	Thickness, in.	Lb. per ft.	Thickness, in.	Lb. per ft.	Thickness, in.	Lb. per ft.	Thickness, in.	Lb. per ft.	Thickness, in.	Lb. per ft.	Thickness, in.	Lb. per ft.	Thickness, in.	Lb. per ft.	Thickness, in.	Lb. per ft.
3	0.39	14.5	0.42	16.2	0.45	17.1	0.48	18.0	.....	.....	.....	.....	.....	.....	.....	.....
4	.42	20.0	.45	21.7	.48	23.3	.52	25.0	.....	.....	.....	.....	.....	.....	.....	.....
6	.44	30.8	.48	33.3	.51	35.8	.55	38.3	0.58	42.5	0.61	44.3	0.65	48.1	0.69	50.5
8	.46	42.9	.51	47.5	.56	52.1	.60	55.8	.66	60.9	.71	66.8	.75	72.3	.80	76.1
10	.50	57.1	.57	63.8	.62	70.8	.68	76.7	.74	86.9	.80	92.8	.86	101.4	.92	107.3
12	.54	72.5	.62	82.1	.68	91.7	.75	100.0	.82	114.6	.89	122.8	.97	136.2	1.04	144.4
14	.57	89.6	.66	102.5	.74	116.7	.82	129.2	.90	145.6	.99	158.8	1.07	175.1	1.16	187.5
16	.60	108.3	.70	125.0	.80	143.8	.89	158.3	.98	180.7	1.08	196.5	1.18	218.0	1.27	233.8
18	.64	129.2	.75	150.0	.87	175.0	.96	191.7	1.07	221.8	1.17	239.3	1.28	268.2	1.39	287.8
20	.67	150.0	.80	175.0	.92	208.3	1.03	229.2	1.15	265.8	1.27	287.3	1.39	321.8	1.51	345.8
24	.76	204.2	.89	233.3	1.04	279.2	1.16	306.7	1.31	359.1	1.45	392.3	1.75	479.8	1.88	510.6
30	.88	291.7	1.03	333.3	1.20	400.0	1.37	450.0	1.55	530.9	1.73	588.8	.....	.....	.....	.....
36	.99	391.7	1.15	454.2	1.36	545.8	1.58	625.0	1.80	738.1	2.02	821.0	.....	.....	.....	.....
42	1.10	512.5	1.28	591.7	1.54	716.7	1.78	825.0	.....	.....	.....	.....	.....	.....	.....	.....
48	1.26	666.7	1.42	750.0	1.71	908.3	1.96	1050.0	.....	.....	.....	.....	.....	.....	.....	.....
54	1.35	800.0	1.55	933.3	1.90	1141.7	2.23	1341.7	.....	.....	.....	.....	.....	.....	.....	.....
60	1.39	916.7	1.67	1104.2	2.00	1341.7	2.38	1583.3	.....	.....	.....	.....	.....	.....	.....	.....
72	1.62	1281.9	1.95	1547.3	2.39	1904.3	.....	.....	.....	.....	.....	.....	.....	.....	.....	.....
84	1.72	1635.8	2.22	2104.1	.....	.....	.....	.....	.....	.....	.....	.....	.....	.....	.....	.....

The above weights are to lay 12 ft. lengths, and include standard sockets.

* Staff Contribution.

Lead-covered Iron Pipe for use in bleacheries, etc., where steam passes through the pipe and the exterior is in contact with acid or corrosive solutions is made in commercial of 1/2, 3/4, 1, 1 1/4, 1 1/2, 2, and 3 in.

Table 17.—Lead Required for Cast-iron Pipe Bell and Spigot Joints*

(U. S. Pipe and Foundry Co., Burlington, N. J.)

Size of Pipe, in.	Lb. of Lead, Joint 2 in. Deep				Lb. of Lead, Joint 2 1/2 in. Deep				Lb. of Lead, Solid Pipe Joint				
	Class A	Class B	Class C	Class D	Class A	Class B	Class C	Class D	Class A	Class B	Class C	Class D	Class E,F,G,H
3	5.86	6.07	6.07	6.07	7.00	7.25	7.25	7.25	9.27	9.60	9.60	9.60	....
4	7.16	7.41	7.41	7.41	8.57	8.88	8.88	8.88	11.38	11.80	11.80	11.80	....
6	9.87	10.13	10.13	10.13	11.85	12.16	12.16	12.16	15.80	16.22	16.22	16.22	21.9
8	12.55	12.55	12.98	12.98	15.21	15.21	15.60	15.60	22.88	22.88	23.47	23.47	28.2
10	15.30	15.30	15.70	15.70	18.42	18.42	18.89	18.89	27.75	27.75	28.46	28.46	34.5
12	18.00	18.00	18.42	18.42	21.70	21.70	22.18	22.18	32.75	32.75	33.45	33.45	40.8
14	20.75	20.75	21.20	21.20	25.00	25.00	25.54	25.54	37.73	37.73	38.56	38.56	47.1
16	28.45	28.45	29.07	29.07	34.50	34.50	35.25	35.25	52.66	52.66	53.82	53.82	53.4
18	31.75	31.75	32.38	32.38	38.50	38.50	39.28	39.28	58.78	58.78	60.00	60.00	59.7
20	35.03	35.03	35.75	35.75	42.50	42.50	43.37	43.37	64.50	64.50	66.25	66.25	66.0
24	41.60	41.60	42.40	42.40	50.50	50.50	51.00	51.00	77.18	77.18	78.18	78.18	79.4
30	50.87	51.25	51.93	52.45	61.75	62.25	63.00	63.68	105.00	106.25	107.55	108.18	122.9
36	60.62	61.15	61.75	63.00	73.60	74.25	75.00	76.95	125.62	126.75	128.00	130.60	146.7
42	70.36	70.82	71.64	73.87	85.46	86.04	87.06	90.09	161.05	162.12	164.16	169.38	....
48	80.25	80.68	81.61	82.50	97.46	98.03	99.14	100.17	183.70	184.76	186.88	189.00	....
54	89.84	90.54	91.63	92.56	109.16	110.00	111.34	112.47	225.11	226.88	229.62	231.94	....
60	99.41	100.40	101.64	102.41	120.81	122.00	123.52	124.45	249.21	251.66	254.78	256.68	....
72	145.01	146.22	147.86	....	177.13	179.37	180.66	....	369.85	374.61	377.22	....	....
84	168.17	170.06	....	....	205.43	207.73	....	....	429.02	423.34	....	....	....

* Table allows 5% to cover compression of lead in calking. To cover variation in lead room, add 10%. Weight of lead taken as 0.41 lb. per cu. in.

Table 18.—Weight and Bursting Strength of Spiral Riveted Pressure Pipe

(Taylor Forge and Pipe Works, Chicago)

Inside Diam., in.	Thickness, U. S. Standard Gage	Weight, lb. per ft.	Approx. Bursting Strength, lb. per sq. in.	Inside Diam., in.	Thickness, U. S. Standard Gage	Weight, lb. per ft.	Approx. Bursting Strength, lb. per sq. in.	Inside Diam., in.	Thickness, U. S. Standard Gage	Weight, lb. per ft.	Approx. Bursting Strength, lb. per sq. in.
3	18	2.3	2000	14	14	15.9	670	26	12	39.5	505
4	16	3.7	1875	14	12	22.2	940	26	10	49.5	650
5	16	4.5	1500	14	10	27.6	1210	26	8	59.8	795
6	16	5.3	1250	15	14	17.0	625	26	6	70.0	935
6	14	6.6	1560	15	12	23.7	875	28	10	51.7	605
6	12	9.2	2170	15	10	29.6	1125	28	8	63.6	735
7	16	6 2	1070	16	14	18.1	585	28	6	76.6	870
7	14	7.7	1340	16	12	25.2	820	30	10	56.8	560
7	12	10.7	1860	16	10	31.5	1050	30	8	68.7	685
8	16	7.1	935	16	8	38.1	1290	30	6	80.5	810
8	14	8.8	1170	16	6	44.7	1520	32	10	61.6	525
8	12	12.3	1640	18	14	19.9	520	32	8	74.3	645
9	16	8 0	835	18	12	27.6	730	32	6	87.1	760
9	14	9.9	1045	18	10	34.5	940	34	10	65.4	490
9	12	13.9	1460	18	8	41.6	1140	34	8	78.8	600
10	16	8.8	750	18	6	49.0	1360	34	6	93.6	715
10	14	11.0	935	20	14	22.1	470	36	10	69.1	470
10	12	15.3	1310	20	12	30.6	660	36	8	83.4	570
11	16	9.7	680	20	10	38.3	840	36	6	97.8	680
11	14	12.0	850	20	8	46.2	1030	38	10	73.2	445
11	12	16.6	1200	20	6	54.1	1220	38	8	88.1	540
12	16	10.6	625	22	12	33.7	595	38	6	103.2	640
12	14	13.0	780	22	10	42.2	765	40	10	76.7	420
12	12	18.2	1080	22	8	50.8	940	40	8	92.4	515
12	10	22.5	1410	22	6	59.5	1108	40	6	108.5	610
13	14	14.1	720	24	12	36.5	540	42	10	80.9	400
13	12	19.7	1010	24	10	45.7	705	42	8	97.2	490
13	10	24.5	1295	24	8	55.2	820	42	6	114.0	580
				24	6	64.6	1015				

Table 19.—Weight per Foot, Pounds, of Seamless Brass Tubes *
(Condensed from Manufacturers' Standard Tables, 1932)

Outside Diam., in.	Thickness of Wall by Stubs' Gage and Decimal Equivalent in Inches												
	2	4	6	8	10	12	14	16	18	20	22	24	26
	0.284	0.238	0.203	0.165	0.134	0.109	0.083	0.065	0.049	0.035	0.028	0.022	0.018
1/16	.....	.....	.....	.....	.....	.....	.....	.....	.....	.....	.....	0.0104	0.0094
5/64	.....	.....	.....	.....	.....	.....	.....	.....	.....	.....	.....	.014	.012
3/32	.....	.....	.....	.....	.....	.....	.....	.....	.....	.....	.....	.018	.016
7/64	.....	.....	.....	.....	.....	.....	.....	.....	.....	.....	.....	.022	.019
1/8	.....	.....	.....	.....	.....	.....	.....	.....	0.043	0.036	0.031	.026	.022
5/32	.....	.....	.....	.....	.....	.....	.....	.....	.061	.049	.041	.034	.029
3/16	.....	.....	.....	.....	.....	.....	.....	.....	.079	.062	.052	.042	.035
7/32	.....	.....	.....	.....	.....	.....	.....	.....	.096	.075	.062	.050	.042
1/4	.....	.....	.....	.....	.....	0.178	0.160	0.139	.114	.087	.072	.058	.048
9/32	.....	.....	.....	.....	.....	.....	.....	.....	.132	.100	.082	.066	.055
5/16	.....	.....	.....	.....	.....	.....	.....	.186	.150	.113	.092	.074	.061
3/8	.....	.....	.....	0.401	0.373	.335	.280	.233	.185	.138	.112	.090	.074
7/16	.....	.....	.....	.....	.....	.....	.340	.280	.220	.163	.133	.106	.087
1/2	.....	.....	0.698	.640	.568	.493	.400	.327	.256	.188	.153	.122	.100
9/16	.....	.....	.....	.....	.....	.....	.460	.374	.291	.214	.173	.138	.114
5/8	.....	1.066	.991	.878	.761	.651	.520	.421	.327	.239	.193	.153	.126
3/4	.....	1.410	1.285	1.12	.955	.808	.641	.515	.397	.290	.234	.185	.152
7/8	.....	1.754	1.578	1.36	1.15	.966	.761	.609	.468	.340	.274	.217	.178
1	2.35	2.098	1.872	1.59	1.34	1.12	.881	.703	.539	.391	.315	.249	.205
1 1/4	3.17	2.786	2.459	2.07	1.73	1.44	1.12	.891	.681	.492	.396	.313	.....
1 1/2	4.00	3.475	3.046	2.55	2.12	1.75	1.36	1.08	.823	.593	.477	.376	.....
1 3/4	4.82	4.16	3.63	3.03	2.51	2.07	1.60	1.27	.964	.694	.558	440	.....
2	5.64	4.85	4.22	3.50	2.89	2.39	1.84	1.46	1.11	.796	.639	.503	.....
2 1/4	6.46	5.54	4.81	3.98	3.28	2.70	2.08	1.64	1.25	.897	.720	.567	.....
2 1/2	7.28	6.23	5.40	4.46	3.67	3.02	2.32	1.83	1.39	.998	.801	.....	.....
2 3/4	8.10	6.92	5.98	4.94	4.06	3.33	2.56	2.02	1.53	1.10	.882	.....	.....
3	8.93	7.61	6.57	5.41	4.44	3.65	2.80	2.21	1.67	1.20	.963	.....	.....
3 1/4	9.76	8.29	7.16	5.89	4.83	3.96	3.04	2.40	1.82	1.30	1.04	.....	.....
3 1/2	10.58	8.98	7.74	6.37	5.22	4.28	3.28	2.58	1.96	1.40	1.13	.....	.....
3 3/4	11.40	9.67	8.33	6.84	5.61	4.59	3.52	2.77	2.10	1.50	1.21	.....	.....
4	12.23	10.36	8.92	7.32	5.99	4.91	3.76	2.96	2.24	1.61	1.29	.....	.....
4 1/4	13.04	11.05	9.51	7.80	6.38	5.22	4.00	3.15	2.38	1.71	1.37	.....	.....
4 1/2	13.87	11.74	10.09	8.28	6.77	5.54	4.24	3.34	2.52	1.81	1.45	.....	.....
4 3/4	14.68	12.42	10.68	8.75	7.16	5.85	4.48	3.52	2.67	1.91	.....	.....	.....
5	15.51	13.11	11.27	9.23	7.54	6.17	4.72	3.71	2.81	2.01	.....	.....	.....
5 1/4	16.34	13.80	11.86	9.71	7.93	6.48	4.96	3.90	2.95	2.11	.....	.....	.....
5 1/2	17.16	14.49	12.44	10.18	8.32	6.80	5.20	4.09	3.09	2.21	.....	.....	.....
5 3/4	17.98	15.18	13.03	10.66	8.71	7.11	5.44	4.28	3.23	2.31	.....	.....	.....
6	18.80	15.87	13.62	11.14	9.09	7.43	5.68	4.46	3.37	2.42	.....	.....	.....
6 1/4	19.62	16.56	14.20	11.62	9.48	7.75	5.92	4.65	3.52	.....	.....	.....	.....
6 1/2	20.46	17.24	14.79	12.09	9.87	8.06	6.16	4.84	3.66	.....	.....	.....	.....
6 3/4	21.29	17.93	15.38	12.57	10.26	8.38	6.40	5.03	3.80	.....	.....	.....	.....
7	22.09	18.62	15.96	13.05	10.64	8.69	6.64	5.22	3.94	.....	.....	.....	.....
7 1/4	22.19	19.31	16.55	13.53	11.03	9.01	6.88	5.40	.....	.....	.....	.....	.....
7 1/2	23.75	20.00	17.14	14.00	11.42	9.32	7.12	5.59	.....	.....	.....	.....	.....
7 3/4	24.57	20.69	17.73	14.48	11.81	9.64	7.36	5.78	.....	.....	.....	.....	.....
8	.....	21.37	18.31	14.96	12.20	9.95	7.60	5.97	.....	.....	.....	.....	.....
8 1/2	.....	22.75	19.49	15.91	12.97	10.58	8.08	6.34	.....	.....	.....	.....	.....
9	.....	24.13	20.66	16.88	13.75	11.21	8.56	6.72	.....	.....	.....	.....	.....
9 1/2	.....	25.50	21.84	17.82	14.52	11.84	9.04	7.10	.....	.....	.....	.....	.....
10	.....	26.88	23.01	18.78	15.30	12.47	.....	.....	.....	.....	.....	.....	.....
10 1/2	33.57	28.26	24.18	19.73	16.07	13.11	10.01	.....	.....	.....	.....	.....	.....
11	35.21	29.63	25.36	20.68	16.85	13.74	10.49	.....	.....	.....	.....	.....	.....
11 1/2	36.85	31.01	26.53	21.63	17.63	14.37	10.97	.....	.....	.....	.....	.....	.....
12	38.50	32.39	27.71	22.59	18.40	15.00	11.45	.....	.....	.....	.....	.....	.....
12 1/2	40.14	33.77	28.88	23.55	19.17	15.63	11.93	.....	.....	.....	.....	.....	.....
13	41.78	35.14	30.06	24.50	19.95	16.26	12.41	.....	.....	.....	.....	.....	.....
13 1/2	43.43	36.52	21.23	25.46	20.72	16.89	12.86	.....	.....	.....	.....	.....	.....
14	45.07	37.90	32.41	26.41	21.50	17.52	13.37	.....	.....	.....	.....	.....	.....
14 1/2	46.71	39.27	33.58	27.37	22.27	18.16	13.85	.....	.....	.....	.....	.....	.....
15	48.36	40.65	34.75	28.32	23.05	18.79	14.33	.....	.....	.....	.....	.....	.....
16	51.64	43.40	37.10	30.23	24.60	20.05	15.29	.....	.....	.....	.....	.....	.....
17	54.93	46.16	39.45	32.14	26.15	21.31	16.25	.....	.....	.....	.....	.....	.....
For I. D. tubes, add†	1.87	1.31	0.954	[illegible]	[illegible]	[illegible]	[illegible]	[illegible]	[illegible]	[illegible]	[illegible]	[illegible]	0.008

* [illegible]

† [illegible] corresponding gage number to the weight of tube of the same outside diameter. Example: Tube, 12 in. O. D., No. 6 gage, weighs 27.71 lb. per ft.; tube, 12 in. I. D., No. 6 gage weighs (27.71 + 0.954) = 28.664 lb. per ft.

Brass and Copper Pipes, Lined with Tin or Lead are made in commercial sizes of 1/2, 3/4, 1, 1 1/4, 1 1/2, and 2 in.

Sheet Lead is rolled to any weight per square foot from 1 to 7 lb. in any width up to 11 ft. 6 in., and from 8 lb. up, 12 ft. wide. A square foot of rolled sheet lead 1 in. thick weighs approximately 59 1/2 lb.

WOOD-STAVE PIPE.—Pipe made of wood staves is extensively used on the Pacific Coast and in certain portions of the East, for conveying water, acids, paper pulp, and also substances which would react on iron or steel pipe. The pipes are made of wood staves, planed to the inner and outer radii of the pipe, with edges cut radial and tongued and grooved to make water-tight joints. The pipes are reinforced with iron or steel bands, or with a spiral of round or flat wire, the size depending upon the pressure to be sustained. Pipes up to 32 in. diameter are assembled and wound in the factory, while larger diameters are built up in the field. Table 25 gives data concerning the sizes and weights of wood-stave pipe.

Square Wood Pipe is extensively used for tanneries. It consists of square blocks with an axial hole and with tenon and socket joints. The sizes range from 4 × 4 in. outside measurement for a 2-in. diameter hole to 12 × 12 in. outside measurement with a 6-in. hole.

Table 20.—Weight of Deoxidized Copper Tubing for Water Piping

(American Brass Co., Waterbury, Conn.)

Nominal Size, in.	Outside Diam., in.	For Underground Service				For Interior Plumbing			
		Thickness, in.	Pound per ft.	Bursting Pressure, lb. per sq. in.*		Thickness, in.	Pound per ft.	Bursting Pressure, lb. per sq. in.*	
				Hard	Soft			Hard	Soft
1/8	0.250	0.032	0.085	11,500	8600	0.025	0.068	8800	6600
1/4	.375	.032	.134	7,400	5500	.030	.126	6900	5200
3/8	.500	.049	.269	9,800	7300	.035	.198	6600	4900
1/2	.625	.049	.344	7,400	5600	.040	.284	5800	4400
3/4	.875	.065	.641	7,000	5300	.045	.454	4600	3500
1	1.125	.065	.839	5,200	3900	.050	.653	3800	2900
1 1/4	1.375	.065	1.04	4,200	3100	.055	.882	3500	2600
1 1/2	1.625	.072	1.36	3,900	2900	.060	1.14	3200	2400
2	2.125	.083	2.06	3,400	2500	.070	1.75	2800	2100
2 1/2	2.625	.095	2.92	3,100	2300	.080	2.48	2400	1800
3	3.125	.109	4.00	3,100	2300	.090	3.33	2300	1700
3 1/2	3.625	.120	5.12	2,800	2100	.100	4.29	2200	1700
4	4.125	.134	6.51	2,800	2100	.110	5.38	2100	1600
5	5.125	.160	9.67	2,600	1900	.125	7.61	2000	1500
6	6.125	.192	13.87	2,600	1900	.140	10.20	1800	1400

* Calculated.

Table 21.—Tin-lined and Lead-lined Iron Pipe

(National Lead Co., New York)

Size, in.	Wt. per ft., lb.		Size, in.	Wt. per ft., lb.		Size, in.	Wt. per ft., lb.		Size, in.	Wt. per Ft., lb.
	Lead-lined	Tin-lined		Lead-lined	Tin-lined		Lead-lined	Tin-lined		Lead-lined
1/2	1 3/8	1	2	6 1/8	5 1/4	4 1/2	18	16	9	66
3/4	1 5/8	1 3/8	2 1/2	8 1/2	7 1/2	5	21 1/2	26 1/10	10	75
1	2 1/2	2 1/4	3	11 1/2	10 1/6	6	29 3/4	19 1/6	12	88
1 1/4	3 1/2	3	3 1/2	14 1/2	12 8/10	7	36	......	......	......
1 1/2	4 3/8	3 3/4	4	15 2/3	14 1/6	8	47	......	......	......

Table 22.—Weight per Foot of Brass-lined and Copper-lined Iron Pipe

(National Lead Co., New York)

		r-lined			Wt., lb. Copper-	lined	ed	Wt., Bra	lined
1/2			1 1/4	2 2/3	2 2/3	/10	6 3/4	19 1/2	19 3/4
3/4	1 3/8		1 1/2	3 1/4	3 1/4	8 3/4	8 8/10	25 1/4	25 6/10
			2	4 1/3		12 6/10	12 7/10	38	38 1/2

Many valuable data on wood pipe and its uses are contained in the catalogs of the A. Wyckoff & Son Co., Elmira, N. Y.; The Standard Wood Pipe Co., Williamsport, Pa.; the Michigan Pipe Co., Bay City, Mich.; and the Continental Pipe Co., Seattle, Wash.

Table 23.—Lead and Tin-lined Lead Pipe

(National Lead Co., New York)

Inside Diam., in.	Classification	Outside Diam., in.	Weight per Foot Lb.	Weight per Foot Oz.	Inside Diam., in.	Classification	Outside Diam., in.	Weight per Foot Lb.	Weight per Foot Oz.	Inside Diam., in.	Classification	Outside Diam., in.	Weight per Foot Lb.	Weight per Foot Oz.
3/8	E	0.520	....	8	3/4	E	0.906	1	....	1 1/2	E	1.740	3	....
3/8	D	.549	....	10	3/4	D	.940	1	4	1 1/2	D	1.776	3	8
3/8	C	.577	....	12	3/4	C	1.006	1	12	1 1/2	C	1.830	4	4
3/8	B	.631	1	....	3/4	B	1.068	2	4	1 1/2	B	1.882	5	....
3/8	A	.725	1	8	3/4	A	1.156	3	....	1 1/2	A	1.984	6	8
3/8	AA	.811	2	....	3/4	AA	1.212	3	8	1 1/2	AA	2.076	8	....
3/8	AAA	.888	2	8	3/4	AAA	1.336	4	12	1 1/2	AAA	2.272	11	4
1/2	E	.628	....	9	1	E	1.192	1	10	1 3/4	D	2.024	4	....
1/2	D	.666	....	12	1	D	1.232	2	....	1 3/4	C	2.086	5	..
1/2	C	.712	1	....	1	C	1.284	2	8	1 3/4	B	2.146	6	....
1/2	B	.756	1	4	1	B	1.356	3	4	1 3/4	A	2.193	6	12
1/2	A	.798	1	8	1	A	1.428	4	....	1 3/4	AA	2.404	10	8
1/2	AA	.876	2	....	1	AA	1.492	4	12	1 3/4	AAA	2.624	14	12
1/2	AAA	1.012	3	....	1	AAA	1.596	6	....	......	....	......	....	....
5/8	E	0.765	....	12	1 1/4	E	1.442	2	....	2	E	2.185	3	....
5/8	D	.803	1	....	1 1/4	D	1.486	2	8	2	D	2.284	4	12
5/8	C	.881	1	8	1 1/4	C	1.528	3	....	2	C	2.354	6	....
5/8	B	.953	2	....	1 1/4	B	1.592	3	12	2	B	2.410	7	....
5/8	A	1.019	2	8	1 1/4	A	1.670	4	12	2	A	2.503	8	12
5/8	AA	1.082	3	....	1 1/4	AA	1.765	6	....	2	AA	2.751	13	12
5/8	AAA	1.137	3	8	1 1/4	AAA	1.889	7	12	2	AAA	3.008	19	8

In all sizes of lead pipe from 3/8-in. diam. and upwards as given in Table 23 the wall thickness is such that all pipes in Class A, regardless of diameter will safely withstand a constant cold water pressure of 50 lb. per sq. in.; Class AA pipes, 75 lb. per sq. in.; Class AAA pipes, 100 lb. per sq. in.

Table 24.—Block Tin Pipe and Tubing

Diam., in. Inside	Diam., in. Outside	Thickness, in.	Wt. per Foot, oz.	Diam., in. Inside	Diam., in. Outside	Thickness, in.	Wt. per Foot, oz.	Diam. in. Inside	Diam. in. Outside	Thickness, in.	Wt. per Foot, oz.
Tubing				Pipe				Pipe			
1/8	0.25	0.062	1.9	3/8	0.495	0.06	4	5/8	0.800	0.037	10
1/8	.202	.0385	1	3/8	.503	.064	4 1/2	5/8	.831	.103	12
3/16	.292	.053	2	3/8	.515	.07	5	3/4	.901	.076	10
3/16	.331	.072	3	3/8	.539	.082	6	3/4	.928	.089	12
3/16	.367	.09	4	3/8	.561	.093	7	1	1.172	.086	15
1/4	.388	.069	3 1/2	3/8	.584	.104	8	1	1.204	.102	18
Pipe				1/2	.632	.066	6	1 1/4	1.436	.093	20
				1/2	.670	.085	8	1 1/4	1.471	.110	24
1/4	.400	.075	4	1/2	.707	.103	10	1 1/2	1.746	.123	32
1/4	.433	.091	5	1/2	.741	.120	12	1 1/2	1.802	.151	40
5/16	.444	.066	4	5/8	.735	.055	6	2	2.236	.118	40
7/16	.562	.065	5	5/8	.768	.071	8	2	2.280	.140	48

Weight of tin taken is 0.2652 lb. per cu. in.

Table 25.—Diameters, Weights, etc., of Wood-stave Pipe

(A. Wyckoff & Son Co., Elmira, N. Y.)

Size, in.	Outside Diameter, in.	Weight per Foot, lb.*	No. Feet in Carload, 40-ft. Car	Size, in.	Outside Diameter, in.	Weight per Foot, lb.*	No. Feet in Carload, 40-ft. Car
6	10 1/8	15	3100	18	22 1/8	35	800
8	12 1/8	17	2600	20	24 1/8	40	700
10	14 1/8	20	2100	24	28 1/8	48	500
12	16 1/8	25	1600	30	36 1/8	90	275
14	18 1/8	28	1200	36	42 1/8	120	160
16	20 1/8	32	1000	48	54 1/8	166	100

* Pipe for 80-lb. pressure.

Table 26.—Dimensions and Weights of Single Ring Segmental Block Sewers
(Clay Products Assoc., Chicago)

Diam. of Sewer, in.	No. of Blocks to Circle	No. of Blocks per Foot	Block Dimensions, in.			Wt. per Foot of Sewer, lb.	Diam. of Sewer, in.	No. of Blocks to Circle	No. of Blocks per Foot	Block Dimensions, in.			Wt. per Foot of Sewer, lb.
			Width	Length	Thickness					Width	Length	Thickness	
30	10	5	9 1/4	24	4 3/4	260	60	18	9	10 3/16	24	6 3/4	760
33	11	5 1/2	9 1/4	24	4 3/4	290	66	20	10	10 3/16	24	6 3/4	840
36	12	6	9 1/4	24	4 3/4	316	72	22	11	10	24	7 7/8	1120
39	13	6 1/2	9 1/4	24	5 1/2	370	78	24	12	10	24	7 7/8	1220
42	14	7	9 1/4	24	5 1/2	416	84	26	13	9 7/8	24	8 1/4	1300
45	15	7 1/2	9 1/4	24	5 1/2	440	90	28	14	9 7/8	24	8 1/4	1400
48	14	7	10 1/2	24	6	480	96	28	14	10 1/2	24	8 1/2	1600
51	15	7 1/2	10 1/4	24	6	520	102	30	15	10 1/2	24	8 1/2	1700
54	16	8	10 1/4	24	6	560	108	30	15	11 1/8	30	10 1/2	1800

Table 27.—Dimensions and Weights of Double Ring Segmental Block Sewers
(Clay Products Assoc., Chicago)

Inside Diam. of Sewer, in.	No. of Blocks to Circle	Thickness of Shell, in.	Thickness of Wearing Surface, in.	Approx. Weight per Lineal Foot, lb.	Inside Diam. of Sewer, in.	No. of Blocks to Circle	Thickness of Shell, in.	Thickness of Wearing Surface, in.	Approx. Weight per Lineal Foot, lb.
30	10	5	1 7/8	335	54	18	6 1/2	2 1/8	725
33	11	5	1 7/8	365	60	20	7 1/4	2 1/4	895
36	12	5	1 7/8	400	66	22	7 1/4	2 1/4	985
39	13	5	1 7/8	435	72	24	7 1/4	2 1/4	1075
42	14	5	1 7/8	470	78	26	8	2 1/4	1220
45	15	5 3/4	2	540	84	28	8	2 1/4	1320
48	16	5 3/4	2	590	90	30	9	2 3/4	1450

Table 28.—Vitrified Clay and Concrete Pipe
(Clay Products Assoc., Chicago)

Inside Diam., in.	Standard Strength Pipe					Double Strength Pipe				
	Approx. Wt. per Foot, lb.	Laying Length, ft.	Depth of Socket, in.	Annular Space, in.	Thickness of Barrel, in.	Approx. Wt. per Foot, lb.	Laying Length, ft.	Depth of Socket, in.	Annular Space, in.	Thickness of Barrel, in.
4	9	2	1 1/2	3/8	9/16	9	2	1 1/2	3/8	9/16
6	15	2	2	7/16	5/8	15	2	2	7/16	5/8
8	24	2 or 2 1/2	2 1/4	1/2	3/4	24	2 or 2 1/2	2 1/4	1/2	3/4
10	33	2 or 2 1/2	2 1/2	1/2	7/8	33	2 or 2 1/2	2 1/2	1/2	7/8
12	45	2 or 2 1/2	2 1/2	1/2	1	45	2 or 2 1/2	2 1/2	1/2	1
15	65	2 or 2 1/2	2 1/2	1/2	1 1/8	75	2 or 2 1/2	2 1/2	1/2	1 1/4
18	85	2 or 2 1/2	3	1/2	1 1/4	105	2 or 2 1/2	3	1/2	1 1/2
21	120	2 or 2 1/2	3	9/16	1 1/2	145	2 or 2 1/2	3	9/16	1 3/4
24	150	2, 2 1/2 or 3	3	5/8	1 5/8	185	2, 2 1/2 or 3	3	5/8	2
27	220	2 1/2 or 3	3 1/2	5/8	2 1/8	235	2 1/2 or 3	3 1/2	5/8	2 1/4
30	260	2 1/2 or 3	3 1/2	3/4	2 1/4	300	2 1/2 or 3	3 1/2	3/4	2 1/2
33	310	2 1/2 or 3	4	7/8	2 3/8	350	2 1/2 or 3	4	7/8	2 5/8
36	360	2 1/2 or 3	4	1	2 1/2	385	2 1/2 or 3	4	1	2 3/4

Table 29.—Dimensions and Weights of Reinforced Concrete Pipe
(Independent Concrete Pipe Co., Indianapolis)

Size, in.	Wall Thickness, in.	Min. Cross-sectional Area of Circular Reinforcement per ft. of Pipe, sq. in.	No. of Rings	Size, in.	Wall Thickness, in.	Min. Cross-sectional Area of Circular Reinforcement per ft. of Pipe, sq. in.	No. of Rings	Size, in.	Wall Thickness, in.	Min. Cross-sectional Area of Circular Reinforcement per ft. of Pipe, sq. in.	No. of Rings
24	3	0.068	1	42	4 1/2	0.153	1	78	8	0.413	2
27	3	.080	1	48	5	.219	2	84	8	.425	2
30	3 1/2	.093	1	54	5 1/2	.260	2	90	8	.495	2
33	3 3/4	.107	1	60	6	.294	2	96	8 1/2	.581	2
36	4	.126	1	66	6 1/2	.333	2	102	8 1/2	.610	2
39	4 1/4	.146	1	72	7	.361	2	108	9	.682	2

Sizes up to and including 72 in. are manufactured in 4-ft. lengths, and above 72 in., in 5-ft. lengths.

HYDRAULIC TURBINES

By R. E. B. Sharp

References: Church, Hydraulic Motors; Camerer, Wasserkraft Maschinen; Kaplan-Lechner, Theory and Construction of High Speed Turbine Runners; Spannhake, Runners as Impellers and Propellers, Berlin, Springer; Wasserkraft Jahrbuch (Munich) 1928–29, 1929-30, 1930–31, etc.; Creager & Justin, Hydro-electric Handbook; others cited hereinafter.

1. GENERAL

Hydraulic turbines are divided into two general types: (*a*) reaction turbines, and (*b*) impulse turbines. Both types, however, actually involve both reaction and impulse. The two classifications are not well named, but the terms are well understood. The essential difference in the two types is that the water enters the runners of reaction turbines after only a portion of its energy has been converted into velocity; that is, it enters the reaction turbine runner under pressure. On the other hand, the water strikes the runners of impulse turbines under atmospheric pressure after all of its energy has been converted into velocity. Fig. 1 shows a hydroelectric unit comprising a reaction turbine driving a generator, and Fig. 2 a hydroelectric unit involving an impulse turbine.

The class of reaction turbine in most extensive use today is the Francis inward-flow type, shown in Fig. 1. For the lower heads, where high specific speed (see p. 2–43) is required, the propeller type, with adjustable blade (Fig. 3), (commonly known as the Kaplan type), is at the present time (1935) being extensively adopted. For all of the above types of reaction turbine, movable gates with axes parallel to the turbine shaft are used to control the flow of water to the turbine runner. The governor actuates the gates through the operating ring as indicated in Fig. 1. The water enters the volute casing from the intake passages, passes through the stay vanes, movable gates, and runner into the draft-tube through which it flows into the tail-race below the power house. The runner blades of the Kaplan type are adjusted during operation synchronously with the movable gates, with resulting high part-load efficiency. The water acts on the runner

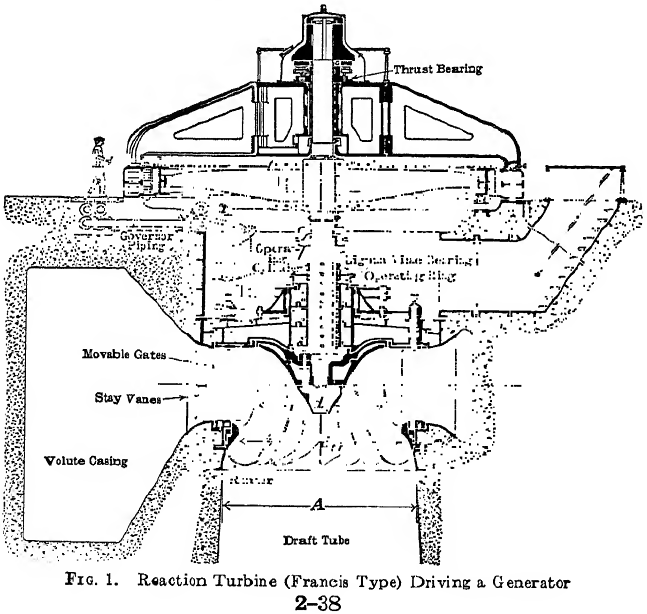


Fig. 1. Reaction Turbine (Francis Type) Driving a Generator

of the impulse turbine, Fig. 2, generally through one nozzle as indicated, although two, three, or four may be used to obtain higher speed. Where more than two nozzles are used it is necessary to adopt the vertical shaft. The power developed is controlled by actuating the needle shown in the center of the nozzle or by deflecting the stream between the nozzle and the runner.

SPECIFIC SPEED.—To compare all turbine runners directly they must be reduced to a common basis. If a turbine runner be reduced mathematically to such a size that it would develop one horsepower under a head of one foot, the speed in revolutions per minute at which it would operate under these conditions is known as its *Specific Speed.* The specific speed is the commonly-accepted basis of comparison of the performance of reaction and impulse turbine runners. Suppose a turbine operates at a speed of r.p.m. under a head of H, and that it develop power = Hp. The power developed by a turbine varies with the head and with the quantity of water flowing. The latter varies as the square root of the head. Therefore, the above turbine under a head of 1 ft. will develop

$$\text{Hp.} \times (1/H) \times (1/\sqrt{H}) = (\text{Hp.}/H^{3/2}) \quad . \quad . \quad . \quad . \quad . \quad . \quad [1]$$

Since the speed varies as the square root of the head, this turbine runner under 1 ft. head

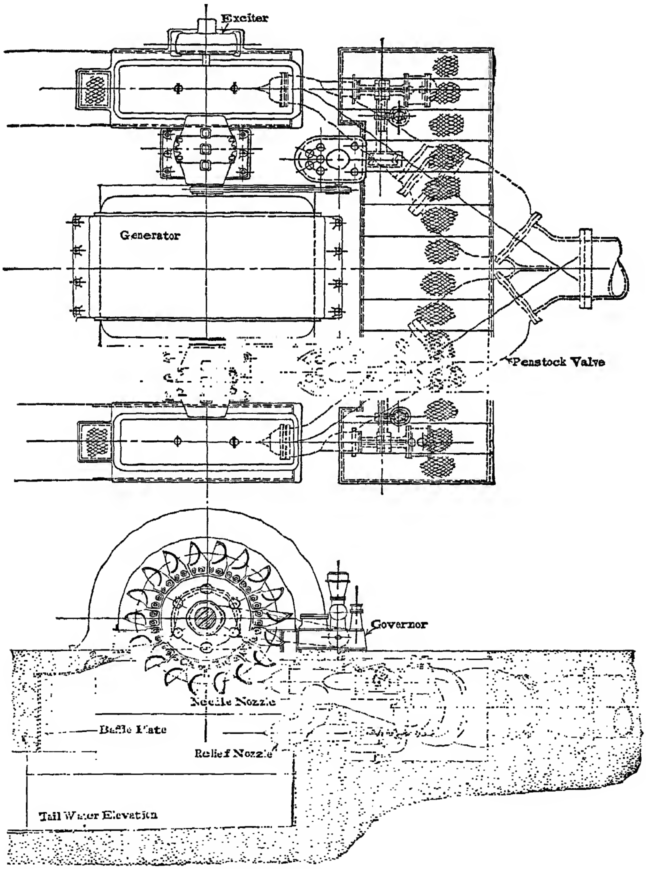


FIG. 2. Impulse Turbine

Table 1.—Values of $(\text{Head})^{5/4}$

Head	0	1	2	3	4	5	6	7	8	9
0	00.00	1.00	2.38	3.95	5.66	7.48	9.39	11.39	13.45	15.59
10	17.80	20.00	22.33	24.68	27.08	29.60	32.00	34.52	37.07	39.67
20	42.32	44.95	47.65	50.37	53.12	55.90	58.71	61.55	64.41	67.30
30	70.21	73.15	76.14	79.09	82.10	85.20	88.17	91.30	94.34	97.46
40	100.6	103.6	106.9	110.2	113.3	116.5	119.7	123.1	126.3	129.6
50	133.0	136.3	139.6	143.0	146.4	149.8	153.2	156.6	160.1	163.5
60	166.8	170.5	174.0	177.5	181.0	184.6	188.1	191.7	195.3	198.9
70	202.5	206.1	209.7	213.4	217.1	220.7	224.4	228.1	231.8	235.5
80	239.3	243.0	246.7	250.5	254.3	258.1	262.2	265.7	269.5	273.4
90	277.2	281.1	284.9	288.8	292.7	296.6	300.5	304.4	308.8	312.3
100	316.2	320.2	324.2	328.1	332.1	336.1	340.1	344.1	348.2	352.2
110	356.2	360.3	364.4	368.4	372.5	376.6	380.7	384.8	388.9	393.0
120	397.2	401.2	405.5	409.6	413.7	417.9	422.1	426.3	430.5	434.7
130	439.0	443.2	447.4	451.7	455.9	460.2	464.4	468.7	473.0	477.3
140	481.6	485.9	490.2	494.5	498.8	503.2	507.5	511.9	516.2	520.6
150	525.0	529.3	533.7	538.1	542.5	546.9	551.3	555.7	560.2	564.6
160	569.1	573.5	578.0	582.4	586.9	591.4	595.8	600.3	604.8	609.3
170	613.8	618.0	622.9	627.4	631.9	636.5	641.0	645.6	650.2	654.7
180	659.3	663.9	668.5	673.1	677.7	682.3	686.9	691.5	696.1	700.8
190	705.4	710.0	714.7	719.4	724.0	728.7	733.4	738.0	742.7	747.4
200	752.1	756.8	761.5	766.2	771.0	775.7	780.4	785.2	789.9	794.7
210	799.4	804.2	809.0	813.7	818.5	823.3	828.1	832.9	837.7	842.5
220	847.3	852.1	856.9	861.7	866.6	871.4	876.3	881.1	886.0	890.8
230	895.7	900.6	905.4	910.3	915.2	920.1	925.0	929.9	934.8	939.7
240	944.6	949.6	954.5	959.4	964.3	969.3	974.3	979.2	984.2	989.1
250	994.1	999.0	1004	1009	1014	1019	1024	1029	1034	1039
260	1044	1049	1054	1059	1064	1069	1074	1079	1084	1089
270	1094	1099	1104	1110	1115	1120	1125	1130	1135	1140
280	1145	1150	1156	1161	1166	1171	1176	1181	1186	1191
290	1196	1202	1207	1212	1217	1222	1228	1233	1238	1243
300	1249	1254	1259	1264	1269	1275	1280	1285	1290	1295
310	1301	1306	1311	1316	1322	1327	1332	1338	1343	1348
320	1353	1359	1364	1369	1375	1380	1385	1390	1396	1401
330	1406	1412	1417	1423	1428	1433	1439	1444	1449	1455
340	1460	1465	1471	1476	1481	1487	1492	1498	1503	1509
350	1514	1519	1525	1530	1536	1541	1546	1552	1557	1563
360	1568	1573	1579	1584	1590	1595	1601	1606	1612	1617
370	1623	1628	1634	1639	1645	1650	1656	1661	1666	1672
380	1678	1683	1689	1694	1700	1705	1711	1716	1722	1727
390	1733	1739	1744	1750	1756	1761	1766	1772	1778	1783
400	1789	1794	1800	1806	1811	1817	1822	1828	1834	1839
410	1845	1851	1856	1662	1867	1873	1879	1884	1890	1896
420	1901	1907	1913	1918	1924	1930	1935	1941	1947	1952
430	1958	1964	1969	1975	1981	1987	1992	1998	2004	2009
440	2015	2021	2027	2032	2038	2044	2049	2055	2061	2067
450	2073	2078	2084	2090	2096	2101	2107	2113	2119	2125
460	2130	2136	2142	2148	2153	2159	2165	2171	2177	2183
470	2188	2194	2200	2206	2212	2217	2223	2229	2235	2241
480	2247	2252	2258	2264	2270	2276	2282	2288	2294	2299
490	2305	2311	2317	2323	2329	2335	2341	2347	2353	2358
500	2364	2370	2376	2382	2388	2394	2400	2406	2412	2418
510	2424	2430	2436	2441	2447	2452	2460	2465	2471	2477
520	2484	2490	2495	2502	2507	2513	2520	2525	2531	2537
530	2543	2549	2555	2561	2567	2573	2579	2584	2591	2597
540	2603	2609	2615	2621	2627	2633	2639	2645	2651	2657
550	2663	2670	2676	2682	2688	2694	2700	2706	2712	2718
560	2724	2730	2736	2742	2748	2755	2761	2767	2773	2779
570	2785	2791	2797	2803	2810	2816	2822	2828	2834	2840
580	2846	2852	2859	2865	2871	2877	2883	2889	2895	2902
590	2908	2914	2920	2926	2932	2939	2945	2951	2957	2963
600	2969	2976	2982	2988	2994	3000	3007	3013	3019	3025
610	3031	3038	3044	3050	3056	3062	3069	3075	3081	3088
620	3094	3100	3106	3113	3119	3125	3131	3137	3144	3150
630	3156	3163	3169	3175	3181	3187	3194	3200	3206	3213
640	3219	3225	3232	3238	3244	3250	3256	3263	3269	3276
650	3282	3288	3295	3301	3307	3314	3320	3326	3333	3339

Table 1.—Values of $(\text{Head})^{5/4}$—*Continued*

Head	0	1	2	3	4	5	6	7	8	9
660	3345	3352	3358	3364	3371	3377	3383	3389	3396	3402
670	3408	3415	3421	3428	3434	3441	3447	3453	3460	3466
680	3473	3480	3485	3492	3498	3504	3511	3517	3524	3530
690	3536	3543	3549	3555	3562	3568	3575	3581	3588	3594
700	3600	3607	3614	3620	3626	3633	3639	3646	3652	3659
710	3665	3671	3678	3684	3691	3697	3704	3710	3717	3723
720	3730	3736	3743	3749	3756	3762	3769	3775	3782	3788
730	3795	3801	3808	3814	3821	3827	3833	3840	3847	3853
740	3860	3866	3873	3879	3885	3892	3899	3905	3912	3918
750	3925	3931	3938	3945	3951	3958	3964	3971	3977	3984
760	3990	3997	4004	4010	4017	4023	4030	4037	4043	4050
770	4056	4063	4069	4076	4083	4089	4096	4102	4109	4116
780	4122	4129	4136	4142	4149	4155	4162	4168	4175	4182
790	4188	4195	4202	4208	4215	4221	4228	4235	4241	4248
800	4255	4261	4268	4275	4281	4288	4295	4301	4308	4315
810	4322	4328	4335	4341	4347	4355	4361	4367	4375	4381
820	4388	4395	4401	4408	4415	4421	4428	4435	4442	4448
830	4455	4462	4468	4475	4482	4489	4495	4502	4509	4515
840	4522	4529	4536	4542	4549	4556	4563	4569	4576	4583
850	4590	4596	4603	4610	4617	4623	4630	4637	4644	4650
860	4657	4665	4671	4678	4684	4691	4698	4705	4711	4718
870	4725	4732	4739	4745	4752	4759	4766	4773	4779	4786
880	4793	4800	4807	4813	4820	4827	4834	4841	4848	4854
890	4861	4868	4875	4882	4888	4895	4902	4909	4916	4923
900	4930	4937	4943	4950	4957	4964	4971	4977	4984	4991
910	4998	5005	5012	5019	5025	5032	5039	5046	5053	5060
920	5067	5074	5081	5087	5094	5101	5108	5115	5122	5129
930	5136	5143	5150	5156	5163	5169	5177	5184	5191	5198
940	5205	5212	5219	5226	5231	5239	5246	5253	5260	5267
950	5274	5281	5288	5295	5302	5309	5316	5323	5330	5337
960	5344	5351	5358	5365	5372	5379	5386	5392	5399	5406
970	5413	5420	5427	5434	5441	5448	5455	5462	5469	5476
980	5483	5490	5497	5504	5511	5518	5525	5532	5539	5546
990	5553	5560	5567	5574	5581	5588	5595	5602	5609	5616

will operate at r.p.m. $\times (1/\sqrt{H})$. For a given head, the power developed varies with the area of the discharge passages, and this in turn varies as the square of the diameter. The latter varies inversely with the speed in r.p.m. That is: $\text{Hp.} \propto d^2 \propto 1 \div (\text{r.p.m.})^2$, whence $\sqrt{\text{Hp.}} \propto 1/\text{r.p.m.}$ or $\text{r.p.m.} \propto (1/\sqrt{\text{Hp.}})$ Therefore, if the runner is reduced to such a size that it will develop 1 horsepower under 1 ft. head, it will operate at a speed of

$$\text{r.p.m.} \times \frac{1}{\sqrt{H}} \times \sqrt{\frac{\text{Hp.}}{H^{3/2}}} = \text{r.p.m.} \times \frac{\sqrt{\text{Hp.}}}{H^{5/4}} = \text{specific speed} = N_s \qquad [2]$$

The head H is always measured in feet.

The horsepower referred to here is the horsepower per runner where the turbine is provided with more than one.

Specific speed is sometimes expressed as the speed at which a turbine runner would operate if reduced to such a size that it would develop one metric Hp. under one meter head.

On this basis, $N_s = 4.45 \times \text{r.p.m.} \times (\sqrt{\text{Hp.}}/H^{5/4})$ . . . . [3]

In this discussion, N_s based on 1 ft. head will be employed. Table 1 gives values of $H^{5/4}$ for values of H from 1 to 999 ft. for convenience in computing specific speeds.

SELECTION OF TYPE.—*Reaction vs. Impulse.*—From the standpoint of efficiency, it is desirable to employ the reaction type. This type is not considered suitable, however, for heads in excess of about 900 ft., and for specific speeds of less than about 12, unless the power to be developed is great, and the water is exceptionally free from abrasive matter. The curve, Fig. 4, indicates, in general, the efficiencies which have been obtained at varying values of N_s, and also indicates the dividing line between the two types of turbines. The impulse turbine efficiencies indicated in Fig. 4 do not allow for the loss in efficiency due to head lost from the center line of the nozzle to tail-water. When the head is comparatively low this loss is perceptible as the lower edge of the impulse turbine buckets must be kept well above the highest tail-water elevation. It is necessary to employ the impulse type in some instances where the head is lower than 900 ft., when the power to be developed is relatively small, as in such a case a reaction turbine (having, necessarily, a higher N_s) would operate at a speed which would be mechanically excessive.

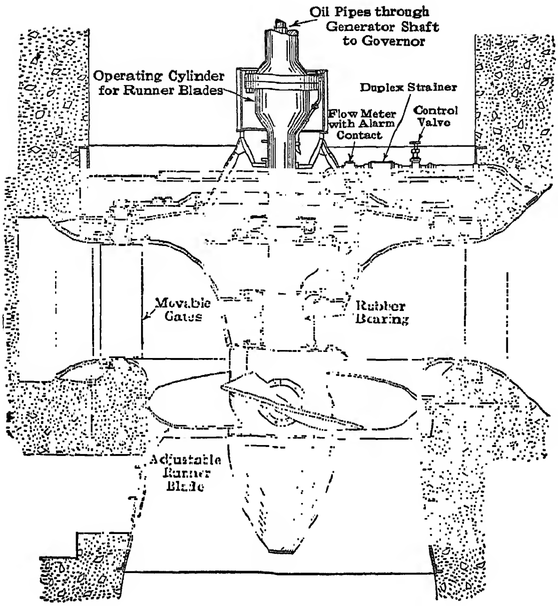


FIG. 3. Reaction Turbine, Kaplan Type

Further, the leakage losses are excessive for reaction turbines of small dimensions where the specific speed is near the dividing line. It is, perhaps, a conservative rule not to allow the speed of turbines, except those of very small capacity, to exceed 1200 r.p.m. For example, suppose it be desired to develop 300 Hp. under a head of 500 ft. For the reaction type, with the minimum value of $N_s = 12$, the resulting r.p.m. $= (12 \times 500^{5/4})/\sqrt{300} = 1640$. It is evident that these conditions call for the impluse type. Also, if the water to be used contains an excessive amount of sharp sand or other abrasive foreign matter, it is often advisable to use the impulse type, even though the other conditions point to the reaction type. This foreign matter is so much more injurious to the latter type than to the former, that the expense entailed by replacement of gates, seal rings, and runners, often exceeds any saving in first cost made by the installation of the reaction type.

SYNCHRONOUS SPEEDS.—Hydraulic turbines are generally direct-connected to alternating-current generators. It is therefore necessary for the turbines to operate at that synchronous speed which is nearest the proper speed from a hydraulic and mechanical standpoint.

The frequency of a generator = (r.p.m./60) × No. of pairs of poles on generator field.

Where a turbine drives an induction generator, the speed should be about 3% above the synchronous value.

2. REACTION TURBINES

SELECTION OF TYPE.—If the conditions outlined in the foregoing pages call for a reaction type turbine, the kind of reaction turbine to be chosen is determined by further consideration of the conditions to be met. A head between 900 and 65 ft. generally indicates the Francis type. For unit capacities above 1000 Hp. the vertical shaft arrangement, Fig. 1, should be used, unless local conditions (*e.g.* equipment to be installed in existing power plant) require a horizontal shaft. For heads above 80 to 100 ft., a metal

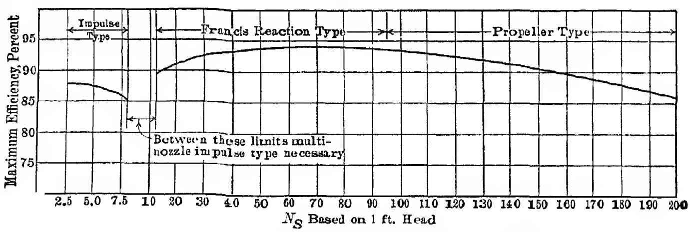


FIG. 4. Efficiency of Various Types of Turbines

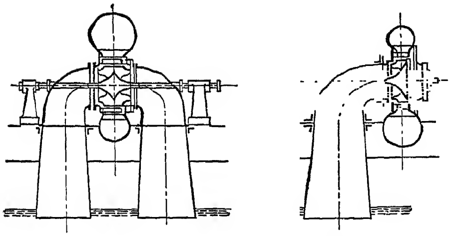


FIG. 5. Horizontal-shaft Reaction Turbine

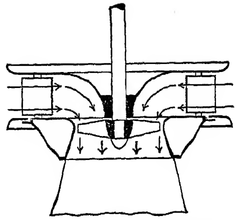


FIG. 6. Vertical-shaft Reaction Turbines

casing is used, of cast steel or plate steel, depending upon the head and size. The horizontal shaft type, Fig. 5(*a*) and (*b*), often is used for these limits of head if the amount of power to be developed is small, it having greater accessibility.

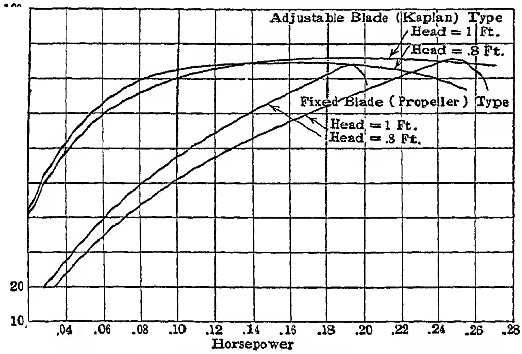


FIG. 7. Comparison of Performance of Kaplan Type and Reaction Type Turbines

Heads below about 65 ft. indicate the propeller type, either with adjustable runner blades, Fig. 3 (Kaplan type) or fixed runner blades, Fig. 6. For run-of-river installations with available head and flow varying, the Kaplan type offers decided advantages in maintaining the rated power output under reduced head, and in maintaining high efficiency under reduced flow. Fig. 7 compares performances of Kaplan type under normal and 80% of normal head, with that of fixed blade type. The marked increase in efficiency for part loads and in the power obtained under the low head are clearly shown. The fixed blade type sometimes is used where the flow and head are fairly constant, or in conjunction with one or more units of the Kaplan type. In the latter case, variations in flow are taken by the Kaplan unit; the fixed blade unit, operates at a more or less uniform load.

For heads below 20 to 25 ft., the siphon setting, Fig. 8, is advantageous. It reduces excavation costs and permits the generator floor to be set above headwater elevation. If headwater level is fairly constant, it may be possible to omit the head gates and to employ stop logs at intake for the infrequent inspections of the casing. Ejectors may be used to exhaust the air from the casing on starting the unit in operation.

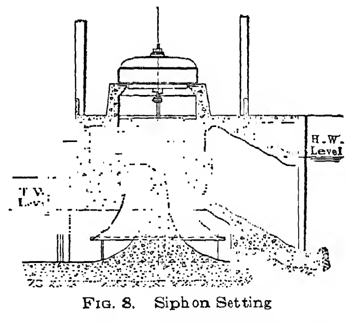


FIG. 8. Siphon Setting

The open flume setting sometimes is used for this range of heads where the power developed is small, and where minimum first cost is decisive. The turbine is completely submerged in an open flume or pit. The disadvantage of this type is the inability to lubricate the operating mechanism properly, with consequent relatively rapid wear. An important feature is the prevention of air vortices. There is no known formula for determining the amount of submergence to prevent these, as the tendency varies with different types of turbine (being greater with horizontal shaft installations) and with different designs of the same type. On this account, and due to poor lubrication, the open flume setting should be avoided where possible.

DETERMINATION OF SPEED.—The curve, Fig. 9, shows usual values of N_s which are being generally adopted at present (1935) for varying values of head H. Local requirements may call for a lower value of N_s than this curve indicates, but higher values should be avoided. The selection of any value of N_s is not only a function of the head H, but of the draft head adopted. Where, for any reason, the latter must be great, the value of N_s should be lowered to give a suitable value of σ. (See page 2–53.) As costs vary inversely with some function of the selected N_s, it is desirable to keep the latter value as high as permissible.

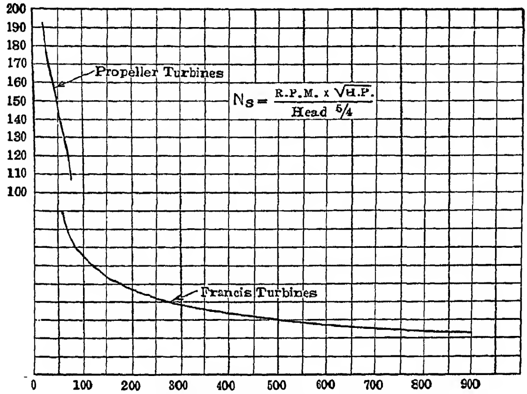


Fig. 9. Values of N_s Generally Used

NUMBER OF UNITS REQUIRED depends upon several considerations, among which are: 1. *Cost.* The cost of the turbines themselves, per Hp. developed, tends to decrease with increasing capacity up to a runner diameter of about 100 in. Above this diameter the cost per Hp. tends to increase with increasing capacity, as the weight and cost increase faster than the horsepower developed. The electrical connections, maintenance, and operating costs, however, tend to decrease with decreasing number of units. Therefore, in general, the smaller the number of units, the lower the overall cost of the development. 2. *Efficiency.* With a given specific speed, the larger the unit capacity the higher the efficiency, as indicated by formula [7]. 3. *Flow Characteristics.* If flow is widely variable, it should be possible to obtain reasonably high efficiency at the minimum value of flow. This may be effected by adopting turbines with high part-load efficiency, such that when only one unit is in operation, minimum flow conditions may be efficiently met; or by selecting a small number of large units, and one small unit of high part-load effi-

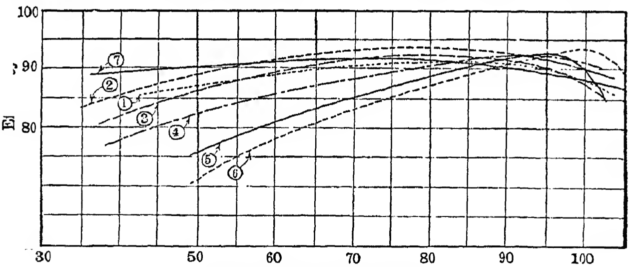


Fig. 10. Typical Characteristic Curves

ciency. The latter scheme has the disadvantage of requiring a more complex plan for operating the units for best efficiency.* 4. *Maximum Unit Capacity Available.* In line with the first consideration, cost, if there is a very great amount of power to be developed, the largest practicable unit capacities should be adopted. No limit of maximum capacity possible of the turbine proper has been reached, but the largest capacities to date (1935) are: Head 48 ft, 45,000 Hp., Wheeler Power Plant, Tennessee Valley Authority (under construction); head 55 ft., 42,000 Hp., Safe Harbor Water Corporation; head 55 ft., 66,000

* See Operating a Hydro-Electric System for Best Economy, E. B. Strowger, *Power Events*, June, 1929, published by Buffalo Niagara & Eastern Power Corp.

Hp., Bonneville Development, U. S. Government (under construction); head 89 ft., 54,000 Hp., Conowingo Development, Susquehanna Power Company; head 116.5 ft., 84,000 Hp., Dneipestroy Development, U.S.S.R.; head 520 ft., 115,000 Hp., Boulder Dam Development, U. S. Government (under construction).

VARIATION OF CHARACTERISTIC CURVES WITH N_s AND TYPE.—Fig. 10 shows a number of typical characteristic curves, plotted between efficiency and percent of rated horsepower, for various values of N_s. These curves show, starting with the lowest N_s, namely 21, how increasing N_s above a value of about 35, has the effect of reducing the part-load efficiency, this effect continuing to the fixed blade propeller type, curve 6, with $N_s = 120$.

The Kaplan type of propeller turbine, with runner blades adjustable during operation, has a very flat curve (7), with high efficiency over wide ranges of horsepower developed and quantity discharged. The part-load efficiencies are even higher than those of the low N_s Francis type turbines.

USE OF MODEL RUNNER TESTS.—The performance of model runners, if correctly interpreted, may be used as a reliable indication of the performance of large units. It is vitally important, however, that in making use of model tests, other considerations than that of the runner alone be taken into account. If the large turbine is to be installed under restricted conditions as regards design of casing and draft-tube, these conditions must be reproduced faithfully in the test of the model runner in order to obtain a prediction of the performance of the large unit. This is particularly important as regards the draft-tube if N_s is high. Hence, to apply these tests to any large installations, the latter should be provided with liberally-designed casings and with modern draft tubes (see p. 2–53.)

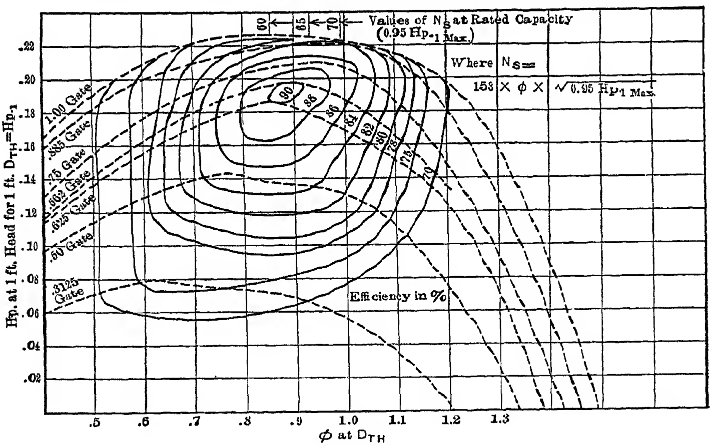


Fig. 11. Model Runner Test Curves

Assume that a water power site is to be developed where the effective head will be 100 ft., and that the unit capacity is to be 10,000 Hp. With N_s = 64, r.p.m. = $(64 \times 100^{5/4})/\sqrt{10{,}000}$ = 202.5. With a frequency of 60 cycles, 200 r.p.m. corresponds to a generator having 18 pairs of poles, and the corrected value of N_s = 63.3. Assume that the test curves shown in Fig. 11 represent performance of a model runner and draft tube that may be stepped up in dimensions for this project. It will be noted that this runner develops a value of N_s = 63.3 at ϕ = 0.891. The term ϕ(peripheral velocity in ft. per sec. ÷ $\sqrt{2gH_0}$), where g = acceleration due to gravity = 32.2 ft. per sec. per sec., and H_0 = effective head on turbine. ϕ is measured at the throat of the runner (see Fig. 3). This value of N_s is not computed at maximum power, but at 0.95 thereof, which corresponds with the rated capacity. The performance curve now may be drawn. For a value of ϕ = 0.891, the corresponding values of efficiency and horsepower, should be tabulated. The throat diameter, D_{th} (in.) of the runner under consideration =

$$(0.891 \times 1836 \times \sqrt{100})/200 = 81.75 \text{ in.}$$

Remembering that Hp. ∝ diam.² $H^{3/2}$, Hp. ∝ for the conditions under consideration = $Hp._1 \times (81.75/12)^2 \times 100^{3/2}$ Hp. = $Hp._1 \times 46{,}500$. At rated capacity as taken from Fig. 11, $Hp._1$ = 0.215, and $0.215 \times 46{,}500$ = 10,000 Hp. This agrees with the rated capacity of the turbine under consideration, thus indicating that the selected value of ϕ is correct. If the head is variable,

Hp.-efficiency curves may be drawn for the values desired by obtaining new values of ϕ and new values of $(D\text{th}/12)^2\,H^{3/2}$. If the minimum head is considerably less than the normal value, a frequent condition at low-level developments, it is desirable that the maximum power possible be developed at the low heads. This is equivalent to stating that the values of Hp._1 at higher than normal should be greater than at normal ϕ.

A complete group of curves, as shown in Fig. 11, is necessary to determine the characteristics of a Kaplan type runner. Each of these curves represents a test at a fixed blade angle. The Hp.-efficiency curve for any value of ϕ is obtained by plotting the Hp. curves for all blade angles at that value of ϕ, and then drawing an envelope curve tangent to each blade angle curve.

Increase in Efficiency with Increase in Runner Diameter.—The formula for stepping up efficiencies of model runners in order to predict that of the large prototype, as developed by Prof. L. F. Moody, has been found to be quite accurate. This formula is

$$E_1 = 1 - (1 - E)\,(D/D_1)^{1/4}\,(H/H_1)^{1/100} \quad . \quad . \quad . \quad . \quad . \quad . \quad . \quad . \quad [4]$$

where E and E_1 = efficiency of large and small runners, respectively; D and D_1 = the diameters, and H and H_1 = the heads acting.

RUNNER PROPORTIONS.—After determining the value of N_s and consequently the r.p.m. of a runner, it is necessary (unless the runner to be constructed is to be stepped up from a small model runner as described above), to select arbitrarily a value of ϕ for the determination of the diameter at the throat and at the vane tips. It is also necessary to select a ratio of e_1/d_{th} (see Fig. 1) in order to fix the former. The curves in Fig. 12 indicate values of ϕ_{th} (at throat), ϕ_1 (at vane tips), and e_1/d_{th} = (distributor width ÷ diameter at throat) for varying values of N_s which have been found by trial to give the most satisfactory results. These values are not to be adhered to rigidly, but are submitted as a guide. The values of ϕ_{th}, ϕ_1 and e_1/d_{th} of a model runner which may be used as the basis of construction of a larger one, fix these values in the case of the larger runner without reference to Fig. 12, except for purposes of comparison. Again, it may be necessary for draft-head reasons (see p. 2–52) to increase somewhat the value of ϕ above that indicated in Fig. 12, thus reducing the throat velocity and increasing the allowable distance that the runner may be placed above tail-water. Fig. 13 gives profiles of runners of varying values of N_s, having proportions obtained from Fig. 12, and drawn to such size that they each would develop 1 Hp. under 1 ft. head. A propeller runner also is shown.

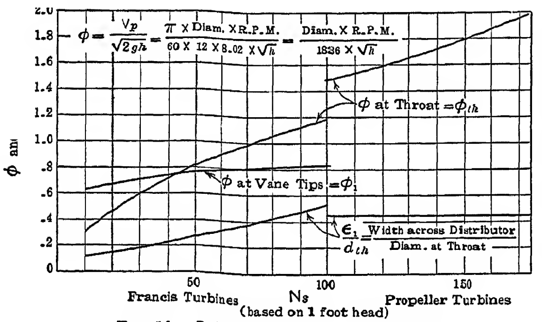


FIG. 12. Relation of ϕ_{th}, ϕ_1, e_1/d to N_s

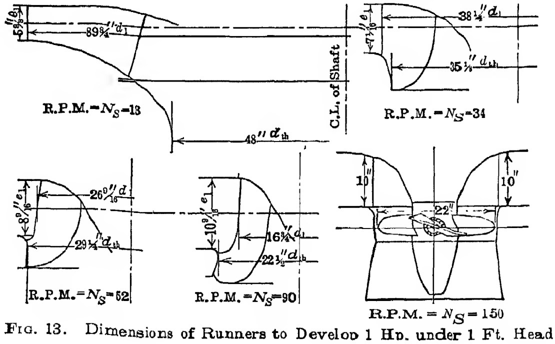


FIG. 13. Dimensions of Runners to Develop 1 Hp. under 1 Ft. Head

The N_s of these runners is therefore synonymous with the r.p.m. as noted. These profiles indicate clearly the desirability of increasing N_s as a means of reducing the cost of the unit.

METHODS OF INCREASING POWER UNDER REDUCED HEAD CONDITIONS.—A characteristic of the majority of low-head developments is the marked reduction in head available with increased flow conditions during flood seasons.

a. For heads above allowable values for propeller type (about 65 ft.). Under these circumstances it is desirable to adopt a runner which develops under higher than normal values of ϕ, a greater power than that at the normal ϕ. As a means of further increasing the power available during low-head conditions, the ejector turbine (Fig. 14) has been devised by L. F. Moody which uses the surplus water as a means of increasing the effective head on the turbine. Fig. 15 shows the increase in power which results from the use of this device. The decreased efficiency during the operation of the ejector as noted is of no consequence since the ejector is used only when surplus water is available. It will be noted that where the head is 0.8 of the normal value, the power with the ejector in operation is very nearly that developed by the turbine at normal head with the ejector closed. The ejector consists of openings provided immediately below the turbine runner from the casing to the draft-tube, which openings are controlled by a cylindrical gate through rods passing up to the operating deck of the turbine.

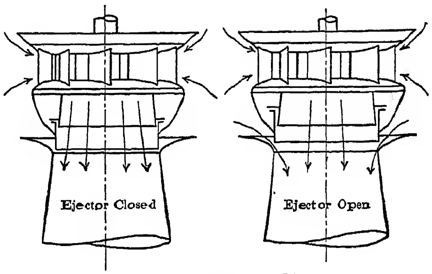


FIG. 14. Ejector Turbine

O. G. Thurlow, Chief Engineer of the Alabama Power Co., has devised a novel means of suppressing the tail-water elevation during flood condition by utilizing the waste water flowing over the spillway. (See J. A. Sirnit, Hydroelectric Power Plant Design, *Trans*. A.S.M.E., xliv, 1922.) This device requires a relatively large amount of waste water, but under these conditions is very effective.

b. For heads permitting use of propeller type (below 65 ft.). Under these conditions the adjustable runner blade or Kaplan type of propeller turbine is decidedly the best. Fig. 7 shows the gain in power as compared to the fixed blade propeller type

3. OUTLINE OF THEORY OF REACTION TURBINE RUNNERS

The design of reaction turbine runners has as its basis the theorem, "The power of a turbine in steady motion equals the angular velocity multiplied by the change of angular momentum experienced by the mass of water flowing in a unit of time in its passage through the turbine." This principle, probably first discovered by Leonhard Euler in 1754, is known as the Eulerian Theorem.

Notation.—All velocities, except where noted, are in ft. per second. Let C_0 = absolute velocity of water leaving movable gates; C_1 = absolute velocity of water at radius r_1, ft.; C_2 = absolute velocity of water at radius r_2, ft.; U_1 = absolute velocity of turbine runner

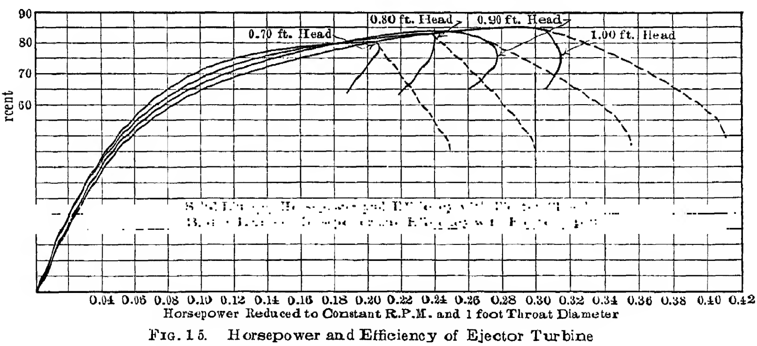


FIG. 15. Horsepower and Efficiency of Ejector Turbine

at radius r_1, ft.; U_2 = absolute velocity of turbine runner at radius r_2, ft.; CU_1 = tangential component of $C_1 = C_1 \cos \alpha_1$; CU_2 = tangential component of $C_2 = C_2 \cos \alpha_2$; C_{m1} = radial component of $C_1 = C_1 \sin \alpha_1$; C_{m2} = radial component of $C_2 = C_2 \sin \alpha_2$; w_1 = velocity of water at r_1 relative to runner; w_2 = velocity of water at r_2 relative to runner; ω = angular velocity of turbine runner, radians per second; M = mass of water discharged by runner per sec. = W/g; Q = quantity discharged, cu. ft. per sec. W = weight of water flowing per sec., lb.; 62.4 = weight of 1 cu. ft. of water, lb.; g = acceleration due to gravity = 32.2 ft. per sec. per sec.; H = head acting on turbine runner; e = hydraulic efficiency of turbine runner; K_1 = moment arm of C_1, ft.; K_2 = moment arm of C_2, ft.

Fig. 16, wherein this theory is applied to the design of a Francis type turbine, shows angular moment at entrance to runner to be $K_1 \times C_1$ and at exit $K_2 \times C_2$.

Power delivered by water to turbine $= P = \omega \times M \times (K_1 C_1 - K_2 C_2)$

$$= \{(\omega \times Q \times 62.4)/g\} \times (K_1 C_1 - K_2 C_2) \quad [5]$$

By similar triangles, $K_1/r_1 = CU_1/C_1$ and $K_2/r_2 = CU_2/C_2$.

Therefore, $$\text{Power} = \{(\omega Q \times 62.4)/g\} \times (r_1 CU_1 - r_2 CU_2) \quad [6]$$

Since (angular velocity × radius) = linear velocity at outer end of radius, $\omega r_1 = U_1$, and $\omega r_2 = U_2$,

and $$P = \{(Q \times 62.4)/g\} \times (U_1 CU_1 - U_2 CU_2) = Q \times 62.4 \times H \times e, \quad [7]$$

whence $$gHe = U_1 CU_1 - U_2 CU_2 \quad [8]$$

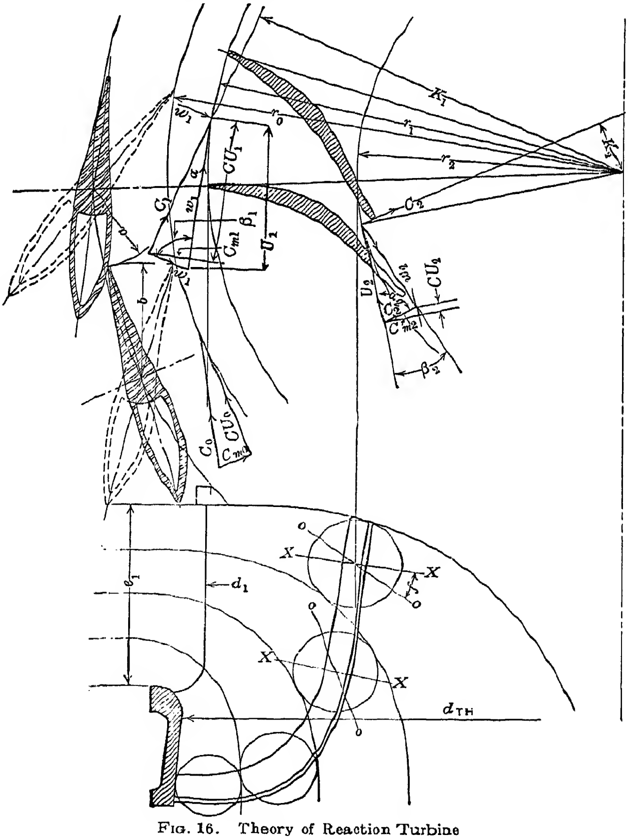


FIG. 16. Theory of Reaction Turbine

Equation [8] may be used as the basis of turbine runner design, for determining the entrance and discharge angles of the runner vanes and also of the movable gates at the entrance to the runner.

Referring to Fig. 17, it will be noted that, except at very low values of N_s, there is a whirl component CU_2 in the direction of rotation at maximum efficiency. At rated capacity, however, there is a whirl component against the direction of rotation for values of $N_s < 50$. The curves shown in Fig. 17 are the results of observations of the flow for turbines of efficient design.

With the controlling data given, *i.e.*, the horsepower to be developed, the effective head, and the r.p.m., the value of N_s may be calculated (see p. 2–39). For the latter value, the values of ϕ_1, ϕ_{th} and e_1/d_{th}, may be found from Fig. 12. For the value of specific speed under consideration, an inspection of the curves in Fig. 10 will show the approximate percentage of the rated turbine capacity at which maximum efficiency will occur. The maximum efficiency which may be attained may be estimated from a consideration of the dimensions of the runner (see formula [4]) and from Fig. 4. The quantity discharged at maximum efficiency may be determined by the formula

$$Q = (\text{Hp.} \times 550)/(H \times 62.4 \times E) = \text{Hp.}/(0.1134 \times H \times E), \quad . \quad . \quad . \quad [9]$$

where H = head acting on turbine, and E = efficiency.

PROFILE OF RUNNER VANES.—The profile of the runner vanes, including the runner band and runner crown, may be laid down as indicated in Fig. 16. The curvature from the lower distributor to the throat of the runner should have as large a radius as practicable, and the flow from the runner band into the top of the draft-tube should be without any sudden break. The shape of the profile should be such as to give sufficient depth to the vanes for proper guidance and flow of the water, without, at the same time, resulting in high friction losses. In particular, with the higher speed runners, where the profile of the vanes at outflow is not at right angles to the direction of flow, the runner should be subdivided into several sections by means of equal quantity flow lines, and the design in each one of these subdivisions should be treated separately. Table 2 may be used as an approximate guide for the number of runner vanes.

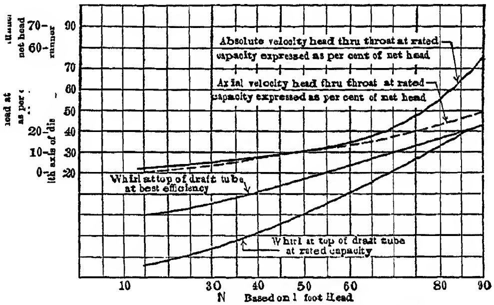


Fig. 17. Amount of Whirl in Draft Tubes

For the specific speed under consideration the amount of whirl in the draft-tube at best efficiency may be taken from Fig. 17. This angle of whirl has been determined by test to be approximately constant at all distances from the center-line of the shaft. Considering the upper subdivision of the runner in Fig. 16, where $(90° - \alpha_2)$ is the angle of whirl, it is seen that $CU_2 = C_{m2} \tan (90° - \alpha_2)$. C_{m2} is the velocity of the water in a vertical plane passing through the axis of the runner, and is determined by the area between the flow lines at radius r_2, allowance being made for the area occupied by the runner vanes. It is then possible to obtain the term CU_1 from the general equation

$$gHe = U_1 CU_1 - U_2 CU_2, \quad . \quad . \quad . \quad . \quad . \quad . \quad . \quad . \quad [10]$$

since all other terms are known. C_{m1} may then be found from the area at the intake to the runner. The values w_2, C_2 and C_{m2} in the plan view of Fig. 16 should be considered as lying in the direction of flow, that is in direction OO. In determining β_2 on the line XX which is at right angles to the surface of the runner vane at outflow, the outflow triangle should be constructed on the basis that C'_{m2} (in the direction XX) $= C_{m2} \times \cos f$.

GATES.—After determining the number of gates and the distance between the center lines of the gate shanks and of the turbine shaft, the radius r_0 may be determined as

Table 2.—Approximate Number of Runner Vanes

	Francis Type					Propeller Type		
N_s.	12–18	18–30	30–45	45–70	70–100	100–130	130–175	175–200
No. of wheel vanes.	19	18	17	16	15	6	5	4

being the radius at which the water leaving the two sides of any gate converges, forming a solid mass. The number of gates is generally between 12 and 20, the number being smaller for mechanical reasons for turbines of smaller proportions. There is no well-defined relation between the number of gates and the value of N_s. From the relation $r_1CU_1 = r_0CU_0$, the value of CU_0 (Fig. 16) can be determined. The value of C_{m0} obviously can also be determined, and hence the value of C_0; the angle and opening of the gates necessary may be determined from the latter value. Additional gate opening must be provided for that portion of the performance curve (see Fig. 10) between maximum efficiency and maximum power. Maximum designed horsepower should be about 5 or 6% greater than the rated or guaranteed value. This margin is necessary to insure the attainment of the rated horsepower under test, to allow for variations between design and actual construction, and for slight inaccuracies in calculations. To design for maximum efficiency at too low a percentage of the rated horsepower is to risk failing to attain the rated capacity. For the application of rational theory to the design of propeller type runners, see W. Spannhake, Problems of Modern Pump and Turbine Design, *Trans.* A.S.M.E., vol. lvi, p. 225, 1934.

4. FEATURES OF TURBINE DESIGN

CASING.—The volute type of casing universally is accepted as the most efficient. The areas of the casing passages surrounding the stay ring (Fig. 18) should be determined on the basis that $CU_x \times r_x = J$, where CU_x = tangential component of velocity of any filament, at radius r_x of filament, measured from the center-line of the shaft, and J = a constant. This results in increasing velocity as the water passes around the casing toward the baffle vane of the stay ring. Casing areas frequently are designed on a basis of constant velocity; the excess area will be occupied by eddies, since the velocity actually increases in accordance with the above law. Hence, an attempt to prevent excessive friction loss may lead to excessive eddy loss. The velocity head at the intake to the casing proper should be roughly 2 to 3% of the effective head on the turbine. This value may, however, be increased without appreciable loss.

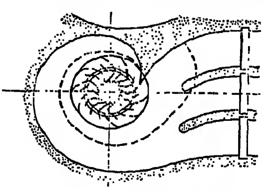


FIG. 18. Volute Casing

STAY RING.—The angle of the stay vanes at the intake may be determined as follows:

The tangential component of velocity at the intake to the vanes is equal to $J/r_g = CU_g$, the constant J being same as that used in the design of the casing and CU_g and r_g being, respectively, the tangential component of velocity at, and the radius of, the stay ring at intake. The radial velocity component obviously can be determined from a consideration of the quantity flowing and the radial inlet area to stay ring. From these two values the proper entrance angle for the stay vanes may be determined. For the true volute casing this angle should be constant for all vanes. The movable gates should be so located in conjunction with the stay ring that the water will have a smooth passage inward at the gate opening giving best efficiency. Attention also should be given to the stream shape at full gate. At low gate openings there is necessarily a certain amount of impact loss due to the gates not coinciding with the stay vanes. This impact is slight, however, since the velocity of flow between the stay vanes and at the outer ends of the gates is low under these conditions.

Turbines of large and medium size often are provided with casings of the partial volute type, due to necessity for reduced unit spacing. The angle of the stay vanes at the upstream portion of such casings is determined by model tests. An efficient angle for these vanes, with the usual design of casing, is 30° with the radial. Usual practice is to provide half as many stay vanes as there are gates.

THRUST BEARINGS.—This part of the hydro-electric unit invariably is supplied with the generator. Two types are in general use, namely, the Kingsbury and the General Electric spring type. Both operate in oil baths under atmospheric pressure, and depend on the viscosity of the oil carrying a wedge-shaped oil film between the rotary and stationary surfaces. Most recent practice (1935), particularly with the larger units, is to have the thrust bearing located below the generator and combined with a guide bearing. This arrangement results in the supporting beams being of reduced length and weight, and in general does away with the need of an upper generator guide bearing.

RUNAWAY SPEED.—Both the generator and turbine parts should be designed safely to withstand the full runaway speed of the turbine, with maximum gate opening and no

load on the generator. For impulse wheels the runaway speed is generally 80% to 90% above normal speed. For Francis turbines of low specific speed it is 65% to 80%; for high specific speed Francis turbines, 80% to 90%, and for the Kaplan type it may be as high as 130% above normal speed. Runaway speeds should be based on the maximum operating head, rather than the normal value.

COMPUTATION OF LOADS ON THRUST BEARINGS.—Allowance must be made for the weight of the turbine runner, turbine shaft, and the amount of hydraulic thrust on the turbine runner in the design of the thrust bearing of the vertical-shaft single-runner turbine. This thrust-bearing also carries the weight of the generator revolving parts. The weight of the turbine shaft readily may be computed. A rough idea of the weight of any Francis runner may be obtained by multiplying the cube of the throat diameter in feet by the constant 35. This applies to cast-iron or cast-steel runners with the vanes and hubs cast integrally. For propeller-type runners with fixed blades, the constant may be taken as 12; for Kaplan-type runners, as 22. This method of determining the weights of a runner, while not accurate, may be sufficiently close for determining the load on the thrust bearing, as the largest factor dealt with is generally the hydraulic thrust. Consequently, a comparatively large percentage of error in the weight of the runner has no great effect on the total estimated thrust-bearing load allowance.

The computation of hydraulic thrust on Francis runners, although somewhat complex, is subject to analysis. In carrying out this process it is necessary to take into account the pressure between the movable gates and the runner; the seal design, and area; the method of venting, and proportions of the runner. Fig. 19 gives a curve for computing the thrust on runners of varying specific speeds, where the runner is vented through a passage in the head cover leading to cored passages through the runner hub.

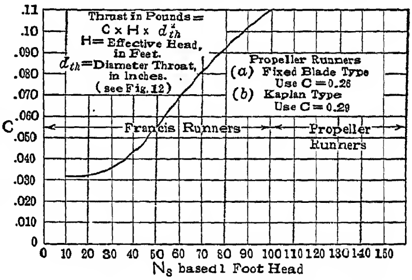


FIG. 19. Thrust on Runners of Varying Specific Speeds

In allowing for the load on the thrust bearing, the weight of the turbine runners theoretically should be reduced by the amount of its loss of weight in water. Actually, however, this reduction of weight and of the load on the bearing is neglected.

RUNNER LOSSES.—Impact and eddy losses in the runner at low gate openings are unavoidable, since the vanes of a Francis runner cannot be made movable. Friction losses can be reduced to a minimum by proper finish of the runner vanes, by proper selection of the number of runner vanes and the amount of vane surface.

The impact and eddy losses in Kaplan runners are reduced effectively by the adjustment of the runner vanes during operation to suit the gate opening. The most serious loss to contend with in this type of runner is that due to friction on the vane surfaces caused by the necessarily high values of ϕ, and resulting high relative velocities. On this account, the proper finishing of the vane surfaces is extremely important, and recourse to machine finishing is frequent.

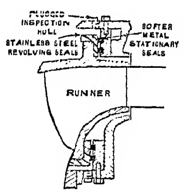


FIG. 20. Labyrinth Seals

Leakage loss around a Francis runner is greatest for low values of N_s, and *vice versa*, since this varies with a function of ϕ/N_s on a basis of uniform seal clearances and design of seals. This loss effectively can be reduced, theoretically, by using labyrinth seals. In practice, elaborate designs of this type of seal are not in wide use, due to the destructive effects of contact between the rotating and stationary seals caused by bearing wear or incorrect alignment. Present practice (1935) for large turbines of low specific speed embodies the use of rotating seal rings on the runner, preferably of stainless steel, in conjunction with stationary seals of a dissimilar softer metal. Very small clearances are used with this design, as indicated in Fig. 20, and if actual contact does occur, the softer metal wears away locally with no injurious generation of heat.

The use of stationary and revolving seals of steel is not good practice, as contact results in generation of excessive heat with consequent tearing, rolling, and welding of the metal.

The friction of the water surrounding the runner against the rotating external surfaces of the runner causes a loss known as disc loss. This may be maintained at a minimum

by finishing external runner surfaces as smoothly as possible, and by reducing the space between the runner and adjacent parts to the lowest practicable value. This loss, like loss from leakage, is greatest at low values of N_s. Experiments which have been made on machined brass discs give a value of K of 0.000,000,000,413 in the formula $\mathrm{Hp}._{df} = KD^5N^3$, where $\mathrm{Hp}._{df}$ = horsepower loss due to disc friction, D = diameter of disc, or runner, ft.; and N = revolutions per minute. This formula takes care of the loss on both sides of the disc. No experiments have been made in this connection on turbine runners but the formula may be applied thereto for comparative purposes. This loss is constant at all loads, provided the speed is constant, and, of course, affects the efficiency at part load to a greater extent than at full load. The experiments on brass discs, cited above, indicated greater loss when the disc was in a large chamber than in a restricted chamber with smooth surfaces close to the disc. Therefore, less disc loss is encountered in a turbine when the stationary surfaces are smooth and are in close proximity to the external surfaces of the runner.

The practice of providing ribs between the head or side cover and the turbine runner is bad, as these ribs merely cause eddies to be set up between them. Such eddies absorb more energy than a large body of water surrounding the runner will absorb if allowed to rotate freely. It is worthy of note that the rapidly mounting leakage and disc losses at the low values of N_s form the economic reason for changing from the reaction to the impulse type of turbine for values of N_s lower than about 12.

WATER PASSAGES.—It is important that all water passages, from intake to tail-race, receive attention, as an undue loss at any point between these limits may result in poor plant efficiency even though the turbine efficiency be exceptionally high. The velocity through the racks at the intake to the penstocks should be not more than from 2 to 3 ft. per second, as the loss there may be a perceptible proportion of the total head when the latter is low. The spacing of the rack bars should be as wide as possible and should be governed by the minimum opening between the turbine runner vanes. A well-proportioned bell-mouth should be provided to lead the water to the penstocks.

The head gate guides should not offer obstruction to flow, and the gates, when open, should be entirely out of the path of the flowing water. The gates should be as large as practicable to reduce to a minimum the number of supporting piers and the losses caused thereby. With long penstocks a careful balance should be made between the initial cost of various diameters and the loss of revenue due to corresponding penstock losses. The question of pressure changes and regulation also must be considered here (see p. 2–55). Attention should be given to the excavation of the tail-race at the discharge from the draft-tubes to prevent undue loss from unnecessarily raising the tail-water level.

CAVITATION.—In any water passage, not occupied by steadily flowing water, eddies of rapidly whirling water are formed. When the head or pressure, acting on this water passage, is reduced to that of vapor pressure (about 1.25 ft. above absolute zero pressure at the usual water temperature), vortices exist in these areas. Voids or cavities form in the center of the vortices, causing what is known as cavitation. Under such conditions, slight changes in static pressure or in velocity of flow, with resultant changes in pressure, cause the alternate formation and collapsing of these cavities. The latter phenomena are accompanied by intense local water hammer, with the formation of high local momentary pressure. If these cavities collapse on the surface of runner blades or draft tubes, the pressure generated tends to enter the microscopic cracks, causing *pitting*.

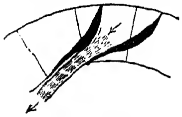


Fig. 21. Cavitation

It thus is important to so design all water passages as to avoid areas where eddies tend to form, as indicated on the back of the runner vane in Fig. 21, and to place the runner sufficiently close to tail-water level. Pitting practically always occurs on the back or underside of the vanes of vertical-shaft type turbines, as this is the low pressure side. The face of the vanes, which receives the reactive force in the form of higher pressure, is relatively much less susceptible to pitting.

High head, low specific speed turbine runners, are, in general, less susceptible to cavitation than are high specific speed, low head runners. Of all types, propeller runners are most susceptible, due to their high relative velocity and small blade area.

ALLOWABLE HEIGHT OF TURBINE ABOVE TAILWATER is one of the most important dimensions in power-house design. Numerous installations exist which have been all but ruined by fixing the runner at an excessive distance above tailwater. As a result, in such cases, excessive pitting and vibration have occurred, with heavy maintenance costs and undue limitations in power developed. Although a given turbine, operating under a high head, must be placed closer to tailwater, than when operating under a low head, it is the low head plants, *i.e.*, below about 60 ft., that are most often in difficulties,

due to excessive draft head. This is due to the use of higher specific speeds with higher relative velocities at the low heads.

L. F. Moody, in a paper at 3d American Hydro-electric Conference, Engrs. Club of Philadelphia, Mar. 10, 1925, presented formula [11] for the absolute pressure at the point of minimum pressure on the discharge side of the runner when a turbine is in operation.

$$H_m = H_a - H_s - \{(E_d C_2^2/2g) + K_c H\}, \quad . \quad . \quad . \quad . \quad . \quad [11]$$

where H_m = absolute pressure at point of minimum pressure for discharge side of runner; H_a = atmospheric pressure head at elevation of runner above sea level, ft.; H_s = static draft head or elevation of runner above tail-water, measured at the throat of a Francis runner, and at the center line of the blades of a propeller runner, ft.; E_d = draft tube efficiency; C_2 = absolute discharge velocity from the runner, ft. per sec.; H = total effective head on the turbine, ft.; K_c = cavitation coefficient.

If we combine $\{(E_d C_2^2/2g) + K_c H\}$ into a single term σH (Dr. Thoma's notation) we have $H_m = H_a - H_s - \sigma H$, or $H_s = H_a - H_m - \sigma H$.

Therefore,

$$\sigma = (H_a - H_m - H_s)/H \quad . \quad . \quad . \quad . \quad . \quad . \quad . \quad . \quad . \quad . \quad [12]$$

The value H_m must be not less than the vapor pressure of water at the temperature existing. The absolute value of H_m varies within narrow limits, and virtually is always between 0.50 and 1.40.

The determination of H_s by this formula requires a value of σ for the particular specific speed, runner and draft tube design under consideration. Therefore, for the exact determination of σ, cavitation tests of a model, which is a small reproduction of the turbine to be used, are necessary. These tests are not required, however, where it is practicable to provide an ample margin in the value of σ selected. High head installations permitting very low values of σ, and very low head installations, come under this classification.

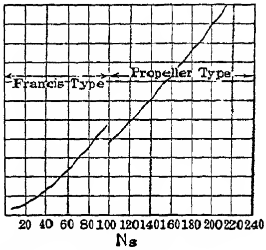


FIG. 22. Relation of σ and Specific Speed

The curve drawn between σ and specific speed, Fig. 22, is based on usual practice. It may be used as a general guide in the absence of cavitation tests. These, however, should be made for confirmation if the propeller type of turbine is used for heads above 30 ft., particularly if local conditions require the use of a low value of σ.

DRAFT TUBES permit the conservation of potential energy referred to tailwater, thus allowing the runner (when the value of σ permits) to be placed some distance above tailwater level. Draft tubes also serve to regain most of the kinetic energy in the water leaving the runner, thus making possible good efficiencies with high outflow velocities from the runner. Inefficient draft tubes have the doubtful virtue of permitting runners to be placed at a higher elevation of tailwater than otherwise would be possible without pitting, this protection of the runners being necessarily accompanied by loss of efficiency of the turbine as a whole.

Two general types of draft tubes are in use, the symmetrical and elbow types. The symmetrical type may be in the form of a straight cone, generally of steel plate, or of the

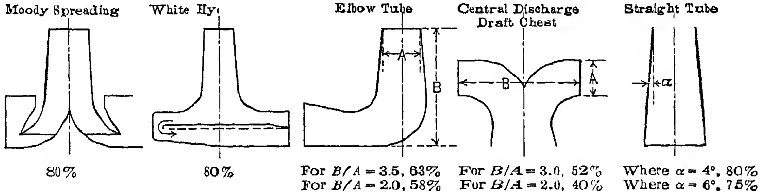


FIG. 23. Types of Draft Tubes

Moody spreading type or the White hydraucone type. All of these symmetrical tubes are capable of developing high efficiency if correctly designed. The elbow type for vertical turbines has, of late years, been developed to high efficiency. Fig. 23 shows the forms mentioned; also the central discharge draft tube, which is a form of the elbow tube.

For the elbow type, B/A should not be less than 2.00; for the central discharge draft chest, not less than 4.0 for good efficiency; for the straight conical tube α should not be greater than 6 to 7 deg.

5. TURBINE TESTS

THE TESTING OF HYDRAULIC TURBINES to determine their efficiency involves the measurement of the work done by the water on the turbine, or the water horsepower, and of the mechanical work done by the turbine, or the developed horsepower.

$$\text{Water horsepower or W.Hp.} = (Q \times 62.4 \times H_0) \div 550 = 0.1134 \times Q \times H_0,$$

where Q = cu. ft. of water passing through the turbine, per second, and H_0 = effective head, ft., acting on the turbine.

Q and H_0 may be measured as discussed below.

Developed horsepower or D.Hp. = $2\pi GAN/550$, if measured by an absorption dynamometer (generally a Prony friction brake), where G = force, lb., exerted by turbine at end of brake arm of length = A, ft., and where N = r.p.m./60.

Developed Hp. = (kw. developed by generator) ÷ (generator efficiency), where the turbine is, as is generally the case, connected directly to a generator. The method of arriving at D.Hp. in this case is described more in detail below. Turbine efficiency is, of course, D.Hp./W.Hp.

See Test Code for Hydraulic Power Plants and Their Equipment, approved by A.S.M.E., 1927, and later editions.

MEASUREMENT OF POWER OUTPUT.—For turbines direct-connected to electrical generators the power output of the turbne may be measured as provided below:

When practicable, the generator is to be separately excited during both turbine and generator tests and the excitation loss is not to be included in computing the generator efficiency. It is, therefore, also be be omitted in computing the turbine output during the turbine test. When determined by the separate-loss method, the generator efficiency in the case of polyphase alternators, when separately excited, is to be taken as

$$\frac{(\text{Kilowatt output at generator terminals})}{\begin{pmatrix}\text{Kilowatt}\\ \text{output}\end{pmatrix} + \begin{pmatrix}I^2R\\ \text{armature}\end{pmatrix} + \begin{pmatrix}\text{Open circuit}\\ \text{core loss}\end{pmatrix} + \begin{pmatrix}\text{Stray load}\\ \text{losses}\end{pmatrix} + \begin{pmatrix}\text{Generator wind-}\\ \text{age and friction}\end{pmatrix}},$$

all losses being expressed in kilowatts.

Stray load losses are to be determined in accordance with Paragraph 458 of the Standardization rules of the A.I.E.E.

MEASUREMENT OF POWER INPUT OR WATER HORSEPOWER.—The effective head on the turbine is to be taken as the difference between the elevation corresponding to the pressure in the penstock near the entrance to the turbine casing, and the elevation of the tail-water, the difference being corrected by adding the velocity head in the penstock at the point of measurement, and subtracting the residual velocity head at the point of measurement in the tail-race. When turbines are set in an open flume, the head is measured by gages located immediately above the center of the turbine and by gages in the tail-race. The effective head on such a turbine is to be taken as the difference between the elevation of the free water surface immediately above the center of the turbine and the elevation of the tail-water, the difference being corrected by subtracting the residual velocity head at the point of measurement in the tail-race.

MEASUREMENT OF QUANTITY OF WATER.—The description of the various methods that may be used, *i.e.*, by weir, salt-velocity method, Pitot tube, etc., are of too great length for reproduction here. Reference is made to the test code. See also *Trans.* A.S.M.E., xlv, 1923, for description of salt-velocity method and of Gibson method. See also p. 2–28.

6. SPEED REGULATION OF HYDRAULIC TURBINES

TURBINE GOVERNORS.—A governor designed for the automatic speed regulation of a hydraulic turbine is not as simple as one for the regulation of a steam engine. The inertia, friction, and hydraulic load acting against the movement of the turbine gates necessitate the introduction of a force external to, but controlled by, the governor for overcoming these resistances. A pump generally is used to force oil under pressure into one or two operating cylinders which actuate the turbine gates.

Fig. 24 is a diagrammatic sketch of a turbine governor. When an increase in speed takes place, the revolving flyballs T cause the right-hand end of the lever L to move upward, and open the ports of the governor valve, thus admitting oil under pressure from A to C and simultaneously connecting B with D. A gate-closing movement of the piston of the operating cylinder results, continuing until the restoring rod E has raised the left-hand end of the lever L sufficiently to close the ports of the governor valve

by lowering the right end of L. During the above operation the right end of L' may be considered as fixed. This leaves the unit at a higher speed than the normal value, but with the ports of the governor valve momentarily closed and the gates momentarily stationary. The spring F, however, is compressed since the left end of L' has moved upward from its original position. This compression causes the lever L' now to move downward slowly at the left end at a rate determined by the dash pot by-pass, L', for the time being, turning about a pivot formed by the upper end of E. This results in an upward movement of the right end of L with consequent closing movement of the turbine gates. The load on the turbine now being steady, the closing movement of the gates results in a reduction in speed until the spring F has reached its normal position, which is necessarily accompanied by normal speed and closed ports in the governor valve. Thus, following a change in load, the functioning of the governor is divided into two distinct processes. The first involves the movement of the gates to supply or cut off the necessary amount of hydraulic energy to suit the new load; the second involves the restoration of speed to the normal value necessitating an additional (but small) movement of the gates.

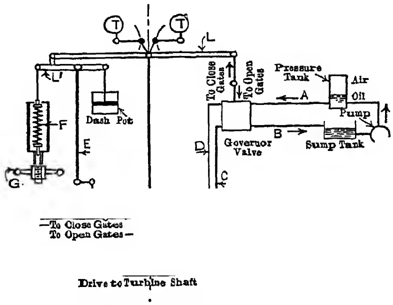


FIG. 24. Diagram of Hydraulic Turbine Governor

The hand wheel G permits the case containing spring F to be raised or lowered, and forms a means of controlling the normal speed of the unit.

DETERMINATION OF GOVERNOR CAPACITY.—This is a function of the hydraulic load on the movable gates, considering all positions of opening, with an allowance for friction and for the power required to accelerate the moving parts. To make a close determination it is necessary to have detailed data relative to the proportions of the turbine and gates. Not only does the relation between the runner diameter and gate circle diameter affect the governor capacity, but the number of gates, their proportions, height, and angular turn as well. The curve, Fig. 25, will serve as a very approximate means of determining the governor capacity required for the movement of the turbine gates. This curve is plotted between unit governor capacity and specific speed. There is indicated in this figure the formula for arriving at the approximate governor capacity for any given conditions of head and N_s. The values for very large turbines tend to be materially smaller than this curve indicates, due to relatively less friction and large number of gates used.

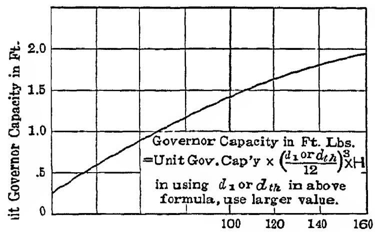


FIG. 25. Governor Capacity Required

The above curve is for use where the operating mechanism is lubricated, as on the outside type (Fig. 1). When the mechanism is submerged, and is lubricated by water only, the values from the above curve should be increased by about 1.25. This curve does not include allowances for relief valves, nor for capacity required for runner vanes of Kaplan type turbines.

REGULATION FOLLOWING SUDDEN LOAD CHANGES.—It is evident that regulation by means of a governor, similar to that described above, is one of degree only, since a change in speed is necessary before the governor will act. The inertia of the water in the turbine water passages, including penstock and draft-tube, prevents a prompt increase or decrease in the energy supplied to the turbine as called for by the governor, thus aggravating the speed change which occurs. The change in speed as the result of a given sudden load change is a function of (1) the inertia effect, or WR^2 of the rotating element of the unit; (2) the inertia effect of the water passages; (3) the relation between time and gate movement.

It should be noted that with the above values and relations fixed, and neglecting the effect of lost motion, all makes of governor will give a uniform degree of speed regulation.

Let N = r.p.m. of unit before gate movement. This is generally the normal or the synchronous value at full load, and from 3% to 5% higher at no load. This difference

in speed is known as the inherent speed change, and is necessary in order that the percentage of load on turbines operating in parallel may be maintained by the governors at about the same value. Therefore, N for loads going off is lower than for loads going on; however, since the term speed regulation generally relates to the percentage of momentary speed change succeeding a load change, the fact that the speed returns to a value of N which is slightly higher or lower than N before the load change may be neglected.

Let N_1 = maximum or minimum r.p.m. succeeding the load change; WR^2 = product of weight of revolving parts of unit (including generator) and the square of the radius of gyration, ft.; H_0 = head acting on turbine before load change; L = length, ft., of enclosed water passages of turbine = sum of L_P (penstock) + L_c (casing) + L_d (draft tube); h = average (as distinguished from maximum) change in head, in ft., during time T, caused by inertia of water in passages; V = average velocity, ft. per sec., in enclosed water passages of turbine = $(L_P V_P + L_c V_c + L_d V_d)/L$; T = time, seconds, for gate movement; Hp._1 = load on turbine before gate movement; Hp._2 = load on turbine after gate movement; a = velocity of pressure wave in penstock, ft. per sec. In order to avoid danger of penstock collapse, T for load increases should be greater than $L_P V_P/12H_0$ and should also be greater than $2L_P/a$. (See page 2–59 for maximum pressure changes in penstocks.) With T in the above limits, h may be considered as LV/gT. Where the profile of the penstock departs appreciably from a straight line between intake and turbine, a greater value of T than otherwise is necessary, and a careful study should be made to avoid collapse.

Consider a sudden load increase, and neglect the time interval between the load change and the beginning of gate movement. The load demand in ft.-lb. during T is $\text{Hp.}_2 \times 550 \times T$. This is supplied partly by hydraulic energy acting on the turbine runner and partly by energy given up by WR^2 of the revolving element in slowing down from N to N_1.

The energy of the rotating mass $= MV^2/2 = WV^2/2g = \{W(2\pi RN/60)^2\}/2g = WR^2N^2/5870$. For a reduction of speed from N to N_1, the energy given up by the revolving mass is $\{WR^2(N^2 - N_1^2)\}$

It is not possible to determine exactly the amount of hydraulic energy supplied to the turbine during transition without analyzing the changes in quantity, head, efficiency, and gate opening, by subdividing T into smaller intervals. The method described by E. B. Strowger and S. L. Kerr in Speed Changes of Hydraulic Turbines for Sudden Changes of Load (*Trans.* A.S.M.E., xlviii, 1926), will give an accurate determination of this energy.

For practical purposes this hydraulic energy may be determined approximately as follows: With an absence of enclosed water passages we may consider that the average energy supplied during transition $= 1/2\,[550 \times T \times (\text{Hp.}_1 + \text{Hp.}_2)]$. If we consider the presence of enclosed passages, the effective head during transition $= (H_0 - h)$, and Q, average, is reduced in the ratio $\sqrt{(H_0 - h)/H_0}$; then the average energy supplied during transition $= 550 \times T \times 1/2\,(\text{Hp.}_1 + \text{Hp.}_2) \times \{(H_0 - h)/H_0\}^{3/2}$. Then we have

$$\text{Hp.}_2 \times 550 \times T = \frac{WR^2(N^2 - N_1^2)}{5870} + 550 \times T \times \left(\frac{\text{Hp.}_1 + \text{Hp.}_2}{2}\right)\left(\frac{H_0 - h}{H_0}\right)$$

or
$$N_1 = \sqrt{N^2 - \frac{\left[2\text{Hp.}_2 - (\text{Hp.}_1 + \text{Hp.}_2)\left(\frac{H_0 - h}{H_0}\right)^{3/2}\right] \times 1{,}620{,}000 \times T}{WR^2}} \quad . \quad . \quad [13]$$

Similarly
$$WR^2 = \frac{\left[2\text{Hp.}_2 - (\text{Hp.}_1 + \text{Hp.}_2)\left(\frac{H_0 - h}{H_0}\right)^{3/2}\right] \times 1{,}620{,}000 \times T}{N^2 - N_1^2} \quad . \quad . \quad . \quad [14]$$

Similarly, for sudden load decreases: The new load on the turbine is equal to the average hydraulic energy supplied during transition minus the energy absorbed by the rotating mass during increase in speed from N to N_1.

Or, transposing,

$$\frac{WR^2(N_1^2 - N^2)}{5870} = \frac{(\text{Hp.}_1 + \text{Hp.}_2)}{2}(550 \times T) \times \left(\frac{H_0 + h}{H_0}\right)^{3/2} - \text{Hp.}_2 \times 550 \times T$$

and
$$N_1 = \sqrt{N^2 + \frac{\left[(\text{Hp.}_1 + \text{Hp.}_2)\left(\frac{H_0 + h}{H_0}\right)^{3/2} - 2\text{Hp.}_2\right] \times 1{,}620{,}000 \times T}{WR^2}}, \quad . \quad [15]$$

or
$$WR^2 = \frac{\left[(\text{Hp.}_1 + \text{Hp.}_2)\left(\frac{H_0 + h}{H_0}\right)^{3/2} - 2\text{Hp.}_2\right] \times 1{,}620{,}000 \times T}{N_1^2 - N^2} \quad . \quad . \quad . \quad . \quad [16]$$

While the above formulas do not exactly take into account all factors, they have been found to be fairly close in actual practice. These formulas contain assumptions which are not correct, but apparently the latter compensate each other to a great extent. The regulation secured in practice is closer than computed by the above method, when the WR^2 of the various motors driven by the unit under consideration is not added to that of the generator. The speed change for increased loads is greater than for decreased loads. The reason is that in the former case, the increased hydraulic losses due to momentary operation at an inefficient relative speed, as well as the friction losses, aggravate or increase the speed change; whereas for decreased loads, these factors decrease the speed change and improve regulation. It will be noted, however, that consideration of the efficiency change and friction loss do not appear in the above formulas. It is generally the case that speed change for decreased loads is, in actual practice, of the greater importance, since loads usually are rejected more suddenly than they are taken on.

The time of gate movement T employed should never be less than about 1.25 seconds for complete travel, and should be slower than this with long penstocks in order that $LV/HT \leqq 12$, and also in order that $T > 2L/a$. If the latter requirements are fulfilled, T, in seconds, for governors should be, roughly, not less than

$$1.25 + (\text{Ft.-lb. of governor}/150{,}000).$$

The time of gate movement for part travel is less than for full travel, although the average rate is slower than for full travel. The following figures may be used as a guide:

Percent load change	100	75	50	25	10
Time of gate movement in percent of time required for full travel	100	87	72	57	48

It is usually allowable to have the speed change 12 to 14% for half the rated turbine capacity rejected, although some loads require closer regulation. A usual value of $WR^2\,N^2/\text{Hp.}$ is from 5,000,000 to 10,000,000. These values generally are exceeded where the value of H_0 is high.

INERTIA DUE TO PENSTOCK LENGTH.—The problem of satisfactorily allowing for the inertia due to excessive penstock length may be taken care of in three ways: 1. By providing a synchronous relief valve in conjunction with the turbine, which prevents velocity change in the penstock when the load on the turbine is changed. 2. By increasing the time of operation of the governor, in this way reducing the pressure change. 3. By the introduction of a surge tank as near the power-house as possible. This prevents sudden changes in velocity in that portion of the pipe line between the surge tank and the intake.

The provision of synchronous relief valves is employed frequently in the West, particularly where irrigation requirements necessitate a constant discharge from the turbine. At small load requirements there is, of course, a correspondingly large waste of water which is by-passed through the relief valve without doing effective work. This type of relief valve is connected directly to the operating ring of the turbine in such a way that a closing of the turbine gates results in a corresponding opposite movement of the relief valve, the sum of the discharge through the relief valve and the discharge through the turbine being a constant. Water-saving relief valves often are provided, which close at a slow rate after having been opened by the governor. While this type of relief valve prevents excessive pressure rises for sudden load decreases, it does not avoid the danger of penstock collapse for sudden load increases and for this condition, it is necessary that the time of governor operation be limited to such a value that LV/HT is not greater than about 12 and that $T>2L/a$. It is extremely important where a relief valve is employed, that its operation be positive. It is preferable that a rigid connection be provided between the valve and the operating ring, so that should the valve become stuck for any reason, the turbine gates also will be prevented from moving in the closing direction. Otherwise, should the relief valve become deranged, the turbine gates might close with disastrous results.

The second solution mentioned, *i.e.*, that of increasing the governor time, requires that means be provided to prevent the possibility of the governor ever operating at a faster rate than that decided upon. It often is not practicable to employ this method, since the flywheel effect resulting from the comparatively long time of gate operation is too great to be taken care of in the generator rotor, and the awkwardness of employing a separate flywheel frequently necessitates the final adoption of either the first or third methods mentioned.

Where practicable, the third solution, *i.e.*, the use of a surge tank, is the most to be desired. The adoption of a surge tank permits quick gate movements in both directions, without adding to the maintenance costs of the turbines by providing the additional

moving parts that are embodied in a relief valve, and without requiring any waste of water for sudden load rejections. The different types of surge tanks are discussed below.

SURGE TANKS.—Two types of surge tanks are in use: 1. The simple tank, consisting of a cylindrical tank with a pipe connection to the conduit; 2. The Johnson differential tank invented by R. B. Johnson. This tank differs from the simple tank primarily by the addition of a riser in the center of the tank proper. At the base of the riser, an annular port communicates with the tank, the port area being proportioned to suit the conditions under which the tank is to operate. See Fig. 26. With a simple tank, when the load is thrown on, the water level falls gradually to meet the increased demand prior to the acceleration of the water in the conduit between the tank and the intake. Since the water level in the tank falls comparatively slowly, there is only a correspondingly small head created to accelerate the water in the conduit to the new velocity required. With the differential tank, the water falls rapidly in the riser to meet the increased demand, establishing in a few seconds a relatively large accelerating head on the conduit. The level in the tank surrounding the riser falls slowly, supplying the demanded increment of water through the ports at the base of the riser. In the simple tank, the corrective action on the conduit velocity accumulates only as fast as the water level changes in the tank. Hence, the duration of the surge is much prolonged and the tank must be made larger. That is, for the same regulating ability, the differential tank can be made of smaller diameter and height than the simple tank.

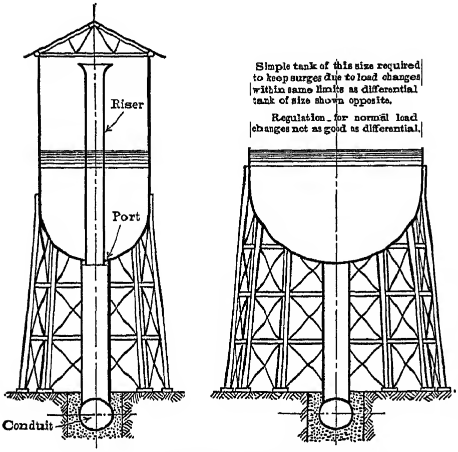


Fig. 26. Differential and Simple Types of Surge Tanks

Fig. 27 shows a comparison of the action of different types of surge tanks, as affecting the water level in the tank and the velocity in the conduit. The upper curves indicate the change of level in the tank and the lower group of curves, the conduit velocity. The time in seconds is reckoned from the instant at which the load is thrown on the turbine, followed practically immediately by the opening of the turbine gates. The curves for the restricted orifice type of tank refer to a form of differential tank, having, instead of a riser, restricted ports at the base of the tank. On account of the restricted openings this type gives relatively great pressure changes at the turbine, and is infrequently used on account of poor governing qualities.

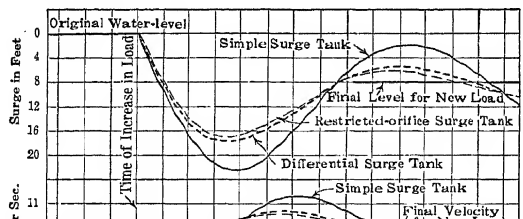


Fig. 27. Comparison of Action of Different Types of Surge Tanks

DESIGN OF SURGE TANKS.—The use of surge tanks to regulate conduit flow is involved and cannot be reproduced here. Reference is made to the following bibliography dealing with this subject: R. D. Johnson, Surge Tanks for Water Power Plants (*Tran-*

A.S.M.E., xxx, p. 443, 1908); by the same author, A Differential Surge Tank (*Trans.* A.S.C.E., lxxviii, 1915); D. W. Mead, Water Power Engineering, Chapter XIII, in which is set forth treatment of simple type of tank; Durand, Control of Surges in Water Conduits (*Trans.* A.S.M.E., xxxiv, 1912); Jakobuss, Surge Tanks (*Trans.* A.S.C.E., 1922). Fig. 27 is reproduced from Hydro-electric Handbook by Creager & Justin.

Maximum Pressure Changes in Penstocks Due to Gate Movements

Notation.—Let a = velocity of pressure wave along pipe, ft. per sec. (See Fig. 29); A = cross-sectional area of penstock, sq. ft.; g = acceleraton due to gravity, ft. per sec. = 32.2; h = pressure rise or excess head above normal, ft., also = pressure drop below normal, ft.; $h_{max.}$ = pressure rise due to instantaneous closure = aV_0/g, ft.; H_0 = initial steady head near turbine gates, corresponding to V_0, ft.; K = pipe line constant = $h_{max.}/2H_0 = aV_0/2gH_0$; L = length of penstock to forebay, or other point of relief, ft.; N = time constant or number of $2L/a$ intervals in time of closure = $aT/2L$; P = pressure rise as a proportion of $h_{max.} = h/h_{max.} = h \div (aV_0/g) = gh/aV_0$; Q_0 = initial steady flow in pipe prior to start of gate closure, corresponding to H_0, cu. ft. per sec.; T = time of gate movement for complete closure, if rate is uniform, sec. If rate is faster in middle portion of stroke than at beginning and end of stroke, as is usually the case, consider that $T = 0.85 \times$ time for complete closure; V_0 = velocity in pipe near turbine gates, corresponding to H_0 and Q_0, ft. per sec.; $Z = (2LV_0/gT\sqrt{H_0})^2$.

THE MAXIMUM PRESSURE RISE theory originally was developed by Joukovsky,* and expanded by Allievi,† Gibson,‡ and others. In his work, Allievi prepared a chart for determination of maximum total pressure. R. S. Quick§ devised the chart shown in Fig. 28 for determining the values of P. This chart may be used to obtain maximum pressure rise with uniform gate motion, and complete gate closure. For values of K and N falling to the left of the line marked Limiting Line for Maximum Pressure at End of First Interval, this chart may be used for partial gate closure.

Fig. 29 is a chart prepared by Quick to assist in determination of value of a for various penstocks. In this chart $a = 4660/\sqrt{1 + (Kd/Eb)} = 4660/\sqrt{1 + (d/100b)}$, where K = bulk modulus of elasticity of water, = 294,000; E = Young's modulus for pipe

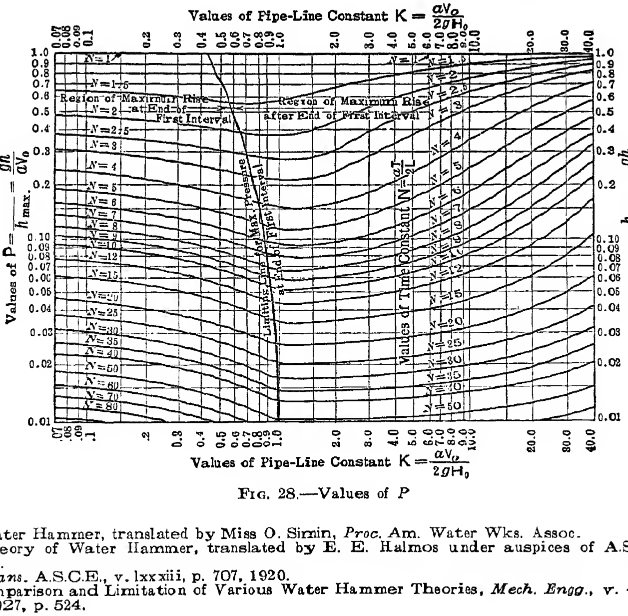


FIG. 28.—Values of P

* Water Hammer, translated by Miss O. Simin, *Proc.* Am. Water Wks. Assoc.
† Theory of Water Hammer, translated by E. E. Halmos under auspices of A.S.M.E. and A.S.C.E.
‡ *Trans.* A.S.C.E., v. lxxxiii, p. 707, 1920.
§ Comparison and Limitation of Various Water Hammer Theories, *Mech. Engg.*, v. 49, Middle May, 1927, p. 524.

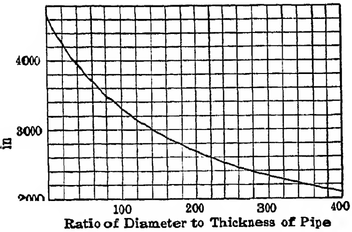


Fig. 29. Values of a

walls = 29,400,000 (approx.) for steel; d = inside diam. of pipe, in., b = thickness of pipe wall, in.

For values of N and K falling to the right of the almost vertical line in Fig. 28, it is important, as demonstrated by S. L. Kerr (New Aspects of Maximum Pressure Rise in Closed Conduits, *Trans.* A.S.M.E., v. 1, 1928), to reduce the rates of movement at the closing end of the gate stroke to prevent the pressure rise for partial gate movements to the closed gate position, from being greater than for complete gate stroke.

MAXIMUM PRESSURE DROP.—Methods for determining pressure drop are similar to those used for pressure rise. Fig. 30 is a chart developed by Kerr (Fall in Pressure in Hydraulic Turbine Penstocks due to Acceleration of Flow, Hydraulic Power Committee, 1924, N.E.L.A. publication No. 24–28) for obtaining value of fall in pressures h, for various values of Z. This chart is based on: Pressure rise h =

$$\tfrac{1}{2}\left(-Z + \sqrt{Z^2 + 4Z\ H_0}\right),$$

where

$$Z = \left\{\frac{2}{g} \times \frac{L}{T} \times \frac{V_0}{\sqrt{H_0}}\right\}^2$$

This formula will hold for any pipe line, where (1) rate of opening of gates is uniform, and (2) where T is not less than $2L/a$. T here is considered as the total time of gate movement. Where the gate movement is not uniform, the formula will hold for all practical purposes, but the maximum pressure drop in this case may occur near the end of the stroke instead of at the end of the first interval ($= 2L/a$).

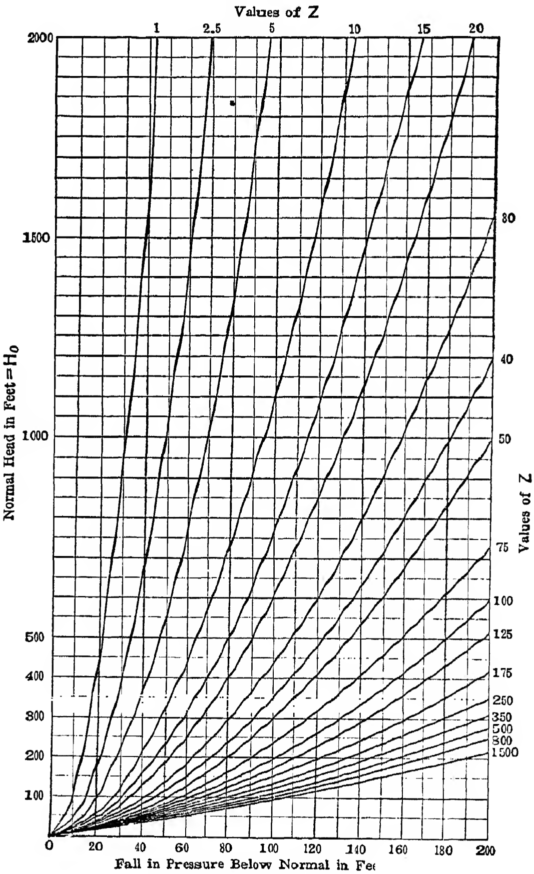


Fig. 30. Values of h

PENSTOCK VIBRATION.—Mechanical vibration in penstocks is discussed by J. P. Den Hartog in *Trans.* A.S.M.E., Hyd. 51–13, 1929. A mathematical analysis shows that objectionable vibrations occur in Francis turbines when the number of buckets on the turbine runner is one less than the number of guide vanes, and that satisfactory operation is obtainable when this relation is avoided. Results of tests showing agreement with the theory are given in the paper

7. IMPULSE TURBINES

The horizontal shaft type of impulse turbine, Fig. 2, is in more extensive use than the vertical type. A few vertical installations have been made with the usual design of bucket, and good efficiencies have been obtained by the provision of a suitable baffle to prevent loss of efficiency due to the discharge from the upper portion of the buckets. It is not considered practicable, unless efficiency is of small consideration, to adopt a value of N_s greater than about 8.0. The interval between this value (see Fig. 4) and the lowest allowable value of N_s with a reaction turbine is covered by the use of, preferably, multi-runner units, or multi-nozzle units or even multi-runner multi-nozzle units. Multi-nozzle units, that is, units with more than one nozzle per runner offer complications in design of operating mechanism and suffer loss of efficiency due to interference of the discharge from the two nozzles, unless very careful attention is given to the relative position of the nozzles. When the power to be developed with the head available results, with a single-runner single-nozzle unit in too low a value of r.p.m., a good arrangement may be adopted by locating a runner on each end of the generator shaft with the runners overhung. In addition to allowing, with a given value of N_s, a value of r.p.m. which is higher by $\sqrt{2}$ than that of a single-runner unit, this arrangement offers the advantage of allowing the shutting down of one runner at part load and the operation of the remaining runner at or about maximum efficiency. For large units it is the best practice, when the foregoing arrangement is adopted, to provide two separate governing mechanisms—one for each runner. A value of N_s = 3.8 generally is considered to be the highest value for which

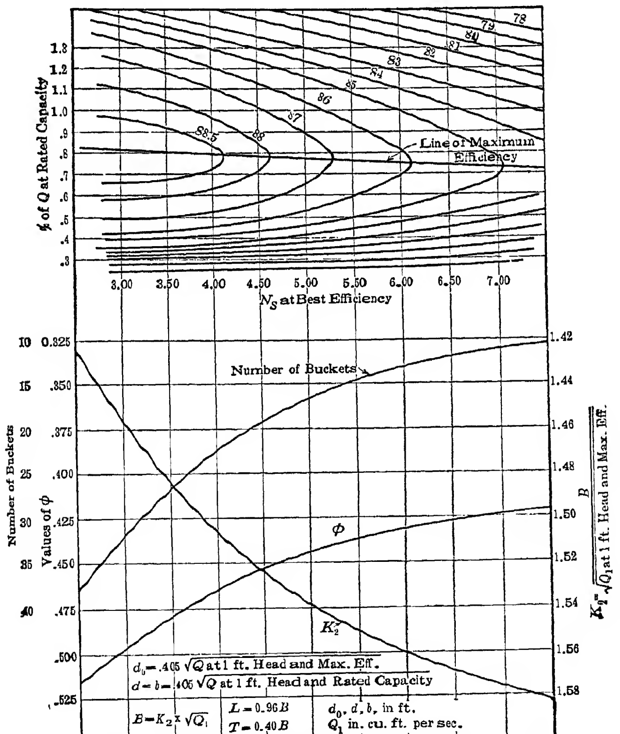


FIG. 31. Relation between ϕ and N_s

maximum possible efficiency may be attained. This value may be reduced indefinitely without appreciable reduction in efficiency. Fig. 31 gives an idea of the manner in which part-load efficiencies vary with N_s.

THEORY OF IMPULSE TURBINES.—The force exerted by a stream upon a bucket (Fig. 32) is $P = MC(1 - \cos\theta) = (W/g) \times C(1 - \cos\theta)$ . . . . . . [17] for a stationary bucket, where M = mass per sec. = W/g; C = velocity of stream, ft. per sec.; W = weight of water flowing per second, lb., and θ = angle through which water is turned relative to bucket, degrees. If the bucket is moving in the direction of the stream with a velocity U at the pitch diameter (see below), $P = (W/g)(C - U)(1 - \cos\theta)$. . [18]

Work done by stream $= PU = (WU/g)(C - U)(1 - \cos\theta)$. . . . . [19]

This is 0 when $U = 0$ and when $U = C$, and is a maximum when $U(C - U)$ is a maximum or when $U = {}^1/_2\,C$. When θ is greater than 90° the cosine becomes negative. For instance if $\theta = 174°$, $\cos = -\sin(174 - 90°) = -0.9945$. Then

$$PU = (WU/g) \times (C - U)(1 + 0.9945).$$

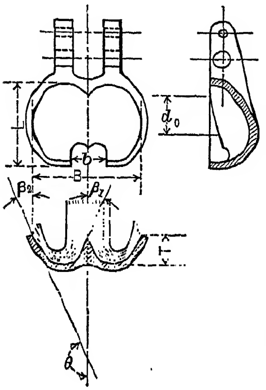


FIG. 32

Actually the water discharged must have some velocity and the value of U consequently is made somewhat less than ${}^1/_2\,C$, except at very low values of N_s. Fig. 31 indicates a relation between ϕ and N_s which has been established by test.* For derivation of ϕ, see page 2–45. The diameter of the circle which falls tangent with the center-line of the stream is known as the pitch diameter of the runner and it is at this point that ϕ is measured (Fig. 33). The value of ϕ at r (Fig. 33), it will be noted, decreases with increasing N_s. That, however, at r_a increases with increasing N_s due to increasing relative size of the buckets. The angle α at the center-line of the bucket should be calculated from the velocity triangle which obtains as the bucket enters the stream, where C = velocity of stream, u_a = that of bucket at radius r_a, and w_a = relative velocity. The under side of the bucket should be finished at a greater angle than α to avoid cavitation of the bucket. The number of buckets on a runner should be such that all portions of the stream will react on the bucket with the maximum attainable efficiency. Fig. 31 indicates the most efficient number and proportions of buckets as determined by the Reichel and Wagenbach tests.* By the relation of stream and bucket velocities it may be established that while the bucket travels from a to b, Fig. 33, the upper edge of the stream travels from a to c, and that the bucket cuts the stream along the line cb. Assume that when the bucket S is in the position shown, the next bucket is in the position T. In order for the particle of water at b to react upon the bucket T the latter must reach the position T' before b reaches the position b'. As a matter of fact the stream, particularly the under surface, reacts only very imperfectly on the bucket when the latter is in the position T'. Therefore the bucket T should be in some position T'', at the time b would reach b' if the bucket were removed. Good design dictates a value of ψ'' of from $0.6\,\psi'$ to $0.7\,\psi'$. On this basis $\psi''/(6.0 \times \text{r.p.m.})$ may be made equal to $(bb'/\sqrt{2gH_0})$. That is, bucket T will reach the position T'' in the time that a particle of water at b would reach b'. The angle between S and T represents the pitch, and for a fixed position for S, the position of T changes with the number of buckets selected. The tests of Reichel and Wagenbach demonstrated that it is important to maintain a small value of discharge angle β_2 (Fig. 32) even though the discharge comes into contact with the back of the bucket just ahead, causing appreciable stream deviation. Hence, backs of the buckets should be smoothly and carefully finished. Attention should be given to finishing the internal or working surfaces, and the intake and discharge angles β_1 and β_2.

BUCKETS.—The buckets are usually of bronze but may be of cast iron where the head is low, or of cast steel where the size of the buckets is relatively large. Accepted practice is to bolt them to the cast-steel hub with two or more fitted bolts, or to cast them integrally with the hub, when N_s is high. The latter should be of sufficient cross-section to resist the stresses at zero speed with maximum $P = (W/g) \times C(1 - \cos\theta)$, and at runaway speed with $P = 0$, also with centrifugal stresses, due to runaway speed of about 1.8 of normal speed. The dimensions B, L, T, of the bucket, it will be noted (Figs. 32 and 33) are considered in terms of $\sqrt{Q_1}$ at best efficiency, although the dimension b is dependent

* Tests of Turbine Buckets by Reichel and Wagenbach, *Zeit.* d. V.D.I., beginning March, 1913.

upon $\sqrt{Q_1}$ at rated capacity. Q_1 is the discharge Q reduced to 1 ft. head and is given by the formula $Q_1 = Q/\sqrt{H}$.

EXAMPLE.—Determine the outline dimensions of the buckets of an impulse wheel to develop 10,000 Hp. when operating at a head of 1000 ft., to generate 60-cycle alternating current.

Solution.—Assume N'_s at rated capacity to be 4.0. Then the speed will be

$$(N'_s \times H^{5/4}) \div \sqrt{\text{Hp.}} = (4.0 \times 1000^{5/4}) \div \sqrt{10{,}000} = 225 \text{ r.p.m.}$$

By adopting a two-runner unit, one-half of the horsepower will be developed in each runner, and the speed will be $(4.0 \times 1000^{5/4}) \div \sqrt{5000} = 318$ r.p.m. A generator at 300 r.p.m. will give 60-cycle current, and, the two-runner unit is selected. The value of N_s for this speed will be $N_s = (300 \times \sqrt{5000}) \div 1000^{5/4} = 3.772$.

The efficiency at rated capacity E_r may be assumed as 88.2%, whence the quantity of water at rated capacity

$Q_r = (\text{Hp.} \times 550) \div (W \times H \times E_r) = (5000 \times 550) \div (62.4 \times 1000 \times 0.882) = 50$ cu. ft. per sec.

In order to arrive at the value of N_s at best efficiency (Fig. 31) it is necessary to adopt the cut-and-try method. Assume as a trial, that Q at best efficiency $= Q_{m.e.} = 0.8 \times Q_r$, whence $Q_{m.e.} = 0.8 \times 50 = 40$ cu. ft. per sec. Assuming again a maximum efficiency E_m of 88.5%, Hp. at this efficiency will be,

$$\text{Hp.}_{m.e.} = (Q_{m.e.} \times W \times H \times E_m)/550 = (40 \times 62.4 \times 1000 \times 0.885)/550 = 4016.$$

The corresponding value of N_s at best efficiency = 3.3805. From Fig. 31, the maximum efficiency may be slightly greater than 88.5 for $N_s = 3.3805$, but it will be well to adhere to this value as the tests do not show higher values actually to have been reached. For $N_s = 3.3805$ at maximum efficiency, the efficiency at rated capacity is about 88.2% which corresponds with our original assumption. Also, it will be noted that the line of maximum efficiency crosses $N_s = 3.3805$ at about 0.8 of Q at the rated capacity, which checks our original assumption.

The best number of buckets (Fig. 31) corresponding to $N_s = 3.3805$ is 28. Also from Fig. 31, $\phi = 0.486$, whence

$$r = (\phi\sqrt{2gH} \times 60) \div (2\pi \times \text{r.p.m.}) = 0.486 \times \sqrt{64.4 \times 1000} \times 60) \div (6.2628 \times 300) = 3.93 \text{ ft}$$

Q_1 at max. effic. $= Q/\sqrt{H} = 40\ \sqrt{1000} = 1.265$ cu. ft. per sec.; Q_1 at rated capacity $= 1.265/0.8 = 1.58$.

Diameter of Jet: At maximum efficiency $d_0 = 0.405\sqrt{Q_1}$ at max. effic. = 0.455 ft. At rated capacity $d = 0.405\sqrt{Q_1}$ at rated capacity = 0.509 ft.

Dimensions of Buckets: From Fig. 31, $K_2 = 1.48$ for $N_s = 3.3805$.

Then B (Fig. 32) $= K_2 \times \sqrt{Q_1}$ at max. effic. $= 1.48 \times \sqrt{1.265} = 1.67$ ft.

$$L = 0.96B = 1.603 \text{ ft.}; \; T = 0.40B = 0.667 \text{ ft.}; \; b = d = 0.509 \text{ ft.}$$

NEEDLE NOZZLE.—The needle nozzle should be placed as close to the buckets as possible, as the stream tends to lose its compactness of form shortly after emerging from the nozzle, due partly to air friction, partly to the centrifugal effect caused by whirl components, and partly to expanding air in the water. The efficiency of a well-designed nozzle is usually between 95 and 97%, corresponding with a velocity in the free jet at its smallest point of between $0.975\sqrt{2gH}$ and $0.985\sqrt{2gH}$, where H = pressure head at d''' (Fig. 33) + the velocity head. Where d is the diameter of the free jet at the rated capac-

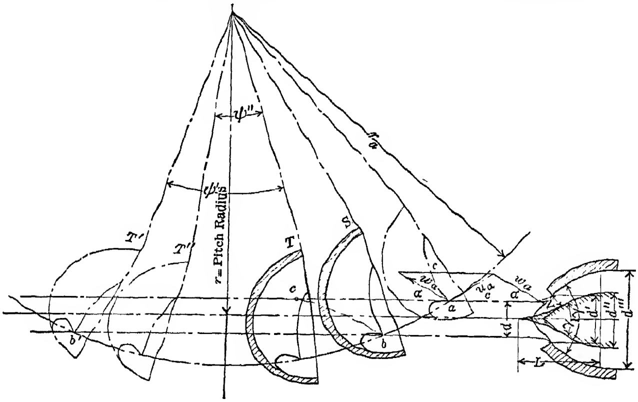


FIG. 33

ity of the unit, the main dimensions of the nozzle (see Fig. 33) may be found as follows:

$$d' = 1.25d;\ \gamma = 70 \text{ to } 80°;\ d'' = 1.55d;\ d''' = 3.2d;\ L = 2.2d.$$

The coefficient of contraction at exit from nozzle is about 0.75; that is, area of d divided by area $N = 0.75$. Care should be taken to have γ_1 smaller than γ (γ_1 being the angle of needle at the portion of greatest taper), otherwise this portion of the needle will be subject to corrosion due to the water not adhering to the needle surface. The needle as well as the nozzle near the outflow should be very smoothly finished. These parts should be made easily removable to provide for replacement when worn. Stainless steel for these parts gives excellent service against abrasion and pitting. It is most important that the flow be free from whirl components in approaching the nozzle. Often vanes, for straightening the flow at this point, are advisable.

The casing, in the vicinity of the nozzle, should be amply large to allow free discharge from the buckets. The remainder of the casing need only be large enough to clear the buckets, as windage is less than with needlessly large clearance. The discharge passage from the impulse wheel should have its outlet end above tail-water; otherwise the confined air in the casing will become entrained in the water, and the ejector action resulting will cause the water level in the discharge passage to rise until it reaches the runner buckets.

RELIEF VALVES.—The long penstocks usually necessary with impulse turbine installations prevent quick movement of the needle by the governor, particularly in the closing direction, unless there is a compensating opening of a relief valve to prevent injurious water hammer. Such relief valves are sometimes rigidly connected so that their movement is opposite to that of the needle, in which case there is no change in penstock velocity and hence no limitation on speed of needle movement. Such an arrangement of course offers the disadvantage that there is a waste of water when the unit is operating at small loads. More often the relief valve is connected to the needle through a dash-pot which allows the valve to close slowly after a quick closing of the needle with simultaneous opening of the valve. With such an arrangement there is danger of penstock collapse if the time of opening the needle be too short.

The governor capacity required is comparatively small, being only that necessary to overcome friction and the hydraulic load on the needle and relief valve or jet deflector; the latter can be balanced to a certain extent. The method of calculating regulation for sudden load changes (p. 2–55) applies to both impulse and reaction turbines.

JET DEFLECTORS often are used between nozzles and buckets. The governor moves these deflectors rapidly into the stream, thus cutting off the load rapidly without endangering the penstock. Following this movement the needle moves slowly to cut off the flow, and at the same time the deflector moves slowly out of the stream. The needle must move slowly in the opening direction for on-coming loads to avoid penstock collapse.

TURBINE COSTS cannot be given here, even approximately, as this item varies greatly with different makes of equipment, and also with labor and material costs which are always fluctuating more or less. Except possibly where the power to be developed is very small, it is advisable for the promoter or owner to have a reliable firm of consulting hydro-electric engineers examine, make surveys of, and submit a report on the advisability of proceeding with the development under consideration. Such a report involves a study of stream flow, head of water and power available, cost of the entire development, market available, probable revenue, and estimated rate of return on the investment.

In obtaining prices and data from turbine manufacturers the following information should be furnished: 1. Flow of water to be utilized, in cu. ft. per sec. 2. Normal head available, also maximum and minimum heads. 3. Maximum fluctuation of upper (head) and lower (tail) water levels. 4. Elevation of site of plant above sea level. 5. Number of units under consideration. 6. Importance of efficiency at part loads. 7. Length and diameter of penstocks, if any. 8. Frequency of electric current to be generated.

This information will enable the turbine manufacturer to make recommendations of type, unit capacity in Hp., and r.p.m. and to estimate the turbine and governor cost, as well as to prepare drawings of the equipment for the determination of power-house cost.

The importance of purchasing reliable and efficient equipment is very great where hydraulic turbines are concerned. This equipment is generally embedded in the substructure of the power house and can be replaced only at very great expense.

Manufacturers.—The principal manufacturers of hydraulic turbines are: Allis-Chalmers Mfg. Co., Milwaukee, Wis.; Baldwin-Southwark Corp., I. P. Morris Division, Philadelphia, Pa.; S. Morgan Smith Co., York, Pa.; The Pelton Water Wheel Co., San Francisco, Cal., a subsidiary of the Baldwin-Southwark Corp. The above firms manufacture reaction turbines of the Francis propeller and Kaplan types, impulse turbines, and governors. The Newport News Shipbuilding and Drydock Co., Newport News, Va., reaction turbines of the Francis and propeller types. James Leffel & Co., Springfield, Ohio, reaction and Kaplan type turbines. The Woodward Governor Co., Rockford, Ill., turbine governors.

PUMPS AND PUMPING ENGINES *

By Robert Thurston Kent

1. STEAM PUMPS

Notation.—B.Hp. = brake horsepower; I.Hp. = indicated horsepower of steam end of steam pump; W. Hp. = water horsepower; d = diam. of pump piston or plunger, in.; E_h = hydraulic efficiency, percent; E_m = mechanical efficiency, percent; E_v = volumetric efficiency, percent; G = U. S. gal. per min. = 7.4805 Q; H = head pumped against, ft.; H_l = total hydraulic losses (see below); h = suction head, ft.; l = stroke, in.; N = number of cycle strokes per min.; n = r.p.m. = number of cycles per min. (in duplex pumps n = 4 single strokes, counting both sides); p = total pressure, including suction lift, lb. per sq. in.; Q = displacement, cu. ft. per min.; Q_a = quantity of water actually pumped, cu. ft. per min.; v = piston speed, ft. per min.

CAPACITY, HORSEPOWER AND EFFICIENCY.—**Single Double-acting Pumps**: $v = lN/12$; $Q = \pi d^2 v/(4 \times 144)$; $G = 0.0408\, d^2 v$.

Duplex Double-acting Pumps: $v = ln/6$; $Q = 0.010908 d^2 v$; $G = 0.0816 d^2 v$. The displacement of piston rod must be deducted. If the piston rod is a single-acting plunger, its displacement will be 1/2 that given by above formulas.

Capacity, $Q_a = Q \times E_v$. For pumps in good condition, $E_v = 90\%$ approx.

Efficiency.—Mechanical efficiency is the ratio of work done in the water end to the work expended in driving the pump. For a steam pump, E_m = W.Hp./I.Hp.; for a power-driven pump, E_m = W.Hp./B.Hp. B.Hp. is taken as the power input at the pump coupling or pulley. Volumetric efficiency = $E_v = Q_a/Q$. Hydraulic efficiency = $E_h = (H + h)/H + H_l$.

Horsepower.—W.Hp. = $GwH/33{,}000 = Gp/1714$. At 60° F., W.Hp. = $GH/3960$. B.Hp. = W.Hp. $\times E_m$. See Table 1 for values of E_m.

Table 1.—Mechanical Efficiency of Various Types of Pumps
(Worthington Pump & Machinery Corp., Harrison, N. J.)

Type of Pump	Maximum Pressure, lb. per sq. in.	Stroke, in.										
		3	4	5	6	8	10	12	15	18	24	36
		Efficiency, percent										
Piston........	250	50	55	60	65	70	75	77	80	82	85	87
Packed plunger	300	47	52	57	61	66	71	73	76	78	81	83
Pressure.......	1000	45	50	54	58	63	67	69	72	74	77	79
"	> 1000	39	43	47	51	55	58	60	62	64	66	68
Thick liquid...	....	39	43	47	51	55	58	60	62	64	66	68

SLIP is the loss in displacement due to defective packing, leaky valves, etc. Slip = $(1 - E_v)$. Slip ranges from 3 to 5% in good pumps, and may be as low as 0.5% or even have a negative value in large pumps.

TOTAL HEAD ON PUMP is the pressure set up to lift the water in the suction pipe and force it through the delivery pipe to point of discharge. It includes the pressure required to overcome the resistance of valves and passages, and the friction losses in suction and delivery pipes.

Let H = total head on pump; H_s and H_d = total suction and delivery heads, respectively; h_f and H_f = friction losses in suction and delivery pipes; h_v and H_v = friction losses in suction and delivery valves; h_e and H_e = head lost at entrance to suction and delivery pipes; h_m and H_m = measured suction and delivery heads; h_p = velocity head in suction pipe; H_p = final velocity head; h_g and H_g = distance from center of pump to center of suction and discharge gages, ft., all measured in ft. of head; v and V = velocity in suction and discharge pipes, ft. per sec.; p_s and p_d = pressure on suction and discharge gages, lb. per sq. in. ($p_s <$ atmospheric pressure); $g = 32.2$; W = weight of water, lb. per cu. ft. Then $H_s = (h_e + h_f + h_v + h_p + h_m)$; $H_d = (H_e + H_v + H_f + H_p + H_m$; $h_p = (v^2/2g)$, and $H_p = (V^2/2g)$. $H = (h_s - h_p + H_d)$ when suction and delivery pipes are the same size. $H = [\{(144 p_s/W) \pm h_g\} - (v^2/2g)] + \{(144 p_d/W) + (V^2/2g)\}$. If

* The author acknowledges the assistance of Mr. Paul Diserens, Consulting Engineer of the Worthington Pump & Machinery Corporation in the revision of this section.

suction pipe gage is above center line of pump, h_g is plus; if below h_g is minus. The friction losses, h_v, H_v, and H_e usually are included in the efficiency factor of the pump; then $H = h_e + h_f + h_m + H_m$.

Loss of head at entrance of suction pipe, $h_e = (mv^2/2g)$; values of m may be taken as: 0.49 for orifice flush with wall; 0.93 for orifice projecting inwardly beyond wall; 0 for pipe with mouthpiece. (See also p. 2–12.) The same formula applies to the delivery pipe, V being substituted for v.

Values of h_f and H_f depend on the roughness of the interior pipe surface and the resistance due to bends, valves, etc. According to Merriman (Treatise on Hydraulics), $h_f = f\,(l/d) \times (v^2/2g)$, where l = length of pipe, ft.; d = diameter, ft.; f = coefficient of roughness of surface, ranging from 0.1 to 0.5 for new clean iron pipe; f increases with increase of d, and decreases with increase of v. (See also pp. 2–14—2–21.)

Loss of head due to bends, $h_b = Kf(l/d) \times (v^2/2g)$, where K = a factor depending on the ratio R/d, where R = radius of bend and d = diam. of pipe; l = length of center line of bend, ft. Values of K are:

R/d =	5	10	15	20	25	30
K =	2.04	1.67	1.52	1.45	1.425	1.42

Loss of head due to valves, cocks, etc., $h_q = (mv^2/2g)$, where m = factor depending on closure of valve. Greene (Pumping Machinery) gives values of m as follows:

d_1/d	= 0.1	0.2	0.3	0.4	0.5	0.6	0.7	0.8	0.86
Gate valves, m	= 0.1	0.5	1.1	2.1	3.2	8.6	20.2	60.0	140
θ	= 10	20	30	40	50	60	61	65	
Cocks, m	= 0.1	1	4	12	34	120	....	280	
Butterfly valves, m	= 1	2	7	18	55	206	280	....	

d = diam. of pipe, in.; d_1 = distance from top of pipe to lowest point of gate disc; θ = angle between axis of pipe and axis of valve, deg. F. W. Isles (*Mech. Engg.*, July, 1932) reports that experiments with various forms of cocks showed that one in which the plug was cored to the form B in Fig. 1 had a frictional resistance to flow of about one-half that of the regular form of plug A, Fig. 1, and that for a given pressure loss would discharge about 35% more fluid than the regular cock.

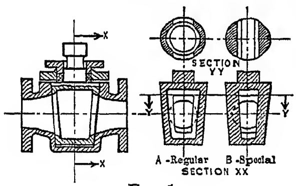


FIG. 1

Loss of head due to sudden enlargement in suction or delivery pipe,

$$h_{en} = (v_s - v)^2/2g = \{1 \cdot$$

where v_s and v = respectively, velocity before and after enlargement; A_s and A = area before and after enlargement.

Loss of head due to sudden contraction in suction or delivery pipe,

$$h_c = \{(1/K) - 1\}^2 \times (v_s^2/2g);\ K = 0.582 + \{0.0418/(1.1 - \rho)\}$$

where ρ = ratio of diameters of small and large pipes. See Greene's Pumping Machinery, 2d ed., chap. v, for a mathematical discussion of the various losses in a pump.

DUTY OF A PUMP is the number of ft.-lb. of energy delivered by the pump for every 1000 lb. of dry steam or every 1,000,000 B.t.u. supplied to it. See Table 2 for probable duty of various types of pumps and pumping engines.

The rate of feedwater consumption is the figure given in Table 2 divided into 1980, thus: if the duty of a pump is 88 million ft.-lb. per 1000 lb. of dry steam, the rate is: 1,980,000,000/88,000,000 = 22.5 lb. per Hp. per hr.

DEPTH OF SUCTION.—The theoretical maximum depth from which a pump will draw water, H_s, is the head corresponding to a perfect vacuum = 14.7 lb. × 2.309 = 33.95 ft. Actually H_s is not over 26 ft. under most favorable conditions, due to valve leakage, entrained air, and the vapor of water itself. In a new installation handling cold water, $H_s \gtreqless 22$ ft., being less as the temperature of the water increases. H_s also decreases as altitude above sea level increases. See Table 3 for effect of increase of temperature and altitude. Hot water should flow to the pump by gravity.

VELOCITY OF WATER IN SUCTION AND DELIVERY PIPES.—Increase of velocity of water increases the friction head. Pump manufacturers recommend the following as *maximum* velocities: In *suction pipe*, 240 ft. per min.; in *delivery pipe*, 300 ft. per min. If d = diam. of pipe, in.; G = gal. of water per min.; V = velocity, ft. per min.; $d = 4.95\sqrt{G/V}$. Suction and discharge lines should be at least of the diameter indicated by the flanges on the pump; for long runs, particularly of the suction line, they should be larger. Air pockets should be avoided. See Fig. 2. The submergence of the suction pipe should be at least $4d$ for large pipes, and $3d$ for small ones.

PISTON SPEEDS of direct-acting steam pumps for various classes of service are given in Tables 4 and 5.

SIZE OF STEAM CYLINDER, DIRECT-ACTING PUMPS.—Let A = area of steam piston, sq in.; F = force of thrust on steam piston, lb. = $A \times$ M.E.P.; W = load on water plunger, lb. = $a \times w = F \times E_m$; a = area of water plunger, sq. in.; W = pressure on or resistance of water plunger, lb. per sq. in.; M.E.P. = mean effective pressure in steam cylinder, lb. per sq. in.; E_m = mechanical efficiency (See p. 2–65 and Table 1. In calculating proportions of steam cylinders, values of E_m equivalent to 90% of those given in Table 1 are recommended); P = absolute pressure in high-pressure cylinder = boiler pressure + 10, lb. per sq. in.; b = back pressure (absolute), lb. per sq. in.; maximum values of b are: non-condensing steam ends, b = 16; compound condensing steam ends, b = 6; triple-expansion condensing steam ends, b = 5; A_h, A_i, A_l = area of high, intermediate and low pressure cylinders respectively; R = ratio of cylinder areas = $\sqrt{P/b}$ = 4 (max.) for compound steam ends; for triple-expansion steam ends, $R_1 = A_l/A_h = \sqrt[3]{(P/b)^2} = 8$ (max.); $R_2 = A_i/A_h = \sqrt{R_1} = 3$ (max.).

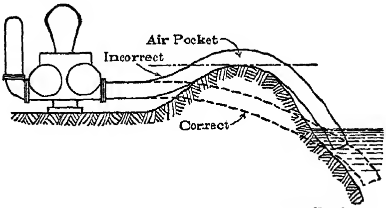


Fig. 2. Correct and Incorrect Installation of Suction Line

Then M.E.P. = $P - b$ (simple); = $\{2P - (P/R) - bR\}$ (compound, referred to h.p. cyl.); = $\{3P - 2(P/\sqrt{R_1}) - bR_1\}$ (triple expansion, referred to h.p. cyl.).

Values determined by the above formulas are not absolute. Actual mechanical effi-

Table 2.—Probable Duty of Steam Pumps

Duty given in million foot-pounds of work done per 1000 lb. of dry steam. Steam cylinder lagged. Wire drawing assumed = 4.7 lb. per sq. in.

Stroke, in.	Direct-Acting Steam Pumps								
	Single-Cylinder, Non-condensing, not Jacketed, 16 lb. Back Pressure			Compound, Non-condensing, not Jacketed, 16 lb. Back Pressure			Compound, Condensing, Steam Jacketed, 6 lb. Back Pressure		
	Boiler Pressure by Gage, lb. per square inch								
	80	120	150	80	120	150	100	150	180
10	18	20	20	25	29	30	40	43	44
15	21	23	24	30	34	36	47	50	51
18	23	25	25	32	36	38	51	54	56
24	25	27	28	35	40	42	54·	58	59
36	26	29	29	37	42	45	58	62	63

Stroke, in.	Direct-Acting Steam Pumps			Crank and Flywheel Pumps with Releasing Gear					
	Triple Expansion Condensing, Steam Jacketed, 4 lb. Back Pressure			Compound Condensing			Triple Expansion Condensing		
				Ratio of Expansion					
				14	15	16	18	21	23
	Boiler Pressure by Gage, lb. per square inch								
	120	160	200	100	130	160	120	160	200
15	87	92	95	92	101	105	104	115	122
18	93	98	101	98	107	112	110	122	129
24	99	104	108	104	114	119	117	130	137
36	105	111	114	111	121	126	123	137	144
48	....	....	....	117	128	134	130	145	153

Table 3.—Maximum Suction Lifts at Various Temperatures and Altitudes

Altitude above Sea-level, ft.	Temperature of Water, deg. F.															
	60	70	80	90	100	110	120	130	140	150	160	170	180	190	200	210
	Suction Lift, Ft.										Suction Pressure, ft.					
0	−22	−20	−17	−13	−13	−11	−8	−6	−4	−2	0	+3	+5	+7	+10	+12
2,000	−19	−17	−15	−13	−11	−8	−6	−4	−2	+1	+3	+5	+7	+10	+12	+15
4,000	−17	−15	−13	−10	−8	−6	−4	−1	+1	+3	+5	+7	+10	+12	+14	....
6,000	−15	−13	−11	−8	−6	−4	−2	+1	+3	+5	+7	+10	+12	+14	+16	....
8,000	−13	−11	−9	−6	−4	−2	0	+3	+5	+7	+9	+12	+14	+16	....	....
10,000	−11	−9	−7	−4	−2	0	+2	+4	+7	+9	+11	+14	+16	+18	....	....

ciencies may exceed those in Table 1, when a pump is in first class condition. The steam end of a direct-acting pump, however, must be designed to take care of unfavorable conditions, since power cannot be increased by later cut-off.

SIZE AND NUMBER OF VALVES.—For the theory underlying the determination of size and number of valves, and also loss of head in the passage of the water through them, see Greene's Pumping Machinery, 2d ed., chap. v.

Let A_p = area of plunger, sq. in.; S = piston speed, ft. per min.; A = gross area of valves, sq. in.; a = free valve area, sq. in., in each corner of pump; n = number of corners = 4 in single pumps and 8 in duplex pumps; N = number of valves; d = diam. of valve seat, in.; L = lift of valve, in. Then $a = \pi dL = 0.5\ A_pS$ to $0.6\ A_pS$; $A = na$, $N = A/0.7854d^2$. These formulas are based on a water velocity of 167 to 200 ft. per min. Thick viscous liquids require greater valve area, which may be as high as 75 to 100% of the plunger area, depending on the viscosity of the liquid.

Figs. 3 to 8 show various forms of pump valves. In rubber disc valves for pressures up to 75 lb. per sq. in., the rubber may be soft; up to 150 lb., medium hard; above 150 lb., hard, of special composition. The disc type metal valve is of bronze with ground joint on the seat. These valves are for general service and thin liquids. The ball valve is used with thick, viscous liquids, as tar, paper pulp, etc. The angle of the seat to center line of valve should not exceed 45°, and lift may be $0.1d$ to $0.25d$. The wing guided valve is used for pressure pumps, and may have a flat or conical ground seat, or be faced with rubber.

Table 4.—Speeds and Capacities of Single Pumps

(Worthington Pump & Machinery Corp., Harrison, N. J., 1935)

Diam., in., of		Stroke, in.	Service					
			Boiler Feed		General		Emergency	
Steam Cyl.	Pump Cyl.		Strokes per min.	U. S. gal. per min.	Strokes per min.	U. S. gal. per min.	Strokes per min.	U. S. gal. per min.
4 1/2	2 3/4	6	54	8.5	90	14.	105	16
5 1/2	3 1/4	7	52	13	86	21.5	100	25
6 1/2	4 1/8	8	50	23	82	38	95	44
7 1/2	4 1/2	10	45	31	75	52	86	60
8	5	12	42	43	70	71	80	82
10	6	12	42	61	70	102	80	118
12	7	12	42	84	70	140	80	160
14	8	12	42	110	70	182	80	210
16	10	18	36	220	60	367	70	425

Table 5.—Speeds and Capacities of Duplex Pumps

(Worthington Pump & Machinery Corp., Harrison, N. J., 1935)

Diam., in., of		Stroke, in.	Service					
			Boiler Feed		General		Emergency	
Steam Cyl.	Pump Cyl.		Cycles per min.	U. S. gal. per min.	Cycles per min.	U. S. gal. per min.	Cycles per min.	U. S. gal. per min.
3	2	3	36	6	60	9.5	70	11
3 1/2	2 1/4	4	31	8.5	52	14.5	60	16
4 1/2	2 3/4	4	31	13	52	21	60	25
5 1/4	3 1/2	5	29	24	48	40	55	46
6	4	6	27	35	45	58	52	67
7 1/2	5	6	27	55	45	92	52	105
7 1/2	4 1/2	10	23	63	38	105	43	120
9	5 1/4	10	23	85	38	142	43	163
10	6	10	23	110	38	185	43	213
12	7	12	21	168	35	280	40	310
14	8 1/2	12	21	250	35	410	40	470
16	10 1/4	12	21	340	35	570	40	660

Table 6.—Sizes and Capacities of Underwriter Duplex Steam Fire Pumps

Size of Pump, in.			Capacity, Underwriter Rating				Pipe Sizes, in.				Floor Space, in.
Diameter		Stroke, in.	Gal. per min.	No. of 1 1/8-in. Fire Streams	Piston Speed, ft. per min.	R.p.m.	Steam	Ex-haust	Suc-tion	Dis-charge	
Steam Cyl.	Water Plunger										
14	7 1/4	12	500	2	140	70	3	4	8	6	110 × 41
16	9	12	750	3	140	70	3 1/2	4	10	8	111 × 48
18	10	12	1000	4	140	70	4	5	12	8	111 × 51
20	20	16	1500	6	160	60	5	[illegible]	[illegible]	[illegible]	[illegible]

Barr, in Pumping Machinery, gives the dimensions of valve discs as follows:

Diam. of disc, in.	2	2 1/2	3	3 1/2	4	4 1/2	5
Thickness, in.	3/8	7/16	1/2	5/8	5/8	3/4	3/4
Diam. of hole	1/2	1/2	9/16	5/8	11/16	13/16	13/16
Size of spring wire, B.W.G.	12	12	10	10	8	8	8

The thread on the valve seat to fit the valve deck should taper 1 in. per ft. Diam. of valve stems = (diam. of hole − 1/16 in.). The plate on top of the valve disc should be 3/4 diam. of disc and 1/32 in. thick; 1/16 in. thick for valves 4 1/2 in. and over. Diameter of coil of springs usually is 1/2 the valve diameter, and 5 turns give sufficient elasticity.

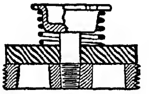


FIG. 3. Rubber Disc

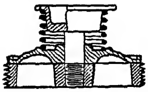


FIG. 4. Metal Disc

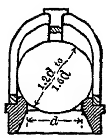


FIG. 5. Ball

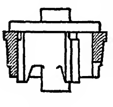


FIG. 6. Flat Face

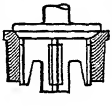


FIG. 7. Bevel Face

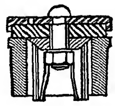


FIG. 8. Rubber or Metal Face

Wing Guided Valves

Types of Valves

AIR CHAMBERS to compensate for irregularities in flow of water should be placed on both delivery and suction lines. Fig. 9 shows approved methods of installation on suction lines. Let V = volume of air in chamber when pump is operating, cu. ft.; V_1 = minimum volume of chamber that will avoid losing water with pump idle; h and h_1 = respectively, head on pump when operating and idle, ft. = maximum permissible variation in pressure in air chamber; D = a factor explained below. Then $V = D/p$ and $V_1 = Vh/h_1$. Value of D may be determined graphically.

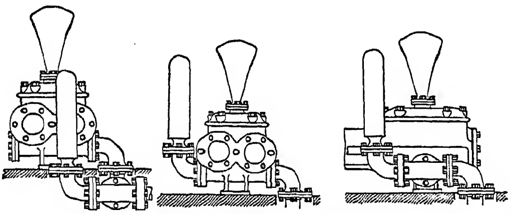


FIG. 9. Methods of Installing Air Chambers

EXAMPLE.—In Fig. 10, the curves PP represent the discharge of the two sides of a duplex double-acting pump, the dotted curves the resultant discharge, and the line AB the mean discharge. Areas e_1, e_2, e_3 represent quantities of water delivered to the air chamber, and areas d_1, d_2, d_3, quantities delivered by the air chamber. Then D = greatest difference between the largest values of e and d respectively. An empirical rule (Greene) is $V = 3C$ to $6C$ for single-acting pumps; $V = 1.5C$ to $4C$ for double-acting pumps, where C = cylinder volume. The volume of the vacuum chamber on the suction line should equal that of the air chamber.

BOILER FEED PUMPS.—In determining the proper size of pump for a steam boiler, allowance must be made for sufficient water to cover all the demands of engines, turbines, steam heating, etc., up to the capacity of the boiler. In practice, engines use from 12 to 50 or more pounds of steam per Hp.-hr., when being worked to capacity. When an engine is overloaded more water per horsepower will be required than when operating at its rated capacity.

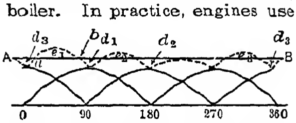


FIG. 10. Determination of Air Chamber Volume

The pump should not be chosen on the basis of Boiler-Hp. It is best to calculate the amount of steam used in the plant and select the size of feed pump that will handle this amount at a moderate speed.

PUMP FITTINGS.—The liquid pumped determines the kind of material used in the pump. The following are the designations of the types of pump fittings in general use. *Plain-fitted Displacement Pumps.*—Bronze-lined pump cylinders, iron pump pistons, steel rods, bronze or rubber valves, bronze valve seats, guards and springs. *Plain-fitted Centrifugal Pumps.*—Iron impellers, steel shafts, bronze impeller bushing rings. *Bronze-fitted Displacement Pumps.*—Bronze lined pump cylinders, bronze piston rods, valve seats, guards and spring, bronze or rubber valves, iron pump pistons. *Bronze-fitted Centrifugal Pumps.*—Bronze impellers, impeller bushing rings, bronze or bronze-covered steel shaft. *Full Bronze-fitted Displacement Pumps.*—Bronze-lined pump cylinders; bronze

pump pistons, piston rods, valve seats, guards, and springs; bronze or rubber valves. *All Iron.*—All parts in contact with liquid of cast or malleable iron. Steel piston rods or shafts. *All Bronze.*—All parts in contact with liquid of bronze. Bronze, or bronze-covered piston rods and shafts. *Cast-iron lined.*—Liquid cylinder of cast-iron, lined with lead or wood.

The Worthington Pump & Machinery Corp. gives the following types of fittings as best adapted to the liquids named:

All Bronze.—Acids: Acetic, Carbolic, Fatty, Hydrochloric, Mine Water, Nitric, Sulphuric (dilute). Alcohol. Beer and Beer Wort. Bitter-water. Brine (over 3% salt). Calcium-acid-sulphate (conc.). Calcium-chlorate. Copper-sulphate. Ferric Hydroxide. Glycerine. Grape Juice. Magnesium-acid-sulphate, magnesium-oxy-chloride. Milk. Nickel-chloride, nickel-sulphate. Hydrogen peroxide. Potassium-chloride, potassium-nitrate, potassium-sulphate. Sodium-chloride. Strontium-nitrate. Sugar. Sulphur-dioxide. Syrup. Tan Liquor. Vinegar. Vitriol (blue). Yeast. Zinc-nitrate, zinc-sulphate.

All Iron.—Acids: Carbolic, cyanic, hydrocyanic, nitric, sulphuric (conc., fuming and 60° B). Alkaline liquid. Ammonia. Ammonium-chloride, ammonium-nitrate, ammonium-sulphate. Aniline water. Asphaltum. Benzine (coal tar product). Bichloride of mercury. Carbonate of soda. Caustic-lye, caustic-manganese, caustic-soda. Chloride of zinc. Cyanide of sodium, cyanide of potassium. Ferrous chloride. Lard. Lime Water. Lye. Magnesium-sulphate. Milk of lime. Oils: Light lubricating, creosote, cocoanut, gas, rape, turpentine, wash. Potash. Potassium-alum, potassium-carbonate, potassium-cyanide. Salammoniac. Soap. Sodium-bicarbonate, sodium-carbonate, sodium-hyposulphite, sodium-sulphate, sodium-sulphide (hot). Tar. Vitriol (green).

Plain Fitted.—Acids: Carbolic in H_2O, citric. Alum. Ammonium bicarbonate. Barium-chloride, barium-nitrate. Carbon tetrachloride. Cellulose. Glucose. Hydrogen-sulphide. Marsh gas. Sodium sulphide. Oils: Cottonseed, fuel, kerosene.

Bronze Fitted.—Oils: Crude, fuel, linseed. Water, fresh.

Full Bronze Fitted.—Beet juice. Benzine (Petroleum ether). Brine. Calcium chloride, salt (3%). Calcium-acid-sulphate. Creosote. Dyewood Liquor. Gasoline. Glue. Heptane. Magnesium-acid-sulphate. Mash. Molasses. Naphtha. Paraffine (hot). Sea water. Sewage. Slop, brewery. Starch. Varnish. Water, -distilled, -sulphur. Wood Pulp. Zinc-chloride, zinc caustic.

2. PUMPING ENGINES

THE WORTHINGTON HIGH-DUTY PUMPING ENGINE substitutes for a flywheel a pair of oscillating hydraulic cylinders, which receive part of the energy exerted by the steam during the first half of the stroke, and give it out in the latter half. For description see catalog of Worthington Pump & Machinery Corporation, Harrison, N. J. A test of a triple-expansion condensing engine of this type is reported in *Eng. News*, Nov. 29, 1904. Steam cylinders 13, 21, 34 in.; plungers 30 in.; stroke 25 in. Steam pressure, 124 lb. Total head, 79 ft.; capacity, 14,267,000 gal. in 24 hr. Duty per million B.t.u., 102,224,000 ft.-lb.

THE d'AURIA PUMPING ENGINE substitutes for a flywheel a compensating cylinder in line with the plunger, with a piston which pushes water to and fro through a pipe connecting the ends of the cylinder. It is built by the Builders' Iron Foundry, Providence, R. I.

A 72,000,000-GALLON PUMPING ENGINE at the Calf Pasture Station of the Boston Main Drainage Works is described in *Eng. News*, July 6, 1905. It has three cylinders, 18 1/2, 33 and 52 3/4 in., and two plungers, 60 in. diam.; stroke of all, 10 ft. The piston-rods of the two smaller cylinders connect to one end of a walking beam and the rod of the third cylinder to the other. Steam pressure 185 lb. gage; revolutions per min. 17; static head 37 to 43 ft. Suction valves 128; ports, 4 × 16 1/4 in.; total port area 8576 sq. in. Delivery valves, 96; ports, 4 × 16 3/4 to 20 3/4 in.; total port area 7215 sq. in. Valves are rectangular rubber flaps, backed and faced with bronze and weighted with lead. They are set with their longest dimension horizontal, on ports inclined about 45° to the horizontal. At 17 r.p.m. the displacement is 72,000,000 gal. in 24 hr.

THE SCREW PUMPING ENGINE of the Kinnickinick Flushing Tunnel, Milwaukee, has a capacity of 30,000 cu. ft. per min. (= 323,000,000 gal. in 24 hr.) at 55 r.p.m. The head is 3 1/2 ft. The wheel 12.5 ft. diam., made of six blades, revolves in a casing set in the tunnel lining. A cone, 6 ft. diam. at the base, placed concentric with the wheel on the approach side diverts the water to the blades. A casing beyond the wheel contains stationary deflector blades which reduce the swirling motion of the water (Allis-Chalmers Co., Bulletin No. 1610). The two screw pumping engines of the Chicago

sewerage system have wheels 14 3/4 ft. diam., consisting of a hexagonal hub surmounted by six blades, and revolving in cylindrical casings 16 ft. long, allowing 1/4 in. clearance at the sides. The pumps are driven by vertical triple-expansion engines with cylinders 22, 38 and 62 in. diam., and 42 in. stroke.

Table 7.—Notable High-duty Pumping Engine Records

VERTICAL TRIPLE EXPANSION CONDENSING. (SEE NOTE 1)

Location	(1) St. Louis, Bissell's Point Station	(2) Cleveland, Division Station	(3) Cleveland, Division Station	(4) Cleveland, Division Station	(5) Milwaukee, Riverside Pumping Station
Date of test	1914	1918	1918	1918	1925
Capacity, gal. per 24 hr	20,576,500	20,000,000	10,000,000	20,000,000	22,000,000
Diameter steam cyls., in	34–64–98	29 1/2–58–92	27–52–84	36–68–108	32–58–96
Stroke, in	66	64	48	66	60
Diameter plungers, in		34	22 7/8	31 5/8	
Piston speed, ft. per min	220	197.41	217.06	230.23	235
Total head, ft	297.7	247.98	382.85	377.19	120.19
Steam pressure, lb. per sq. in.	159.4	195.8	199.3	207.12	200.5
Superheat, degrees F	105.3*	120.7	102.5	136	118.8
Indicated horsepower	1165	943.22	737.31	1416.6	1128.89
Friction, mech. and hyd., %	7.8	7.18	8.76	5.21	3.5
Efficiency, mech. and hyd., %	92.2	92.82	91.24	94.79	96.5
Steam per I.Hp.-hour	8.98	8.86	8.96	8.89	8.96
Duty, B.t.u. basis	166,719,796†	172,780,000	169,680,000	188,743,000	176,170,000
Duty per 1000 lb. of steam	202,628,616	207,357,000	201,581,000	210,992,000	213,270,000
Thermal efficiency, %	21.4	22.21	21.81	24.27	22.65

HORIZONTAL CROSS-COMPOUND CONDENSING. (SEE NOTE 2)

Location	(1) Kansas City, Kan.	(2) Amalgamated Phosphate Co., Florida	(3) York Water Company, York, Pa.	(4) City of Kenosha, Wis.
Date of test	1913	1914	1915	1917
Capacity, gal. per 24 hr	12,110,000	9,500,000	8,249,758	6,237,028
Diameter of steam cyls., in	27 1/2–62	27–60	24–54	18–40
Diameter of plungers, in	19	17	17	16
Stroke, in	42	42	36	36
Revolutions per minute	42.31	42.8	41.96	35.53
Piston speed, ft. per min	296.17	300	251.76	213.18
Total head, ft	285	423	312.18	191.38
Steam pressure, lb. per sq. in.	147.7	175	148.8	161.7
Superheat, degrees F		18	126.4	
Delivered horsepower	605	700	454.5	
Steam per delivered Hp.-hr	12.36	12.25	11.27	13.21
Duty per 1000 lb. of steam	160,040,000	161,386,000	174,805,000	149,822,000

* Duty with 120° F. superheat, 206,000,000. † Without reheaters.

NOTE 1.—The above were all vertical triple-expansion crank-and-flywheel pumping engines, each with three single-acting plungers. Nos. 1 and 3 were Worthington-Holly engines, Nos. 2, 4, and 5 were Allis-Chalmers engines. The data are taken from official reports. Nos. 2 and 3 had been installed about 20 years and are good examples of economy after a long period of operation. Nos. 1, 4 and 5 were new when tested. No. 4 thermal efficiency of 24.27% included a feedwater heater attached to the second receiver; its thermal efficiency without this heater was 21.61%. Nos. 1, 2, and 3 had no second receiver-heaters.

NOTE 2.—The above are crank-and-flywheel engines each with two double-acting plungers and were new when tested. The data are taken from official reports. Nos. 1 and 4 are Allis-Chalmers engines. Nos. 2 and 3 are Worthington-Snow engines.

DUTY TRIALS OF PUMPING ENGINES. For method of conducting tests of pumping engines, see p. 16-36.

3. POWER PUMPS

A power pump is driven by a source of power not integral with the pump, as a motor, turbine, steam or oil engine, water wheel, line-shaft, etc. The standard types generally used have a common crankshaft to which the pumps are connected. A duplex pump usually has two double-acting pumps with cranks set 90° apart. A triplex pump has three single- or double-acting pumps with cranks set 120° apart.

The advantages of these pumps over single pumps is a more even flow, which means more frequent but much less severe pulsations in the discharge, with consequently lower

stresses in discharge piping due to water hammer. Sizes and capacities of power pumps are given in Tables 8 and 9.

4. ROTARY PUMPS

A rotary pump is a positive displacement pump driven from an independent prime mover through a rotating shaft. The most commercially successful types are: 1. The gear pump. A pair of gears rotate in a casing. They entrain liquid between the teeth

Table 8.—Dimensions and Capacities of Horizontal Duplex Double-acting Piston Power Pumps for Ordinary Service

(Worthington Pump & Machinery Corp., Harrison, N. J., 1935)

Size, in.		Max. Pressure, lb. per sq. in.	Rev. per min.	Displacement		Pipe Sizes, in.		Pulley	
Diam. Piston	Stroke			Gal. per rev.	Gal. per min.	Suction	Discharge	Diam., in.	Face, in.
2 1/2	× 6	500	75	0.50	37.5	2 1/2	2	26	7 1/2
3	× 6	500	75	.72	54	2 1/2	2	26	7 1/2
3 1/2	× 6	390	75	1.00	75	2 1/2	2	26	7 1/2
4	× 6	250	75	1.25	94	3	2	26	7 1/2
5	× 6	125	75	2.00	150	4	3	26	7 1/2
5 3/4	× 6	125	75	2.70	202	4	3	26	7 1/2
2 1/2	× 10	800	60	0.71	42.5	4	3	36	8 1/2
3	× 10	700	60	1.08	65	4	3	36	8 1/2
3	× 10	800	60	1.08	65	4	3	42	10 1/2
3 1/2	× 10	515	60	1.51	91	4	3	36	8 1/2
3 1/2	× 10	800	60	1.51	91	4	3	42	10 1/2
4	× 10	400	60	2.03	122	4	3	36	8 1/2
4	× 10	715	60	2.03	122	4	3	42	10 1/2
4 1/2	× 10	315	60	2.61	157	4	3	36	8 1/2
4 1/2	× 10	565	60	2.61	157	4	3	42	10 1/2
5	× 10	250	60	3.25	195	4	3	36	8 1/2
5	× 10	460	60	3.25	195	4	3	42	10 1/2
6	× 10	175	60	4.77	286	5	4	36	8 1/2
6	× 10	250	60	4.77	286	5	4	42	10 1/2
7	× 10	130	60	6.53	392	6	5	36	8 1/2
7	× 10	150	60	6.53	392	6	5	42	10 1/2
8	× 10	100	60	8.52	514	6	5	36	8 1/2
8	× 10	150	60	8.52	514	6	5	42	10 1/2

Table 9.—Dimensions and Capacities of Vertical Triplex Single-acting Power Pumps for Ordinary Service

(Worthington Pump & Machinery Corp., Harrison, N. J., 1935)

Size, in.		Max. Pressure, lb. per sq. in.	Rev. per min.	Displacement		Pipe Sizes, in.		Pulley		Gear Ratio
Diam. Piston	Stroke			Gal. per rev.	Gal. per min.	Suction	Discharge	Diam., in.	Face, in.	
1 1/4	× 2	250	130	0.032	4.16	1	3/4	12	1 3/4	5–1
1 3/4	× 2 1/2	250	120	.078	9.36	1	1	12	3	5–1
2	× 3	250	112	.122	13.6	1 1/4	1 1/4	15	3 1/2	5–1
3	× 4	250	102	.367	37.4	2	1 1/2	18	3 1/2	5–1
3 1/2	× 4	200	102	.50	51	3	2 1/2	18	4 1/2	5–1
4	× 6	200	87	.97	84	4	3	20	5 1/2	5–1
4 1/2	× 6	250	87	1.24	107	4	4	20	6 1/2	5–1
5	× 6	200	87	1.53	133	4	4	20	6 1/2	5–1
6	× 8	150	75	2.93	220	5	4	36	6 1/2	4.91–1
6	× 8	250	75	2.93	220	5	5	36	8 1/2	5–1
7	× 10	150	68	5.00	339	8	6	36	8 1/2	4.88–1
7	× 10	200	68	5.00	339	8	6	42	7 1/2	5–1
7	× 10	300	68	5.00	339	8	6	54	9 1/2	5–1
8	× 10	125	68	6.5	443	8	6	36	8 1/2	4.88–1
8	× 10	150	68	6.5	443	8	6	42	7 1/2	5–1
8	× 10	250	68	6.5	443	8	6	54	9 1/2	5–1
9	× 10	125	68	8.25	561	8	8	42	7 1/2	5.28–1
9	× 10	200	68	8.25	561	8	8	54	9 1/2	5–1
10	× 12	200	60	12.2	732	10	8	66	13	5–1
12	× 12	150	60	17.6	1056	12	12	66	13	5–1
12	× 12	225	60	17.6	1056	12	12	84	15	5.14–1
14	× 12	100	50	24	1200	12	12	66	13	5–1
14	× 12	175	50	24	1200	12	12	84	15	5.14–1

and carry it around to ... the discharge
A single shaft drives one g ... gear. 2. The lobed imp
pump. This acts on the ... gear pump but has a limited numb
teeth or lobes. As one : cannot the other, an external set of driving gea
required. 3. The screw ... ing screw threads receive the liquid at one
end and force it through ... 4. A bored casing containing an
eccentric or cam-shaped rotor fitted ... swinging or rotating abutments or blades
to form the required seal between ... discharge connection.

ADVANTAGES OF ROTARY ... ed to reciprocating pumps the rotary pump is more compact and ... It has no valves or mechanism for converting rotary into reciproc ... occupies less space and requires less attention. Compared wi ... rotary pump delivers practically constant volume at a given ... of change in head. It will pick up its suction. It can ... ities and high heads, and can be run at low speed ... thick viscous liquid.

LIMITATIONS OF ROTARY PUMPS.—The greatest limitation is lack of means of compensating for wear. Close-fitting surfaces in metallic contact, or with slight clearance, are required. Wear or damage of these surfaces lowers capacity and efficiency of the pump. There being no adjustments or packings, normal condition can be restored only by installation of new parts. Foreign substances in the liquid are most injurious. Corrosion of finished surfaces produces an oxide that causes rapid wear. Since lubrication from outside would contaminate the liquid, running parts depend for their lubrication on the liquid being pumped. The rotary pump is best adapted to pump oils or other liquid with considerable lubricating value, although it often is used for water and liquids of low lubricating value. It is well adapted to load and discharge bulk cargo mineral or vegetable oils, to pump petroleum products in refineries, and lubricating oil for machinery.

CAPACITY AND PRESSURE of rotary pumps range from a fraction of a gallon per min. to 5000 gal. per min. In general, pressures are low, and many manufacturers limit their product to 100 or 200 lb. per sq. in. Some builders offer special construction for pressures up to 5000 lb. per sq. in.

EFFICIENCY of the better class of rotary pumps is good. In small-capacity pumps, efficiencies run considerably higher than those of centrifugal pumps. In large-capacity pumps at higher pressures, the efficiency may range from 85 to 90%. The speed of rotation and viscosity of liquid being pumped materially affect efficiency.

5. DEEP WELL PUMPS

Deep well turbine pumps in recent years have largely displaced the reciprocating type of deep well pump although the latter still is (1935) used. Present capacities of deep well turbine pumps range from 20-gal. per min., from 4- or 6-in. wells, to several thousand gallons per min. in 24-in. wells. Table 10 gives approximate capacities of both reciprocating and turbine pumps.

Turbine pumps have standardized impeller units, and are almost invariably multistage. The number of stages depends on the total head against which the pump operates. The drive usually is a direct-connected vertical motor. Usual speeds are 1750, 1450, or 1150 r.p.m., depending on size and current characteristics, except that with 4- and 6-in. pumps, 3500 r.p.m. is common. The motor is generally a vertical, hollow-shaft type with a ball or roller bearing in the upper bracket large enough to carry both

Table 10.—Approximate Capacities of Deep-well Pumps

Size of Well, Inside Diam. of Casing, in.	Largest Reciprocating Plunger, Commercial Size, in.	Reciprocating Pump Displacement, gal. per min., at 100 ft. per min. Piston Speed			Turbine Pump, gal. per min.	
		Single Plunger, Single-acting	Two Plunger	Triple Plunger	Medium Capacity	High Capacity
4	2 3/4	16	....	....	50	75
6	4 1/4	36	....	....	250	300
8	5 3/4	67	133	156	450	600
10	7 3/4	122	235	290	1000	1,200
12	8 3/4	156	310	370	1500	2,000
14	10 5/8	....	450	540	1800	3,000
16	11 5/8	....	550	640	2400	4,000
18	14	....	800	950	3000	6,000
20	15	....	....	1100	3500	8,000
24	19	....	....	1830	5000	12,000

the weight of the rotor and the pump thrust load. Other drives used are: Vertical, solid shaft motor with flexible coupling; vertical steam turbine; right-angle gear; multi-V-belt.

Impeller units usually are assembled in a group at the lower end of the pump and always are submerged, with at least 10 ft. of suction pipe and a strainer below them. The column pipe to conduct the water to the surface, and the line shafting to transmit power to the impeller shaft, connect the impeller units with the discharge fitting at ground level. Line shafting usually is enclosed in a cover pipe with bronze bearing at suitable intervals to avoid critical running speed. With such construction, bearings are lubricated by oil introduced at the top of the cover pipe. With an open line shaft and non-metallic (usually rubber) bearings, lubrication is by the water being pumped. Fig. 11 shows a typical Worthington deep well pump driven by a hollow shaft motor.

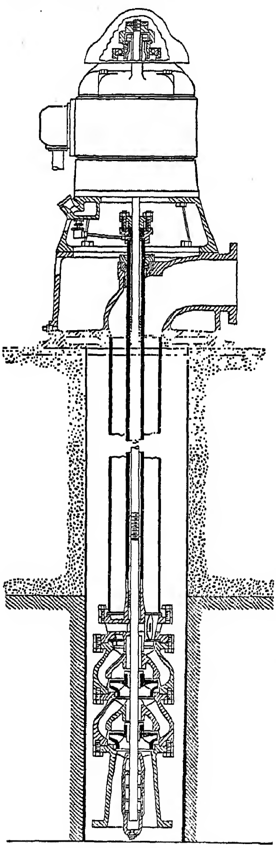


Fig. 11. Deep Well Turbine Pump

6. PUMPING BY COMPRESSED AIR

THE AIR-LIFT PUMP comprises a vertical water, or eduction, pipe, whose lower end is submerged in a well, and a smaller pipe delivering air into it at the bottom through a foot-piece. The rising column in the eduction pipe is a mixture of air and water, which is lighter than a column of water of equal height. Hence, the column in the pipe is raised above the level of the surrounding water. The *standing level* is the level of the water before pumping begins. The *pumping level* is the level to which the water recedes after pumping begins. The *lift* is the height of point of discharge above pumping level. *Submergence* is height from point of air injection to pumping level. If H = lift, ft., and S = submergence, ft., Submergence Ratio $= S/(S + H)$.

The upward velocity of each air-bubble in the rising air-water mixture in the eduction pipe is due to a force equal to the difference in weight of the bubble and of the water it displaces. The water may be raised a certain distance without flowing from the eduction pipe, the air escaping at the same rate that it is admitted. All its energy is dissipated in slippage without performing useful work. Slippage may be decreased by reducing the size of the bubbles and by increasing velocity of mixture in the eduction pipe. Other energy losses are: 1. Friction of mixture in eduction pipe. 2. Friction losses in the water prior to admitting air. 3. Kinetic losses in water due to velocity of discharge.

FRICTION LOSSES.—Gibson (Hydraulics, p. 703) gives a formula of the form $h_f = \{f(H + S)v^2\}/2gm$, where h_f = head lost in friction, ft. of column of mixture; f = coefficient of friction, H = lift, ft.; S = submergence, ft.; v^2 = mean square of the velocity in the pipe, ft. per sec.; m = mean hydraulic depth; g = 32.2. Gibson gives values of $f = 0.033$ in 3-in. pipes, with = 12.16, to $f = 0.023$ in a 12-in. pipe with $v = 6.5$. The value of f for a mixture of air and water is about 6 times that of f for water alone. Owens gives f = 0.025 in a 6 5/8-in. pipe with a mean value of $v = 13.3$.

Kinetic Losses may be reduced by a conical or expanding mouthpiece on the discharge outlet.

THEORY OF THE AIR-LIFT PUMP has been approached from three standpoints. They are discussed by A. W. Purchas (*Proc.* Inst. M.E., Nov., 1917) as follows: 1. Loren's theory which has been shown to contain so many errors as to be totally unreliable. 2. A theory which assumes the motive power operating the pump to be the difference in specific gravity between the water outside the eduction pipe and the mixture of air and water within it. 3. The Displacement Theory, which assumes each bubble of air to displace an equal volume of water in the eduction pipe, and therefore raises the water level therein a distance $h = V/A$, where V = volume of bubble and A = area of pipe.

Let H = lift, ft.; S = submergence, ft.; V_a = volume of air, cu. ft. per gal. of water, required at absolute atmospheric pressure; P_a = lb. per sq. ft., to raise a volume V_w, cu. ft. of water; P_s = absolute pressure of air leaving foot-piece, lb. per sq. ft.; h_f = friction loss in eduction pipe; h_s = head lost in slip in eduction pipe; h_k = head equivalent to velocity with which water leaves the outlet. Then for the theory based on the difference in specific gravity (2) Purchas gives:

$$\frac{V_a}{V_w} = \frac{H + h_f + h_s + h_k}{\div (P_s - P_a)}$$

Ivens (Pumping by Compressed Air) gives the following modification of this formula: $V_a = H \div C \log \{(S + 34)/34\}$, in which values of C are as follows:

H = 10–60	61–200	201–500	501–650	651–750
C = 245	233	216	185	156

Discussing the displacement theory, Purchas assumes that the injection of a bubble of air of volume V, causing a rise h in the height of water in the eduction pipe, does work equal to $^1/_2\ Vh$. The bubble expands isothermally as it rises, increasing h, but losing none of its energy until it escapes at the surface. This theory assumes that no energy is lost in slip until the bubble escapes. When a sufficient volume of air V_x has accumulated in the eduction pipe, so that $h = (V_x/A) > H$, the pump will discharge a volume of liquid equal to $A(h - H)$, and the further operation may be regarded as a head $(h - H)$ forcing liquid into that area of the eduction pipe not already filled with air.

The energy supplied by the air to the water is $P_a V_a \log_e (P_s/P_a)$ ft.-lb., and the work done on the water is 62.5 $V_w H$ ft.-lb., V_a and V_w being in cu. ft. per min. Therefore, if E is efficiency of the pump,

$$\{P_a V_a \log_e (P_s/P_a)\} E = 62.5 V_w H.$$

For values of P_s, see Submergence, below.

The volume of air V_p passing any point x in the eduction pipe per unit of time, is given by

$$V_p = V_1(P_a/P_s) + V_1(x/H)\{1 - (P_a/P_s)\},$$

where V_1 = volume of air delivered to pump, cu. ft. per sec., measured at pressure P_a and temperature of liquid being pumped, x = distance, ft., of the point under consideration above the point of air injection. Volume of air and area of pipe being known, average velocity may be determined.

SUBMERGENCE.—The most suitable value is found by experiment; it usually lies between 65% and 75%. The relation of submergence to lift has an important bearing on efficiency (Owens). A. W. Purchas (*Proc.* Inst. M. E., Nov., 1917, p. 613) gives the following values as a fair average of current practice.

Lift, ft	25	50	75	100	150	200	250	300
Ratio, Submersion/Lift...	4.0	3.0	2.33	2.00	1.70	1.38	1.22	1.00
Submersion, ft...........	100	150	175	200	255	275	305	300

Submergence Ratio.—The most suitable value is found by experiment, and after the piping is in place it is altered to suit. Submergence ratio ranges from 30 to 70%. It has an important effect on the cubic feet of air required, and thus reflects on the efficiency. Average values are given on p. 2–77.

AIR PRESSURE REQUIRED.—Based on the values of submergence ratio given above, Purchas gives for air pressure required, $P_s = P_a + 0.434S$, where P_s = absolute pressure of air, lb. per sq. in., under the submergence head S, P_a = absolute pressure of atmosphere, lb. per sq. in., S = submergence, ft. The starting pressure is higher than operating pressure. It is equivalent to the height of water level (at rest) above the end of the air pipe.

VELOCITY OF RISE OF BUBBLES.—Experiments by Owens (*Engg.*, Sept. 23, 1921) show that the velocity of rise of bubbles depends on their diameter. The following values were determined in water at 76° F.:

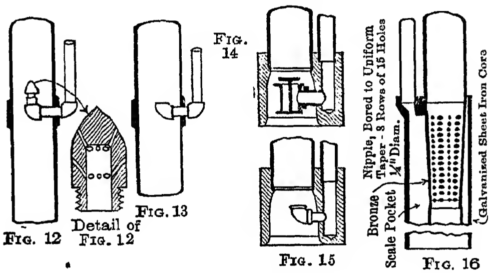
Diam. of Bubble (approx.) in........	0.05	0.10	0.125	0.15	0.25	0.38
Velocity of Rise, ft. per sec..........	0.53	0.57	0.67	0.70	0.80	0.81

The relative decrease in the rate of rise with the larger bubbles appears to be due to surface tension and to the flattening of the larger bubbles.

EFFICIENCY OF THE AIR-LIFT. (A. W. Purchas, *Proc.* Inst. M. E., Nov. 1917.)—An efficiency of 70% for all lifts up to 300 ft. may be assumed. This efficiency may not be attained with the first eduction pipe installed, which usually is put down to find the service pumping level and capacity of the well, and is not the final one. Two measures of efficiency are used: Overall efficiency, which includes the efficiency of the compressor; and eduction pipe efficiency. Let E_a = overall efficiency, E_p = efficiency of

eduction pipe, $A.Hp.$ = energy contained in the air at footpiece, $I.Hp.$ = indicated horsepower in air cylinders of compressors, and $W.Hp.$ = horsepower required to raise the water delivered by the pump. Then $E_a = W.Hp./I.Hp.$, and $E_p = W.Hp./A.Hp.$ The low efficiencies shown by earlier air-lift pumping plants probably was due to imperfect design and to inaccuracies in measurements of air used. Tests based on volumetric efficiency of the air compressor, the basis of nearly all earlier tests, probably were inaccurate, as this method of determining quantity of air supplied usually has a large percentage of error. In later tests the air has been measured by an orifice, with an accuracy within 2%.

FOOT-PIECES.—A. W. Purchas (*Proc.* Inst. M. E., Nov., 1917) says that design of foot-piece has small effect on efficiency of the pump if the fundamental laws of losses in transmission of air and water are followed in design, and if the air is injected in small bubbles. Attempts to use the injector principle are useless. Throttled air and water passages, and back pressure on the air lines generally result. Figs. 12 to 17 show



Foot Pieces for Air-lift Pumps

foot-pieces used in tests described in the paper. Those in Figs. 12 to 15 contract the water passage; those in Figs. 13 and 15 tended to become choked with rust and scale. The following points are desirable in foot-piece design: Avoid loss of air pressure in transmission from well-head to foot-piece; keep air pipe outside of eduction pipe; provide a pocket for rust and scale forming in the air pipe, which will prevent its lodging in the foot-piece; introduce air in the smallest bubbles possible; provide orifices that will not corrode and choke; proportion water passages so that there will be no sudden changes in velocity of the rising column. These conditions are filled by foot-pieces, Figs. 16 and 17.

PERFORMANCE OF AIR-LIFT PUMPS.—*Eng. News*, June 18, 1908, describes tests of 11 wells at Atlantic City, N. J. The wells were 10 in. diameter; water pipes, 4 to 5 1/4 in.; air pipes, 3/4 to 1 1/4 in.; maximum lift, 26 to 40 ft.; submergence, 37 to 49 ft.; submergence ratio, 0.9 to 1.8;

Table 11.—Tests of Air-Lift Pumps in California Oilfields

1. Lift when pumping, ft	295	297	302	312	314
2. Submergence, ft	315	314	308	298	296
3. Ratio submergence ÷ lift	1.17	1.17	1.02	0.95	0.94
4. Water raised per min., cu. ft	26.0	31.6	34.5	39.5	41.6
5. Free air per min., cu. ft	143	172	186	226	257
6. Free air per min., cu. ft. per cu. ft. of water	5.48	5.43	5.39	5.73	6.18
7. Water horsepower	14.5	17.7	19.7	23.3	24.7
8. Air horsepower, compressor cylinders	33.2	39.1	42.6	48.9	55.9
9. " " air leaving foot-piece	21.1	25.2	27.2	32.6	37.1
10. Overall efficiency, percent (Item 7/Item 8)	43.7	45.3	46.2	47.6	44.1
11. Eduction pipe efficiency, percent (Item 7/Item 9)	68.9	70.4	72.4	71.4	66.5
Energy Balance					
12. Lost in compression and after cooling of air, percent	36.7	35.4	35.7	32.6	32.6
13. Lost in transmission, receiver to well-head, percent		0.2	0.7	0.5	0.9
14. Lost in transmission, from well-head through foot-piece, percent		0.2	0.2	0.3	0.3
15. Lost in kinetic energy of discharge, percent	0.4	0.6	0.7	1.0	1.3
16. Energy equivalent to water raised, percent	43.7	45.3	46.2	47.6	44.1
17. Energy lost in friction, percent	2.4	3.8	4.5	6.0	6.5
18. Energy lost in slip and entry, percent	16.8	14.7	12.4	12.0	14.3

submergence, percent of length of pipe, 53 to 64. Capacity tests: Delivery, 3,544,900 gal. pe. 24 hr.; mean lift, 26.88 ft.; air pressure, 31 lb. per sq. in.; duty of whole plant, 19,900,000 ft.-lb. per 1000 lb. of steam used by compressors. Two-thirds capacity test: Delivery, 2,642,900 gal.· mean lift, 25.43 ft.; air pressure, 36 lb. per sq. in.; duty, 24,207,000.

Air-lifts for deep oil-wells are described by E. M. Ivens (*Trans.* A.S.M.E., xxxix, p. 341, The following results were obtained in wells at Evangeline, La.:

Cu. ft. free air per minute, compressor displacement	650	442	702	536
Cu. ft. oil pumped per minute	4.35	4.87	13.7	5.54
Air pressure at well, lb. per sq. in.	155	200	202	252
Pumping head, from oil level while pumping, ft.	1155	1081	1076	917
Submergence, from oil level to air entrance, ft.	358	412	419	583
Submergence ÷ total ft. of vertical pipe, %	23.6	27.6	28	39
Pumping efficiency, %	9.3	13.4	19.5	10.3

A. W. Purchas (*Proc.* Inst. M. E., Nov., 1917) describes tests on air-lift pumps forming the water supply system of the California Oilfields, Ltd., in Coalinga district, California. The foot-piece shown in Fig. 16 was used in these tests. Air pressure at compressor ranged from 132.5 to 136.4 lb. per sq. in. The results of the tests are given in Table 11.

Average practice (1935) is given by Worthington Pump & Machinery Corp. as follows:

Total lift, ft.	20–100	100–150	150–350	350–550	550–750
Submergence ratio, %	70– 50	65– 50	65– 50	50– 40	45– 30
Approximate air consumed, cu. ft. free air per gal. of water	0.21–0.55	0.45–0.70	0.55–1.30	1.30–2.30	2.30–3.65

Higher lifts and less submergence sometimes can be used.

The capacity of air-lifts may be taken as follows:

Water Pipe Diam., in.	Average Capacity, gal. per min.	Water Pipe Diam., in.	Average Capacity, gal. per min.
3	40– 80	8	500– 600
4	80–120	9	600– 700
5	140–200	10	700– 800
6	200–400	11	800– 900
7	400–500	12	900–1000

A pump for acid mine water (John S. Owens, *Engg.*, Sept. 23, 1921) has two stages, the first from the 210- to the 180-meter floor; the second from the 180- to the 150-meter floor. The foot-piece used, shown in Fig. 16, was designed to provide an unobstructed passage of water, to keep the size of bubbles at a minimum, to arrange for uniformity of pressure at top and bottom of foot-piece by means of a Venturi throat, so that air distribution could be controlled, and to avoid a sudden increase in the velocity at the foot-piece. The Venturi-throat-piece was perforated with from 550 to 600 ¼-in. holes. The angle of opening of the upper part of the throat did not exceed 5°. The diameter at the throat was made such that loss of head due to velocity at throat would somewhat more than balance static head due to increased depth at throat. The holes were drilled in rings around the throat, the number increasing toward the top, so that air was admitted in an increasing volume as velocity of water increased. Table 12 gives result of typical tests of this pump.

Table 12.—Tests of Air-lift Pumps for Mine Liquors

(Delivery pipe, 6 ⅝ in. diam.; air pipe, 3 in. diam.)

Throat-piece, No.*.	1	1	2	2	2	3	3	3
Submergence, ft.	156.0	156.0	200	202	156.0	202	202	158.5
Lift, ft.	105.0	105.0	99.5	98.0	105.0	97.0	97.0	97.7
Submergence, percent †	60.0	60.0	67.0	67.5	60.0	67.5	67.5	62.0
Water raised per min., cu. ft.	67.0	82.5	87.4	96.3	69.2	86.2	101.0	76.0
Free air per min., cu. ft.	218	296	224	261	199	219	287	222
Free air, cu. ft. per cu. ft. of water	3.23	3.59	2.57	2.71	2.88	2.65	2.84	2.92
Water horsepower	14.35	17.62	17.74	19.01	14.80	17.06	19.96	15.19
Air horsepower	24.8	33.6	27.66	32.57	22.4	27.37	35.30	24.84
Efficiency, percent	57.9	52.5	64.13	57.37	66.1	62.33	56.60	61.16

* Diameter of throat-pieces: No. 1, 4.59 in.; No. 2, 3.25 in.; No 3, 3.9 in. † Submergence, percent = (100 × submergence) ÷ (lift + submergence).

7. VACUUM PUMPS

THE PULSOMETER raises water by suction into the pump chamber by condensing the steam within it. The water then is forced into the delivery pipe by the pressure of a new quantity of steam on the surface of the water. Two chambers are used, one raising while the other discharges. Maximum suction lift is about 25 ft., the best working range being 7 to 15 ft. The discharge head may be as high as 150 ft. Steam pressure may be at least 50% higher than total water pressure.

DeVolson Wood (*Trans.* A.S.M.E. xiii) describes a test of a pulsometer 40 in. high, weighing 695 lb. Suction pipe, 3½ in. diam.; steam pipe, 1 in. diam. Table 13 gives the results of four tests.

Table 13.—Test of a Pulsometer

No. of Test	1			
Strokes per minute	71	60	57	64
Steam pressure in pipe before throttling, lb. per sq. in.	114	110	127	104.3
Steam pressure after throttling	19	30	43.8	26.1
Steam temperature after throttling, deg. F.	270.4	277	309.0	270.1
Steam superheating, deg. F	3.1	3.4	17.4	1.4
Steam used, lb	1617	931	1518	1019.9
Water pumped, lb	404,786	186,362	228,425	248,053
Water temp. before entering pump, deg. F.	75.15	80.6	76.3	70.25
Water temperature, rise of, deg. F.	4.47	5.5	7.49	4.55
Water head by gage on lift, ft.	29.90	54.05	54.05	29.90
Water head by gage on suction, ft.	12.26	12.26	19.67	19.67
Water head by gage, total (H), ft.	42.16	66.31	73.72	49.57
Water head by measure, total (h), ft.	32.8	57.80	66.6	41.60
Coefficient of friction of plant, h/H	0.777	0.877	0.911	0.839
Efficiency of pulsometer	0.012	0.0155	0.0126	0.0138
Efficiency of plant exclusive of boiler	0.0093	0.0136	0.0115	0.0116
Efficiency of plant if that of boiler be 0.7.	0.0065	0.0095	0.0080	0.0081
Duty, if 1 lb. evaporates 10 lb. water	10,511,400	13,391,000	11,059,000	120,36,300

A very high record of test of a pulsometer is given in *Engg.* Nov. 24, 1893. Height of suction, 11.27 ft.; total lift, 102.6 ft.; horizontal length of delivery pipe, 118 ft.; quantity delivered per hr., 26,188 British gal.; steam used per Hp.-hr., 92.76 lb.; work done per lb. of steam, 21,345 ft.-lb., equal to a duty of 21,345,000 ft.-lb. per 100 lb. coal if 10 lb. of steam be generated per pound of coal.

THE JET-PUMP works by means of the tendency of a stream or jet of fluid to drive or carry contiguous particles of fluid along with it. Capacities range up to about 700 gal. per min. In some experiments on a small scale, greatest efficiency of a jet-pump was when the depth from which the water was drawn by suction was about 0.9 of the height from which the water fell to form the jet. The flow up the suction pipe then was about $^1/_5$ that of the jet, and the efficiency consequently was $0.9 \times {}^1/_5 = 0.18$. This efficiency is low, but probably could be increased by improvements in proportions of the machine. The preferable height of suction should not exceed 15 ft.

THE INJECTOR when used as a pump has a very low efficiency. See p. 6–59.

8. THE HYDRAULIC RAM

The hydraulic ram is used where a considerable flow of water with moderate fall is available to raise a small portion of the flow to a height greater than the fall. It acts on the principle of water-hammer. The water in the pipe line (drive-pipe) escapes through a weighted "clack-valve." The velocity of flow through the valve is sufficient to lift and close the valve, whereupon the momentum of the water in the pipe line sets up sufficient pressure to force water past a check valve into an air chamber. As the excess pressure falls in the pipe line, the clack-valve reopens and the cycle repeats. When sufficient pressure has been built up in the air chamber, the water in it discharges through an outlet pipe.

Let Q = total water supplied to ram; q = quantity delivered by ram, both in cu. ft. per sec.; H = head on ram, ft., h = height above pond to point of delivery, L = length of drive-pipe, from pond to clack valve, ft., D = diameter of drive-pipe, ft. Then $D = \sqrt{1.63Q}$; L (minimum) $= 5h$ or $^3/_4\ (H + h)$.

Volume of air chamber = volume of delivery pipe.

Efficiency $= q\,(H + h)/QH$.

Efficiency depends on ratio of lift to fall. Clark gives the following figures:

Ratio lift to fall	4	6	8	10	12	14	16	18	20	22	24	26
Efficiency, percent.	72	61	52	44	37	31	25	19	14	9	4	0

L. F. Harza (*Bull.* 205, Univ. of Wis., 1908) reports tests of a Rife "hydraulic engine." Length of 2-in. drive-pipe, 85.4 ft. Supply head, H = 8.2 ft. Maximum velocity V in drive-pipe, ranged from 1.5 to 6 ft. per sec. Efficiencies were as follows:

h, ft.,	v = 1.5	2	3	4	5	6
2.6	..	30	20	15	7	0
12.3	60	60	45	33	18	0
23.2	60	65	53	40	20	0
43.5	55	60	53	42	30	0
63.1		60	55	50	28	0

Four tests by Prof. R. C. Carpenter (*Engg. Mechanics*, 1894) to determine length of stroke of clack-valve giving highest efficiency, showed highest efficiency at 60% travel, full travel being 15/16 in. The $1\frac{1}{2}$-in. supply pipe was 50 ft. long with 3 elbows. Supply head H ranged from 5.58 to 5.77 ft. Delivery head h was constant at 19.75 ft. Results of the tests were as follows:

Length of stroke, percent	100	80	60	46
Strokes per min	52	56	61	66
Total water pumped, lb.	297	296	301	297.5
Total water supplied, lb	1615	1567	1518	1455.5
Efficiency, percent	64.9	66.0	74.9	70.0

Four rams were tested by the Columbia Steel Co., Portland, Ore., 1908. Results are shown below. L = length, ft., and D = diam., in., of drive-pipe; l and d = length and diameter of delivery pipe; H = head on ram, ft.; h = height of lift above original source of supply; Q and q = respectively water supplied and delivered, gal. per min.

Size of Ram	H	$(h+H)$	Q	q	L	D	l	d	Effic. %
3-in.	4	28	35	3.5	28	3	1008	1 1/2	58.9
4 1/2-in.	5	45	100	8	40	4 1/2	325	....	72.0
6-in.	12	36.4	200	50.5	60	4 1/2	945	2 1/2	76.6
6-in.*	37.6	144.1	2809	516	192.5	6	1785	10 *	70.4

* Eleven rams discharge into one 10-in. jointed wood pipe. Loss of head in drive pipe, 0.7 ft.; in discharge pipe, 2.7 ft. On one test, 1 cu. ft. per sec. was delivered with less than 5 cu. ft. entering drive pipe; at 5 cu. ft. efficiency = 76.6%.

CENTRIFUGAL PUMPS

By V. deP. Gerbereux

A centrifugal pump comprises essentially an impeller or set of vanes enclosed in a housing or casing. Fundamentally, it adds energy in the form of velocity to an already flowing liquid; it does not, in the usual sense, add pressure. All basic considerations of the centrifugal pump must consider kinetic energy; all relations must be expressed in linear measure.

THE THEORY OF THE CENTRIFUGAL PUMP is developed from the common kinetic energy relation (law of falling bodies),

$$\{H_a + (V_a^2/2g)\} = \{H_b + (V_b^2/2g) + H_F\}.$$

Various authorities differ as to the development of the theory from these relations and much of the design is based on empirical constants developed by test. Figs. 1 and 2 illustrate a simple generally accepted consideration of the theory. Fig. 1 is a vector diagram of the exit of a centrifugal pump impeller. With zero speed at impeller intake, head energy is found to be the product of CU_2, *i.e.*, the tangential component of C_2 (or CU_2) and of U_2, whence $H = (CU_2U_2)/g$. With zero flow $CU_2 = U_2$, and theoretical head is $H = U_2^2/g$. As the quantity flowing increases, CU_2 decreases in value and the theoretical curve shown in Fig. 2 is obtained. This curve is based on a 100% transfer of energy and an infinite number of vanes of zero thickness. Practically, vane losses and shock losses occur at greater or less capacities than those for which the impeller has been designed. Fig. 2 shows how these losses affect the theoretical curve to produce the actual curve.

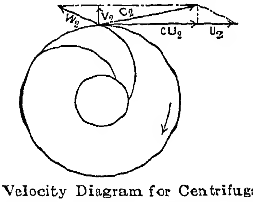


Fig. 1. Velocity Diagram for Centrifugal Pump

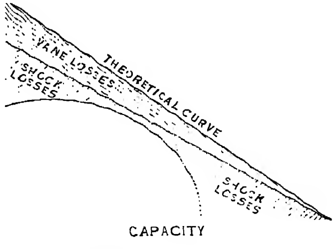


Fig. 2. Losses in a Centrifugal Pump

For actual pumping there must be an appreciable inlet speed; at the point where the liquid enters the impeller the inlet diagram is similar to Fig. 1. The theoretical head which the pump will deliver is $H = (CU_2U_2/g) - (CU_1U_1/g)$, and the theoretical head

at no flow in Fig. 2 would be $(U_2^2 - U_1^2)/g$ instead of U_2^2/g. The liquid leaving the impeller goes through a diffusion vane or a volute which converts into head energy most of the kinetic energy imparted by the impeller.

USEFUL APPROXIMATE RELATIONS for the selection or application of centrifugal pumps are:

Impeller diameter $D = (1840K_u\sqrt{H})/\text{r.p.m.}$, where K_u is a constant varying from 0.9 to 1.15 for various types of impellers.

At constant impeller diameter, $\dfrac{\text{Capacity } A}{\text{Capacity } B} = \dfrac{\text{r.p.m. } A}{\text{r.p.m. } B} = \dfrac{\sqrt{\text{head } A}}{\sqrt{\text{head } B}}$.

At constant r.p.m., $\dfrac{\text{Capacity } A}{\text{Capacity } B} = \dfrac{\text{Impeller diam. } A}{\text{Impeller diam. } B} = \dfrac{\sqrt{\text{head } A}}{\sqrt{\text{head } B}}$.

POWER REQUIREMENTS.—Besides vane and shock losses a centrifugal pump has the following hydraulic losses: Internal leakage, stuffing box leakage, entrance losses,

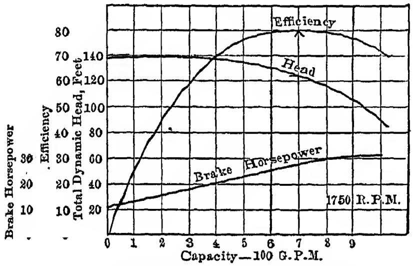


FIG. 3. Characteristics of Straight-vane Backward-angle Impeller

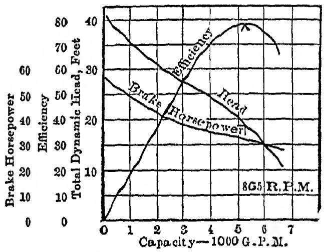


FIG. 4. Characteristics of Low-head Screw-vane Impeller

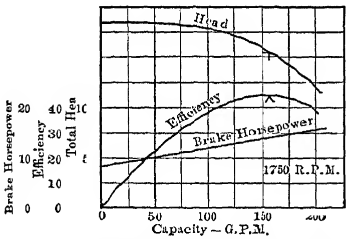
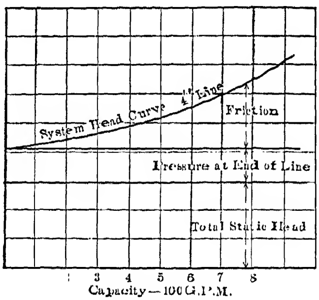


FIG. 5. Characteristics of Extreme Straight-vane High-head Impeller



. 6. System Head Curve

impeller entrance losses and conversion losses. Mechanical losses comprise casing friction, disc friction, stuffing box friction and bearing friction. The power required is the fluid horsepower $wh/33000$ plus power losses, or fluid horsepower divided by efficiency.

Useful formulas are:

B.Hp. = (gal. per min. $\times H_t \times$ specific gravity)/(3960 $\times E$)

= (gal. per min. $\times$ lb. per sq. in.)/(1715 $\times E$), where H_t = total head, ft., and E = efficiency expressed as a decimal.

THE CHARACTERISTICS OF A CENTRIFUGAL PUMP generally are shown in the form of curves. Head, efficiency and horsepower usually are ordinates, with capacity as abscissa at constant speed. The shape of the various curves depends on the type of impeller used. Centrifugal pumps are classified as to type by their specific speed, N_s, which is calculated from head, capacity and r.p.m. by the formulas:

On the cu. ft. basis, $N_s = (\text{r.p.m.}\sqrt{\text{gal.}}$

On the gal. per min. basis (G.P.M.), $N_s = (\text{r.p.m.}\sqrt{\text{gal. per min.}})/H^{3/4}$.

Any two centrifugal pumps having the same specific speed type will have like characteristics. This fact is used in the design of large pumps, as small size models can be made to predict the performance of larger sizes. The application of the laws of similitude as applied to hydraulics makes it possible to design the larger pump accurately by factoring the dimensions of the small model.

Fig. 3 shows characteristics of the most usual type of straight-vane backward-angle impeller which is non-overloading. This would have a specific speed of 58 on the cu. ft. basis or 1230 on the GPM basis. Fig. 4 shows characteristics of the low-head higher speed type of screw or Francis vane impeller having a specific speed of 327 on the cu. ft. basis or 6880 on the GPM basis. Fig. 5 shows the extreme straight-vane high-head impeller characteristics. The specific speed here would be 24.3 on the cu. ft. basis and 514 on the GPM basis.

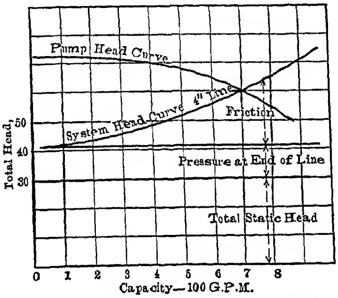


FIG. 7. Application of Pump to System Head Curve

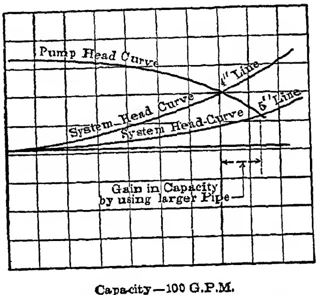


FIG. 8. Effect of Using Larger Piping

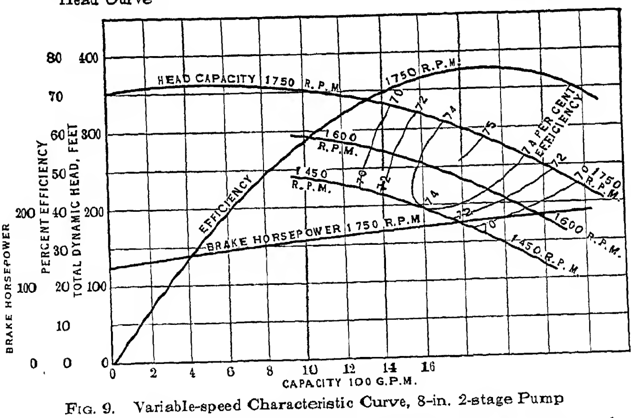


FIG. 9. Variable-speed Characteristic Curve, 8-in. 2-stage Pump

The application of a centrifugal pump to a pumping problem generally can be worked out graphically by plotting the system head curve in Fig. 6, then applying the head curve of the pump to the conditions, as shown in Fig. 7. If, in the future, additional capacity may be required a larger pipe could be used. See Fig. 8. Fig. 9 shows the effect of speed variation on the characteristics of a centrifugal pump.

The shape of the head-capacity curve can be varied within limits by variation in the impeller and casing design. For general service a curve similar to Fig. 3 is most satisfactory. Some services, where the head varies and the capacity should remain constant, require a *steeper* characteristic; others, as sprinkler or air washing service, require a *flatter* characteristic which keeps the head practically constant over a wide range of capacity. Boiler-feed pumps which generally must parallel require *steady* rising curves to as close to shutoff as design permits.

CLASSIFICATION.—Centrifugal pumps are classified into types and classes depending on: 1. The shaft position; 2. Number of stages; 3. Impeller arrangement; 4. Type

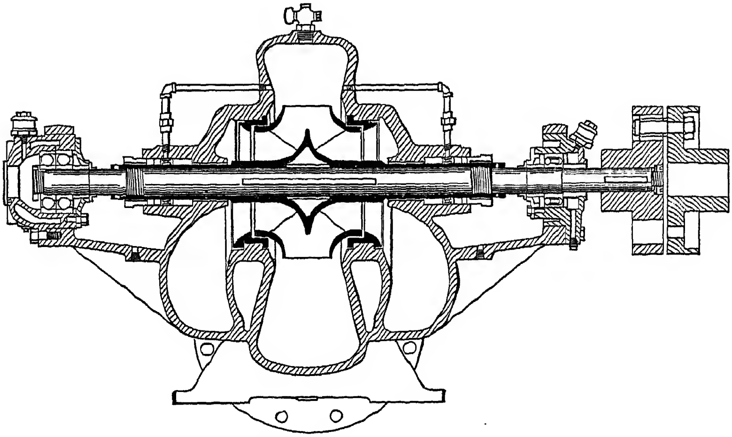


Fig. 10. Single-stage Double-suction Volute Pump

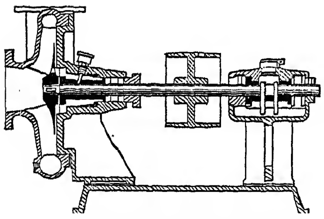


Fig. 11. Single-stage Single-suction Volute Pump

of drive; 5. Service for which they are designed. Table 1 shows the groups used commercially. The most common types are the horizontal single-stage double-suction volute pump for general service, Fig. 10; the single-stage single-suction volute, Fig. 11; and the multi-stage single-suction turbine, Fig. 12.

The Nominal Size of a Centrifugal Pump is the diameter of discharge opening. Some manufacturers give sizes as 6 × 8 meaning 6 in. discharge and 8 in. suction opening. It is not good practice to specify the nominal size of a centrifugal pump for a given capacity, as two equally reliable and efficient pumps for the same capacity

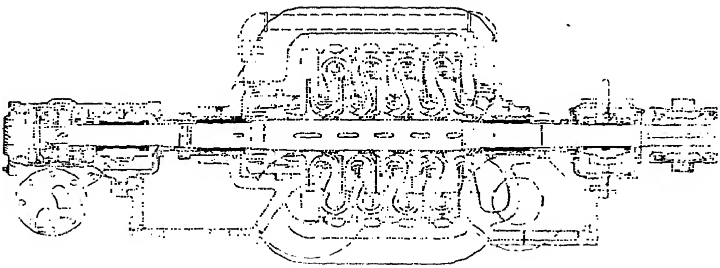


Fig. 12. Multi-stage Turbine Centrifugal Pump

and head would have different size openings according to the particular manufacturer's range of sizes.

Centrifugal pumps are available in sizes from 1/4-in. to over 100-in. nominal discharge, but all sizes over 36-in. are designed to order. Few manufacturers stock parts for sizes over 12-in. pumps. Centrifugal pumps also are available for heads from 5 ft. to over 2000 lb. per sq. in.

SUCTION LIFT.—Centrifugal pumps usually are designed for 15 ft. total dynamic suction lift. Any increase of suction lift (or decrease of absolute pressure available at suction opening of pump) will tend to decrease capacity of the pump. See Fig. 13. This is due to lack of available energy on the suction to create sufficient entrance velocity into the impeller. A pump operating with excessive suction lift is likely to be subject to cavitation and air binding, to be noisy in operation, and to wear rapidly. Centrifugal pumps can be designed for higher suction lifts than 15 ft. (usually at a lower r.p.m.), but only at a sacrifice of pump efficiency. Fig. 14 shows the limits of specific speed and suction lift agreed on as good practice by members of the Hydraulic Institute. Centrifugal pumps handling volatile liquids must be considered in the same class as pumps having a high suction lift.

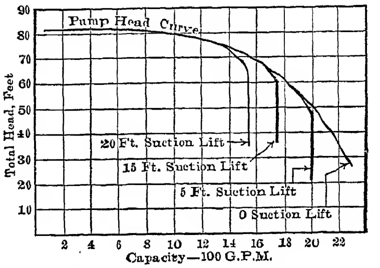


Fig. 13. Effect of Suction Lift

A centrifugal pump pumping a viscous liquid will show a loss of head and an increase in power required. Fig. 15 shows the effect of viscosity on a double-suction single-stage pump. In general, the centrifugal pump should not be used for viscosities over 1500 Saybolt Universal Seconds unless the capacity is over 2000 GPM, although this limit is fixed only by the extremely great decrease in efficiency with higher viscosities.

Uses of Centrifugal Pumps

GENERAL SERVICE PUMPS.—The so-called general service type of centrifugal pump comprises the horizontally-split case, double-suction volute, Fig. 10, the horizontally-split case single-suction volute, Fig. 11, and the multi-stage volute pump, Fig. 16. This service involves innumerable installation problems in the field of water supply, drainage, industrial and chemical plants. The pumps generally are considered as water pumps, and are standard fitted, *i.e.*, cast-iron casing, bronze impeller and bronze-covered steel shaft. The latest developments in this field are the monobloc pump (pump and motor built with a common shaft), and the use of automatic or self-priming (pump primed by automatic apparatus). Table 2 gives ratings and power requirements for general service pumps up to 5000 GPM and 440 ft. head.

BOILER-FEED PUMPS generally are multi-stage single-suction volute pumps, Fig. 16, for pressures up to 400 lb. per sq. in., and multi-stage single-suction turbine pumps, Fig. 12, for pressures of 200 to 1000 lb. per sq. in. For extreme pressures, pumps of similar design to Fig. 12 are used, but they are built heavier and in either vertically-split or solid-barrel type casings of cast steel or steel forgings.

Table 1.—Classification of Centrifugal Pumps

	Volute Pumps	Turbine Pumps
Shaft Position	Horizontal * Vertical	Horizontal * Vertical
Number of Stages	Single * Multi	Single Multi *
Impeller Arrangement	Single Suction Closed Single Suction Open Double Suction Closed * Double Suction Open	Single Suction Closed * Double Suction Closed
Diffuser Arrangement	....	Removable * Not Removable
Type of Drive	Belt or Chain Electric Motor * Steam Turbine Water Wheel Gasoline Engine Geared Steam Turbine Diesel Engine Geared Diesel Engine Steam Engine	Belt or Chain Electric Motor * Steam Turbine * Water Wheel Gasoline Engine Geared Steam Turbine Geared Diesel Engine Steam Engine

* Most common.

In general, boiler-feed pumps are selected for 5 to 25% greater pressure than the rated boiler pressure, and for at least 200% of the rated boiler capacity. Common practice, where the size of boiler permits, is to use several pumps in parallel to permit variation of capacity without too great a loss in efficiency. In this case, all pumps operating on the same system should have similarly-shaped characteristic curves and a steady rising characteristic. As centrifugal pumps for boiler-feed service often have steam turbine or dual drive the rising characteristic is desirable as it also permits closer regulation of the capacity by speed variation.

Table 2.—Ratings of General Service Centrifugal Pumps

(Worthington Pump & Machinery Corp., Harrison, N. J.)

Gal. per min.		Total Head, Feet														
		40	60	80	100	120	140	160	180	200	240	280	320	360	400	440
100	Size *	2	1 1/2	1 1/2	1 1/2	1 1/2	1 1/2	1 1/2	1 1/2	2	1 1/2	1 1/2	1 1/2	2	2	2
	Stages	1	1	1	1	1	1	1	1	1	2	2	2	2	2	2
	R.p.m.	1750	3500	3500	3500	3500	3500	3500	3500	3500	3500	3500	3500	3500	3500	3500
	Hp.†	2	3	5	5	5	7 1/2	7 1/2	10	10	15	15	15	30	30	30
200	Size *	2 1/2	2 1/2	2 1/2	2 1/2	2 1/2	2	2	2	2	2 1/2	2	2	2	2	2
	Stages	1	1	1	1	1	1	1	1	1	1	2	2	2	2	2
	R.p.m.	1750	1750	1750	1750	3500	3500	3500	3500	3500	3500	3500	3500	3500	3500	3500
	Hp.†	5	5	7 1/2	10	10	15	15	15	15	20	25	30	30	40	40
300	Size *	3	2 1/2	2 1/2	2 1/2	2 1/2	2 1/2	2 1/2	2 1/2	2 1/2	2 1/2	2 1/2	2 1/2	2 1/2	2 1/2	2
	Stages	1	1	1	1	1	1	1	1	1	1	1	1	1	1	2
	R.p.m.	1750	1750	1750	3500	3500	3500	3500	3500	3500	3500	3500	3500	3500	3500	3500
	Hp.†	5	7 1/2	10	15	15	15	20	25	25	30	40	40	40	50	60
400	Size *	3	3	3	3	3	3	3	3	3	3	3	3	3	3	3
	Stages	1	1	1	1	1	1	1	1	1	1	1	1	1	2	2
	R.p.m.	1750	1750	1750	1750	1570	1750	3500	3500	3500	3500	3500	3500	3500	1750	1750
	Hp.†	7 1/2	10	15	15	20	20	25	30	30	40	50	50	60	75	75
600	Size *	4	4	4	3	3	3	3	3	3	3	4	4	4	4	4
	Stages	1	1	1	1	1	1	1	1	1	1	1	1	1	1	2
	R.p.m.	1750	1750	1750	1750	1750	3500	3500	3500	3500	3500	3500	3500	3500	3500	1750
	Hp.†	10	15	20	25	30	40	40	40	50	50	75	75	75	100	125
800	Size *	4	4	4	4	4	4	4	4	4	4	4	4	4	4	5
	Stages	1	1	1	1	1	1	1	1	1	1	1	1	1	2	2
	R.p.m.	1750	1750	1750	1750	1750	3500	1750	1750	3500	3500	3500	3500	3500	1750	1750
	Hp.†	15	20	25	30	40	40	50	60	60	75	75	100	100	125	150
1000	Size *	5	5	4	5	4	4	4	4	5	6	4	4	4	5	6
	Stages	1	1	1	1	1	1	1	1	1	1	1	1	1	2	2
	R.p.m.	1750	1750	1750	1750	3500	3500	3500	3500	1750	1750	3500	3500	3500	1750	1750
	Hp.†	15	20	30	40	40	50	60	60	75	100	100	125	125	150	200
1500	Size *	6	6	6	6	6	6	6	6	6	6	6	6	6	6	6
	Stages	1	1	1	1	1	1	1	1	1	1	1	1	1	2	2
	R.p.m.	1750	1750	1750	1750	1750	1750	1750	1750	1750	1750	1750	1750	1750	1750	1750
	Hp.†	20	30	40	50	60	75	75	100	100	125	150	200	200	200	250
2000	Size *	8	8	8	8	8	8	8	8	8	8	8	6	6	8	8
	Stages	1	1	1	1	1	1	1	1	1	1	1	2	2	2	2
	R.p.m.	1750	1750	1750	1750	1750	1750	1750	1750	1750	1750	1750	1750	1750	1750	1750
	Hp.†	25	40	50	75	100	100	100	125	125	150	200	250	250	300	300
2500	Size *	10	10	8	8	8	8	8	8	10	10	10	8	8	8	8
	Stages	1	1	1	1	1	1	1	1	1	1	1	2	2	2	2
	R.p.m.	1150	1150	1750	1750	1750	1750	1750	1750	1750	1750	1750	1750	1750	1750	1750
	Hp.†	40	50	60	75	100	125	125	150	200	200	250	300	300	350	400
3000	Size *	12	10	10	10	10	10	10	10	10	10	10	....	....	....	....
	Stages	1	1	1	1	1	1	1	1	1	1	1	....	....	....	....
	R.p.m.	1150	1750	1750	1750	1750	1750	1750	1750	1750	1750	1750	....	....	....	....
	Hp.†	40	60	75	100	125	125	150	200	200	250	[illegible]	....	....	....	....
3500	Size *	12	10	10	10	10	10	10	10	10	10	....	....	....	....	....
	Stages	1	1	1	1	1	1	1	1	1	1	....	....	....	....	....
	R.p.m.	1150	1750	1750	1750	[illegible]	[illegible]	[illegible]	[illegible]	1750	1750	....	....	....	....	....
	Hp.†	50	75	100	100	[illegible]	[illegible]	[illegible]	[illegible]	250	250	....	....	....	....	....
4000	Size *	12	12	12	10	10	12	12	12	12	10	....	....	....	....	....
	Stages	1	1	1	1	1	1	1	1	1	1	....	....	....	....	....
	R.p.m.	1150	1150	1750	1750	1750	1750	1750	1750	1750	1750	....	....	....	....	....
	Hp.†	60	75	100	125	150	200	200	250	250	300	....	....	....	....	....
4500	Size *	12	12	12	12	12	12	12	12	12	....	....	....	....	....	....
	Stages	1	1	1	1	1	1	1	1	1	....	....	....	....	....	....
	R.p.m.	1150	1150	1750	1750	1750	1750	1750	1750	1750	....	....	....	....	....	....
	Hp.†	60	75	125	150	200	200	250	250	300	....	....	....	....	....	....
5000	Size *	12	12	12	12	12	12	12	12	12	....	....	....	....	....	....
	Stages	1	1	1	1	1	1	1	1	1	....	....	....	....	....	....
	R.p.m.	1150	1150	1750	1750	1750	1750	1750	1750	1750	....	....	....	....	....	....
	Hp.†	75	100	125	150	200	200	250	300	300	....	....	....	....	....	....

* Diameter, in., of discharge opening, nominal size. † Recommended motor rating for pumping water.

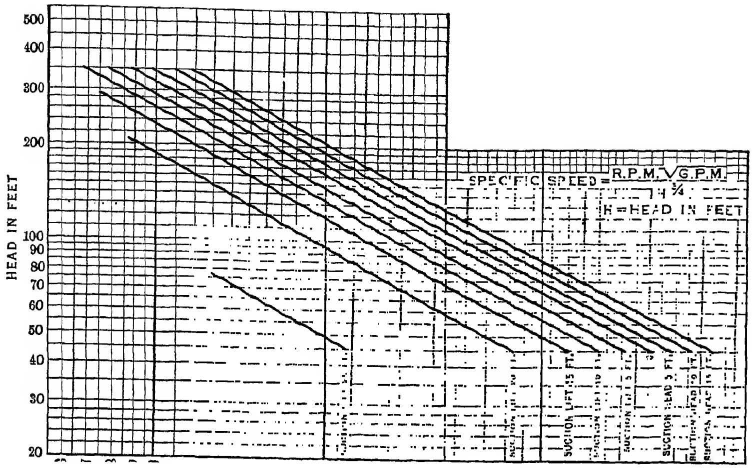


Fig. 14. Upper Limits of Specific Speed for Double-suction Single-stage Centrifugal Pumps

Centrifugal boiler-feed pumps usually pump preheated water, and the water should come to the pump under a positive head. For water at 212° F., centrifugal boiler-feed pumps should be installed with a minimum of 12 ft. submergence.

Table 3 gives ratings of standard boiler-feed pumps.

HOTWELL PUMPS.—Centrifugal condensate, or hotwell, pumps are generally multi-stage volute pumps, although the single-stage volute pump is used for low-head work. These pumps take their suction from a vacuum and usually pump water which is just below its vaporization point. They

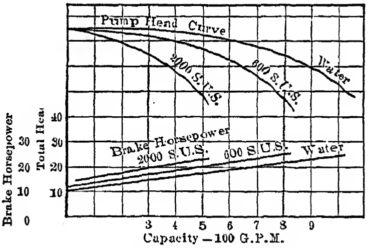


Fig. 15. Effect of Viscosity

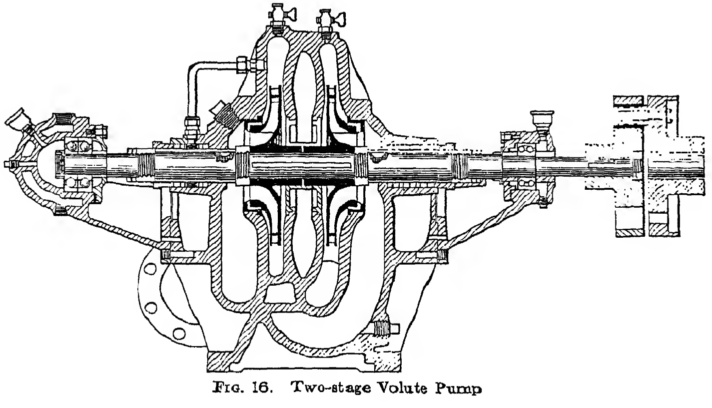


Fig. 16. Two-stage Volute Pump

are designed for extremely low suction velocities and generally low rotative speeds. The capacity which a given pump can deliver depends on the submergence available. Provision must be made to seal the stuffing box on the suction side to prevent air leakage and to vent the high spot of the suction passage to the vapor space above the hotwell.

Calculation of the total head must include vacuum in the condenser, plus the discharge pressure less the submergence on the suction, plus the suction pipe losses. In general 10% excess head should be allowed to steady the operation of the pump. The power required by a hotwell pump in actual operation cannot be measured on a cold water test due to the fluctuation in capacity and presence of vapor in the water. The efficiency of a given pump will be 5 to 10% less on hotwell service than on cold water service, which should be considered in selecting the driver.

SEWAGE AND SUMP PUMPS can be divided into two classes, the raw sewage pump (including sludge pumps), and the storm water pump. The latter is properly a drainage pump. Centrifugal pumps for pumping unscreened or partially screened sewage generally are single-stage volute, end suction pumps, either horizontal or vertical shaft. They usually are fitted with non-clogging type impellers which will pass solids of diameter 1 in. less than that of the discharge opening of the pump. Greater pump efficiency can be obtained if the pump is not required to pass the maximum size solids. Table 4 gives ratings of standard sewage type pumps. Centrifugal pumps can pump sewage sludge only when the sludge is sufficiently fluid to flow readily in the pump. The pump selected for sludge pumping should have extremely low velocities throughout.

Table 3.—Ratings of Centrifugal Boiler-feed Pumps
(Worthington Pump & Machinery Corp., Harrison, N. J.)

Gal. per min.		Total Head, lb. per sq. in.									
		100	200	300	400	500	600	700	800	900	1000
100	Size *	1 1/2	2	2	......	......	......	......	......	......	......
	Stages	2	2	2	......	......	......	......	......	......	......
	R.p.m.	3500	3500	3500	......	......	......	......	......	......	......
	Hp.†	10	30	50	......	......	......	......	......	......	......
200	Size *	2	2	2	2 1/2	2 1/2	......	......	......	......	......
	Stages	2	2	2	3	3	......	......	......	......	......
	R.p.m.	3500	3500	3500	3500	3500	......	......	......	......	......
	Hp.†	20	40	75	75	100	......	......	......	......	......
300	Size *	3	3	3	3	3	3	......	......	......	......
	Stages	2	2	2	3	3	3	......	......	......	......
	R.p.m.	1750	3500	3500	3500	3500	3500	......	......	......	......
	Hp.†	30	75	100	125	150	200	......	......	......	......
400	Size *	3	3	3	3	3	3	......	......	......	......
	Stages	2	2	2	3	3	3	......	......	......	......
	R.p.m.	1750	3500	3500	3500	3500	3500	......	......	......	......
	Hp.†	40	75	100	150	200	250	......	......	......	......
600	Size *	4	4	4	4	4	4	5	5	5	5
	Stages	2	2	2	3	3	3	3	4	4	4
	R.p.m.	1750	3500	3500	3500	3500	3500	3500	3500	3500	3500
	Hp.†	60	125	200	250	300	400	400	450	500	550
800	Size *	4	4	4	5	5	5	5	5	5	5
	Stages	2	2	2	4	5	6	3	4	4	4
	R.p.m.	1750	3500	3500	1750	1750	1750	3500	3500	3500	3500
	Hp.†	75	125	200	300	350	400	450	550	600	650
1000	Size *	5	6	5	5	5	5	5	5	5	5
	Stages	2	3	3	4	5	6	3	4	4	4
	R.p.m.	1750	1750	1750	1750	1750	1750	3500	3500	3500	3500
	Hp.†	100	200	250	350	450	500	600	650	750	850
1200	Size *	5	6	5	5	5	5	5	5	5	5
	Stages	2	3	3	4	5	6	7	7	8	8
	R.p.m.	1750	1750	1750	1750	1750	1750	1750	1750	1750	1750
	Hp.†	100	200	300	400	500	600	700	800	900	1000
1400	Size *	6	6	6	6	6	6	6	6	6	6
	Stages	2	2	3	3	4	4	5	5	6	7
	R.p.m.	1750	1750	1750	1750	1750	1750	1750	1750	1750	1750
	Hp.†	125	250	350	500	600	700	800	950	1100	1200
1600	Size *	6	6	6	6	6	6	6	6	6	6
	Stages	2	2	3	3	4	4	5	5	6	7
	R.p.m.	1750	1750	1750	1750	1750	1750	1750	1750	1750	1750
	Hp.†	125	250	450	550	650	800	950	1050	1200	1300

* Diam., in., of discharge, nominal size. † Recommended motor rating for boiler-feed service.

Vertical shaft pumps can be either dry pit or submerged; installation cost on the former is higher but it is accessible and easily maintained. The wet pit type can be mounted on a drop pipe supported from the motor floor, which makes it possible to remove the entire pump through the pit cover. See Fig. 17.

Sump pumps can be of the non-clogging type similar to the wet pit sewage pump, or can have a normal open or closed centrifugal impeller instead. The axiflo type, using a propeller wheel, makes a simple low cost unit.

IRRIGATION AND DRAINAGE PUMPS.—The centrifugal pump originally was designed to handle large quantities of water at low heads. One of its first uses was in the irrigation and drainage field. Earlier pumps were end-suction, single-stage volute pumps running at low speeds, direct connected or belted from steam engines. The use of double-suction pumps increased rotative speed, but due to the low heads, 15 to 30 ft., were also inefficient and expensive. Two main types are now in service, the propeller type, Fig. 18, and the mixed flow or screw type, Fig. 19. These are high specific speed types, having characteristics similar to Fig. 4, and run at relatively high speed. They are suitable for direct connection to Diesel engines and synchronous motors. Vertical pumps of both types often are used. The vertical axiflo propeller pump, Fig. 20, is the most common type for this service.

REFINERIES AND PIPE LINES.—Centrifugal pumps, extensively used in oil refineries, must be selected with consideration of the physical characteristics of the liquid handled, *viz.*, viscosity, specific gravity, temperature, vapor

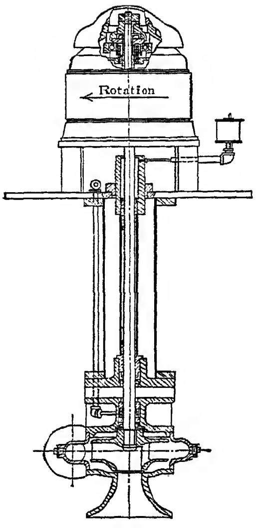


Fig. 17. Vertical Wet Pit Pump

Table 4.—Sewage Pump Ratings
(Worthington Pump & Machinery Corp., Harrison, N. J.)

Gal. per min.		Total Head, Feet					
		20	40	60	80	100	120
200	Size *	3	3	3	3	....	....
	R.p.m.	1150	1750	1750	1750	....	....
	Hp.†	2	5	7 1/2	10	....	....
400	Size *	3	3	3	3	3	3
	R.p.m.	1150	1150	1750	1750	1750	1750
	Hp.†	5	7 1/2	10	15	20	20
600	Size *	4	4	4	4	4	4
	R.p.m.	860	1150	1750	1750	1750	1750
	Hp.†	5	10	15	20	25	30
800	Size *	5	4	4	4	4	5
	R.p.m.	860	1150	1750	1750	1750	1750
	Hp.†	7 1/2	15	20	25	30	40
1000	Size	5	5	5	5	5	....
	R.p.m.	860	1150	1150	1750	1750	....
	Hp.†	10	15	25	30	40	....
1200	Size *	5	5	5	5	5	....
	R.p.m.	860	860	1150	1750	1750	....
	Hp.†	10	20	30	40	40	....
1400	Size *	6	5	5	5	5	....
	R.p.m.	860	1150	1150	1750	1750	....
	Hp.†	10	25	40	40	50	....
1600	Size *	8	5	5	5	8	....
	R.p.m.	860	1150	1150	1750	1150	....
	Hp.†	15	25	40	50	60	....
1800	Size *	10	6	6	6	8	....
	R.p.m.	680	1150	1150	1150	1150	....
	Hp.†	15	30	40	50	75	....
2000	Size *	10	6	6	8	8	....
	R.p.m.	680	1150	1150	1150	1150	....
	Hp.†	15	30	50	60	75	....

* Diameter, in., of discharge, nominal size. † Recommended motor rating.

pressure and the presence of grit, coke or foreign material. For non-volatile liquids at temperatures under 200° F., the so-called standard type of centrifugal pump often is used. As temperatures and pressures increase, special design features are required. Fig. 21 is a typical refinery pump with water-cooled stuffing boxes and bearings, vent

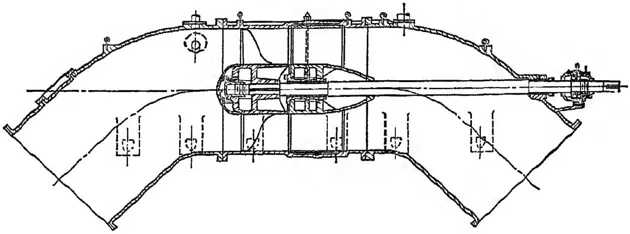


Fig. 18. Propeller Pump

connections and other features. Good practice uses cast or forged steel for pump casings subject to high pressures and temperatures over 450° F. Other special materials, as stainless steel and Monel metal often are required to withstand the action of the liquid and temperature. Multi-stage turbine pumps with forged steel barrel-type casings have been used for temperatures up to 1000° F. and pressures above 1500 lb. per sq. in.

Viscous Liquids reduce the output of a centrifugal pump and also reduce its efficiency. For pumps with 10-in. discharge and smaller, the maximum viscosity at which it is economical to use a centrifugal pump is 1500 Saybolt Universal Seconds.

Vaporizing Liquids require a careful study of suction conditions when selecting a refinery pump. It is imperative that sufficient submergence (static head on suction) be provided. Many refinery pump troubles are due to insufficient suction head.

Oil and Gasoline Pipe Lines require highly efficient, reliable centrifugal pumps. These are generally multi-stage volute pumps with loop inter-stage passages, designed for quick accessibility and repair. The prime mover usually is a squirrel-cage induction motor; Diesel or gas engines driving through step-up gears with ratios as high as 1 : 12 are not unusual. Usual station pressures

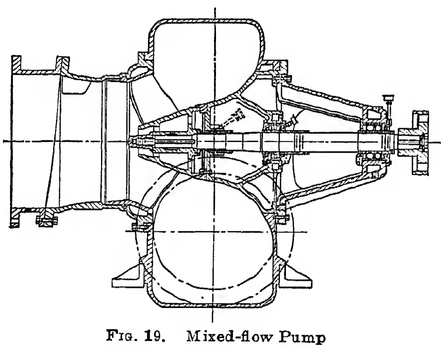


Fig. 19. Mixed-flow Pump

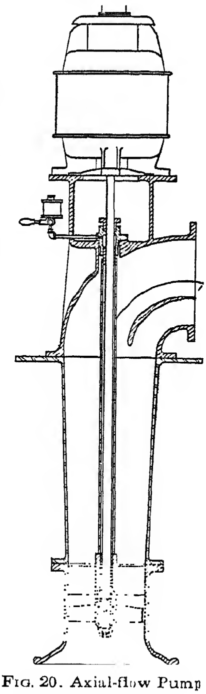


Fig. 20. Axial-flow Pump

are 800 to 900 lb. per sq. in., developed by a single pump or two pumps in series, depending on the flexibility required. Pump head curve should be matched against system head curve to obtain the best performance. Variation of viscosity of oil with temperature must be considered, together with the effect of this viscosity on both pump and system head curves.

PAPER MILLS.—Centrifugal pumps are used almost exclusively in paper mills. The principal services are general service, circulating, and boiler feeding.

The pumping of *paper stock* at consistencies as high as 8% (*i.e.*, 8% of solids) is common. The stock pump usually is an end-suction, single-stage, open-impeller pump, with the casing split on an angle to facilitate removal of the upper half without disturbing pipe connections. The bronze impeller must be of such design as will prevent dewatering of stock.

To pump various *acids and alkali solutions* in the paper mill requires centrifugal pumps of special metals, as stainless steel. The pumps should be of the simplest possible construction, usually end-suction, single-stage, open-impeller type with rugged bearings, heavy metal sections and designed for low rotative speed and low velocities.

DEEP WELL PUMPS.—(See also p. 2–73.) The centrifugal deep well pump is generally a turbine pump with diffusion vanes instead of volute casings. Since the diameter of the well is limited, the head per stage generally is low, but the number of stages is limited only by the depth of the well and the shaft diameter necessary to transmit the power. Most deep well turbine pumps are driven by hollow shaft motors, arranged to carry the pump thrust, no pump thrust bearing being used. The driving head can be made with a thrust bearing for steam turbine or gear drive when required. In figuring total head, friction loss in the drop pipe and draw-down of the water in the well must be considered.

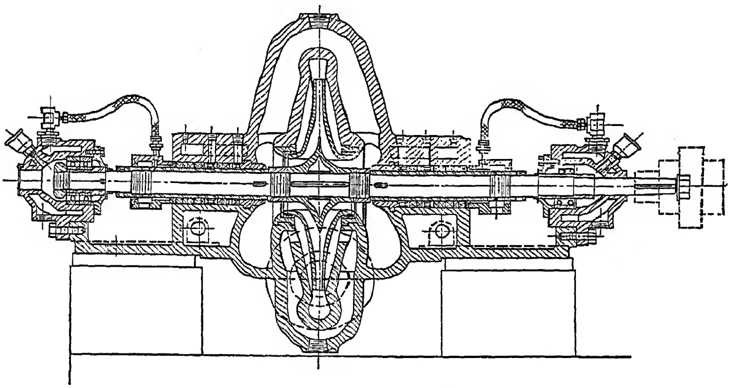


Fig. 21. Refinery-type Centrifugal Pump

FIRE PUMPS.—Centrifugal fire pumps are single or multi-stage, generally volute and designed to meet the requirements of the National Board of Fire Underwriters and the Factory Mutual Inspection Service, and have been approved by them. Table 5 gives typical ratings. *Booster pumps* are similar to fire pumps but for lower pressures, and no manifold is required. They also must be approved. *Tank filling* pumps are standard general service pumps of capacities up to 450 GPM and heads up to 231 ft., used for filling fire tanks. No special construction is required, but the pumps must be approved and listed.

Table 5.—Underwriter Centrifugal Fire Pump Ratings

For 100 lb. per sq. in. total head.

(Worthington Pump & Machinery Corp., Harrison, N. J.)

Pump Size, gal. per min.	No. of Stages	R.p.m.	Hp., Motor	Hp., Gasoline Engine	Diam. of Suction, in.	Diam. of Manifold Discharge, in.
500	2	1200–1750	60	75	6	6
500	1	3500	40	..	6	6
750	2	1150–1750	75	100	8	8
1000	2	1150–1750	100	125	8	8
1000	1	1700–1800	100	125	8	8
1500	2	1075–1750	150	168	10	10
1500	1	1700–1800	150	168	10	10

OTHER SERVICES.—Centrifugal pumps can be used for many services including contractors' service, with and without self-priming apparatus, jetting, in chemical plants.

breweries and distilleries, for refrigeration and air conditioning, and for marine work. The number of sizes, types and constructions is unlimited. Each field has a pump particularly adapted to it with the features required by that service. For example, contractors' pumps must be rugged and low in cost, brewery pumps must be quick cleaning, marine-type pumps must be light weight, etc. Manufacturers should be consulted for recommendations of the best type for a given service.

Installation and Operation

SELECTING LOCATION.—The pump should be easily accessible for inspection during operation. At the same time, the suction and discharge piping arrangement should be as simple as possible. In general, the pump should be located as close to the water supply as possible. Large size pumps should have head room for a crane or supporting structure sufficiently strong to lift the heaviest part of the unit. Motor-driven units should not be located in damp or moist places, unless special provision has been made for this condition.

SUCTION LIFT.—In ordinary pumping installation it is recommended that the suction lift should not exceed 15 ft. With warm water this must be reduced. Water at 212° F. must flow to the pump under net head of from 7 to 15 ft., depending on capacity at which pump is operating. Lift must be decreased or head increased 1 ft. for every 1000 ft. above sea level.

FOUNDATION.—The foundation should be heavy enough to afford permanent rigid support at all points of the baseplate and to absorb any normal amount of vibration that may develop from any cause. Concrete foundations built up from solid ground are the most satisfactory. In building the foundation ample allowance for grouting should be made. Foundation bolts of specified size should be accurately located from drawings or template and surrounded by a pipe sleeve, three or four diameters larger than the bolt. See Fig. 22.

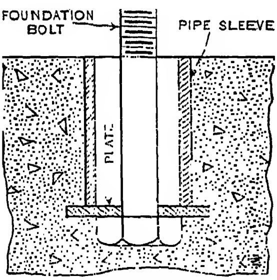


Fig. 22. Foundation Bolt

When the unit is mounted on steel work or other structure, it should be set directly over, or as near as possible to the supporting beams and walls and be so supported that the baseplate cannot be distorted and alignment disturbed by any yielding or springing of the structure.

ALIGNMENT.—Correct alignment is absolutely essential to successful operation. A flexible coupling is no excuse for misalignment. Its purpose is to eliminate transmission of end thrust from one machine to the other, and to compensate for slight changes in alignment that may occur during normal operation.

Factory Alignment.—Every unit that is assembled should be accurately aligned by the manufacturer, by placing the baseplate on a surface plate, leveling the machine pads, and shimming where necessary under pump and driver feet to obtain perfect alignment. Since all baseplates are elastic, no matter how deep or heavy they may be, shop alignment must be reproduced after the unit is erected on its foundation, to insure proper mechanical operation.

Field Alignment.—Pumps, with few exceptions, are shipped on their baseplates and it usually is unnecessary to remove pump and driver from the baseplate while leveling. The pumping unit should be placed on the foundation, supported by plates and wedges near the foundation bolts, allowing from 3/4 to 2 in. opening between bottom of baseplate and top of foundation for grouting. Coupling bolts, and one pump bearing cap, preferably on the inboard bearing, and the corresponding top half bearing bushing, should be removed. With a small spirit level on the pump shaft, the wedges under baseplate are adjusted to bring pump shaft level and pump nozzles into a true vertical plane. When the pump is level and at the desired position for piping connections, foundation bolts on the pump end should be pulled down sufficiently to hold pump in position.

During the above procedure little attention is given to the driver half of the unit; in most cases the coupling halves will be appreciably out of alignment. In any event the alignment must be checked and corrected by adjusting wedges under the driver end of the baseplate to bring driver and pump half couplings in perfect alignment. Alignment can be checked by a straight edge across the top and sides of the coupling, at the same time checking faces of the coupling halves for parallelism by a thickness gage or set of

feelers. See Fig. 23. When both couplings are perfectly true on both faces and outside diameters, exact alignment will show the distance between faces to be the same at all points, and a straight edge will lay squarely across the rims at any point. If the faces are out of parallel, the thickness gage or feelers will show a variation at different points. If one coupling is higher than the other the amount may be determined by the straight edge and feelers.

It sometimes may be found that couplings are not perfectly true and not the same diameter. In checking trueness of either coupling, revolve it while holding the other half stationary, checking alignment at each quarter turn of the half being rotated. Next revolve and check alignment of the half previously held stationary. If any variation is found in either of the half couplings, proper allowance must be made in aligning the unit.

Where pumps are driven by steam turbines which are subject to temperature changes in operation, final alignment should be made with the driver heated to operating temperature. If this is not possible at time of alignment, proper allowance should be made. Similarly, where the pump is subject to expansion in service, due to handling hot liquids, allowance must be made for expansion. In any case the cold alignment should be checked when hot, and adjusted as required before the unit is placed in service.

Clearance between coupling faces should be set so that they cannot strike, rub or exert pull on either pump or driver. The amount of clearance may vary with the size and type of coupling used. The best rule is to allow sufficient clearance for unhampered endwise movement of the shafts of either element to the limit of their respective thrust bearing clearances. On motor-driven units, the magnetic center of the motor will determine the running position of the motor half-coupling. It is well to check this point by operating the motor light before coupling bolts are replaced. If electric power is not available to operate the motor light, the motor shaft should be moved in both directions as far as bearings will permit, and shaft adjusted centrally between these limits, assembling the unit with the correct gap between coupling halves. Coupling bolts are not put in until piping is complete and driver has been tested for correct direction of rotation.

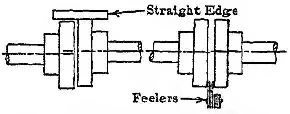


Fig. 23. Method of Aligning Pump and Driver

Alignment must be checked after the pump has been completely piped, as pumps are easily sprung and pulled out of position by drawing up bolts in piping flanges not brought squarely together before bolts are tightened. Suction and discharge piping should be so supported that they cannot exert a strain or pull on the pump. Pipe strains are a common cause of misalignment, hot bearings, worn couplings and vibration.

GROUTING.—The baseplate usually is grouted in before piping connections are made, but, in special cases, the reverse is permissible. The usual mixture for grouting a machine is one part pure cement, two parts sand, with sufficient water to cause the mixture to flow freely under the baseplate. A form should be built around the outside of the baseplate to hold the grout and provide sufficient head to assure a flow of the mixture underneath the entire baseplate. Grout should be allowed to set for 48 hours.

SUCTION PIPING.—By far the greater number of centrifugal pump troubles, outside of misalignment, can be traced to a faulty suction line. Suction piping should never be of less diameter than the full size of pump suction opening. It should be as short and direct as possible. Where a long suction line cannot be avoided, the size of the piping should be increased. Air pockets or high spots in a pump suction line invariably cause trouble. Piping must be so laid as to provide a continual rise without high spots from source of supply to pump. The suction pipe should be submerged when the water is at its lowest level, with the pump operating. Large pipes usually are submerged four times their diameter, while small pipes require from 2 to 3 ft. submergence. Suction pipes should be blanked off and hydrostatically tested for air leaks before starting.

Foot Valve.—A foot valve may be installed on the end of the suction pipe for convenience in priming or where the pump runs intermittently. The size and type of foot valve should be selected to avoid excessive friction loss through the valve. An ordinary swing check valve should not be used as a foot valve.

A Strainer should be placed in the suction pipe to prevent lodgment of foreign material in the impeller. It is important to take every possible precaution to protect the pump against clogging. Clear and free opening through strainer should be three to four times the area of the suction pipe. If the strainer is subject to frequent clogging, the suction pipe should be accessible. For large pumps, removable screens should be placed at the entrance to the suction well.

DISCHARGE PIPES should be installed with a check valve and gate valve near pump outlet. The check valve gives protection against excessive pressure or water hammer.

On units with no suction foot valve, a check valve eliminates the possibility of the pump running backward if the driver ceases to function.

JACKET PIPING.—If a thrust bearing is supplied, the jacket cooling water pipe should be so connected that the supply enters at the bottom and discharges from the top. For observing flow and regulating the amount of jacket water, it is good practice to pipe the discharge to an open flow into a funnel connected to a drain.

DRAIN PIPING.—All drain connections should be piped to a pump pit or suction well.

GLAND PIPING to each gland should have a valve to control the amount of water necessary to feed and seal each gland and to permit enough seepage to lubricate the gland.

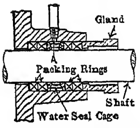


Fig. 24. Pump Packing

PACKING.—Square, soft, asbestos graphited packing, or equal, is recommended for either hot or cold service. Flax packing should not be used under any circumstances on centrifugal pumps having bronze shaft sleeves, as rapid wear of sleeves may result. To pack a centrifugal pump properly, rings should be cut slightly short, to prevent butting of the ends and buckling. Each ring should be inserted separately and pushed as far into the stuffing box as possible by means of the gland. The splits of each successive ring should be placed 90° apart. After two or three rings are inserted, the water seal cage should follow so as to bring it directly under the water pipe connections. See Fig. 24, at *A*. Enough additional packing then is inserted to allow the gland to be loosely drawn up.

WATER SEAL.—Stuffing boxes are water sealed to prevent air from leaking in and to keep the packing wet. A composition cage, in halves, inserted in the middle of the packing space is supplied with water piped directly from the discharge, if the pump handles clear, cold water.

Independent water sealing should be provided from an independent source of cold, clean water, at a pressure of 15 to 40 lb. under the following conditions: 1. On suction lifts of 20 ft. to 25 ft. Ordinarily operation above 15 ft. is not recommended. 2. When discharge pressure is less than about 10 lb., or 23 ft. head. 3. When pumps are handling water over 175° F. 4. When the water is muddy, sandy or gritty. 5. In general, when liquid handled is other than water, as acid, juice, molasses and sticky substances. 6. On all hotwell pumps.

PRIMING.—A centrifugal pump never should be operated unless filled with water. The pump cannot deliver water when operated dry, and wearing rings are likely to seize and cause serious damage. The only exception to this rule is a pump specially designed to start dry, and fitted with water-lubricated wearing rings whose water supply is from a source external to the pump.

With a foot valve on the suction pipe the pump can be primed by venting the high point of the pump casing and admitting water from an outside source until suction pipe and casing are completely filled. All air must be exhausted from suction pipe and pump casing, as any entrapped air will interfere with operation or hinder the pump from lifting water.

If the suction pipe has no foot valve, an ejector of good design and ample size, should be connected to the high point on pump casing. Ejectors can be operated with air, steam or water. With water, the supply must be of sufficient pressure to give high velocity through the ejector.

Priming is unnecessary if the suction supply is under sufficient head to fill the pump casing, which is vented at the high point to permit the water to expel entrapped air.

Another satisfactory method of priming is to use a motor-driven rotary vacuum pump. Several types, which will handle both air and water successfully, are available. If a dry vacuum pump is used for priming, the vacuum must be prevented from drawing over any water.

STARTING AND OPERATING.—Before starting the first time, the prime mover should be disconnected and tested for correct direction of rotation, as indicated by the arrow on the pump casing.

All bearings should be washed with kerosene, and oil rings should be free to turn. Bearings should be filled with a good grade of dynamo oil. After checking jacket cooling water piping on thrust bearings to make sure that inlet is at bottom and discharge at top, the supply may be turned on, allowing an ample flow to keep the bearing cool. The water seal supply to glands next is turned on and the pump primed. The pump should not be run unless primed and full of water, as interior parts which depend on water for lubrication may be injured. Careful final inspection should be made of all parts before starting, and then the pump may be brought to speed. After a pressure is built up on the discharge, the discharge gate valve should be opened slowly.

A compound gage and a pressure gage connected respectively to the suction and discharge of the pump, and mounted at a convenient place, will greatly help the operator.

During the routine operation of pumps, bearings should be occasionally inspected to insure that sufficient oil is in the oil pockets, and also that oil rings are turning freely and supplying enough oil to shaft and bearings. Bearings should be drained and washed, and the oil renewed every week or ten days during the first month of operation, and as often after that as conditions warrant.

Stuffing box glands must be so adjusted as to permit a slight seepage of water out of the stuffing box at all times during operation. Otherwise, the packing will cause excessive wear on shaft sleeves.

LOCATING TROUBLES.—The most common troubles, together with their causes, which may occur with a centrifugal pump are: 1. *Failure to Deliver Water.* *a.* Pump not primed. *b.* Insufficient speed. *c.* Discharge head too high. *d.* Suction lift too high (over 15 ft.); check with vacuum gage. *e.* Impeller plugged up. *f.* Wrong direction of rotation. 2. *Insufficient Capacity.* *a.* Air leaks in suction or stuffing boxes. *b.* Speed too low. *c.* Total dynamic head higher than that for which pump is rated. *d.* Suction lift too high (over 15 ft.); check with vacuum gage. *e.* Impeller partially clogged. *f.* Insufficient suction head for hot water. *g.* Mechanical defects, *i.e.*, wearing rings worn, impeller damaged, casing packing defective. *h.* Foot valve too small or restricted by trash. *i.* Foot valve or suction pipe not deeply enough immersed. 3. *Insufficient Pressure.* *a.* Speed too low. *b.* Air in water. *c.* Mechanical defects, *i.e.*, wearing rings worn, impeller damaged, casing packing defective. 4. *Pump Loses Water after Starting.* *a.* Leaky suction line. *b.* Water seal plugged. *c.* Suction lift too high (over 15 ft.). *d.* Air or gases in water. 5. *Pump Overloads Driver.* *a.* Speed too high. *b.* Total dynamic head lower than rating—pumping too much water. *c.* Liquid pumped of different specific gravity and viscosity than that for which pump is rated. *d.* Mechanical defects. 6. *Pump Vibrates.* *a.* Misalignment. *b.* Foundation not sufficiently rigid. *c.* Impeller partially clogged, causing unbalance. *d.* Mechanical defects, *i.e.*, shaft bent, rotating element binds, worn bearings.

DRIVERS.—The electric motor is the most common driver for the centrifugal pump. The low starting torque of the centrifugal pump (10 to 25% of full load) avoids complex problems and permits use of the standard squirrel-cage induction motor. Synchronous motors must be selected for a pull-in torque to match pump characteristics. Variable speed motors and two-speed motors are desirable for some types of installations. Motors selected can be rated for the actual pump brake-horsepower. The steam turbine, either direct-connected or geared, is a common driver for the centrifugal pump. It has the advantage of variable speed, easily regulated by steam flow. Steam turbines selected can be rated for the actual pump brake-horsepower. Internal combustion engines (Diesel, gas and gasoline) involve a little care in installation, particularly when step-up gears are used. Critical speeds should be avoided by the selection of flexible couplings which do not transmit torsional vibration. The engine should be selected for 25% greater rating than the pump brake-horsepower.

Water-wheels, when used as drivers for centrifugal pumps, must be accurately matched to the pump load. Both machines have similar torque-speed relations and a careful study of both is necessary to proper installation. The water-wheel-driven unit is simple to operate and control when properly installed.

The use of belts (excepting V belts) and steam engines as pump drivers is rapidly losing favor with a few exceptional cases. Both are relative low-speed drivers and therefore are not applicable to higher speed pumps. Flat belts should be selected for not over 4500 ft. per min. belt speed. Steam engines should be selected with 25% reserve horsepower.

TURBINE-DRIVEN PUMPS.*—Steam-turbine-driven centrifugal pumps have greater flexibility than motor-driven units. The speed may be readily adjusted through any range to obtain the desired head, while alternating-current motors have a limited number of steps in speed. It is often desirable and economical to operate large centrifugal pumps at speeds lower than the most economical turbine speed, using mechanical reduction gears with flexible couplings on both sides of the gear. Such geared turbines operate at speeds up to 7000 r.p.m. Speed ratios in reduction gears range, for single reduction sets, from 25 to 1 down to 3 to 1.

Geared turbine-driven centrifugal pumps have practically supplanted vertical triple-expansion and horizontal cross-compound pumping engines for steam-driven municipal pumping equipment. Geared pumps cost much less and require less space and attendance. With the steam pressures, temperatures, and vacua commonly used (1934), the

* Contributed by A. G. Christie.

efficiency of a turbine-driven pump can be made to equal the best performance of a triple-expansion engine where the power exceeds 500 Hp.

Turbine-driven centrifugal pumps for regular water supply can be fitted for fire service by providing a device to cut out the ordinary governor, and furnishing additional steam nozzles to be opened by hand. Turbine speed then increases and automatically delivers a greater quantity of water against the higher head needed for fire service.

Larger boiler-feed pumps exceeding 1500 GPM capacity may be gear driven. Other boiler-feed pumps are direct driven, as pump speed can be made high enough for direct connection with reasonable efficiency. These pumps have been made as small as 100 GPM at speeds of 6000 to 7000 r.p.m. Small turbine-driven feed pumps are used on some locomotives. The turbines generally operate non-condensing, the exhaust being used to heat feedwater.

Main feed pumps in central stations generally are motor-driven centrifugal pumps, but a turbine-driven pump is used for standby services. Centrifugal boiler-feed pumps cause no pulsations on the boiler feed lines, and usually have check valves on the discharge side. A turbine-driven boiler-feed pump has a limiting head against which it will discharge at constant speed. Excessively high pressures on boiler-feed lines thus are impossible. Such pumps, even in small sizes are much more economical of steam than direct-acting steam pumps.

Turbine-driven centrifugal pumps for direct connection are built for various speeds. For central station work, centrifugal boiler-feed pumps, practically always, run at either 1750 r.p.m. or 3500 r.p.m. depending on the size.

Standard governors for turbines driving centrifugal pumps are constant speed with an emergency overspeed trip. Either of two controls also may be used; *viz.*, regulators to maintain constant pressure on the discharge line or controls to maintain a constant differential pressure between feed line and boiler drum.

When hot water is furnished to the suction of a turbine-driven centrifugal boiler-feed pump, a positive excess of pressure is needed at the pump suction for a margin to cover inaccuracies in estimating entrance loss, pipe friction, velocity head and total lift.

Section 3

HEAT

HEAT TRANSMISSION

By W. J. King

EVAPORATORS AND EVAPORATION

By W. L. Badger

DRYERS AND DRYING

By Francis E. Finch

KILN DRYING OF LUMBER

By L. V. Teesdale

HEAT INSULATION

By P. Nicholls

THERMODYNAMICS

By A. G. Christie

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HEAT

METHODS OF MEASURING TEMPERATURE

REFERENCES.—Technologic Paper No. 170, U. S. Bureau of Standards, Washington. High Temperature Measurements, Burgess-LeChatelier, John Wiley & Sons, New York. Pyrometry: A Symposium, Am. Inst. Min. & Met. Engrs., New York. Practical Pyrometry, Ferry, John Wiley & Sons. Methods of Measuring Temperature, Griffiths, J. B. Lippincott & Co. Power Test Code, A.S.M.E., Part 3, Instruments and Apparatus. Pyrometry, W. P. Wood and J. M. Cork, McGraw-Hill.

The several methods of measuring temperature are: 1. Liquid-in-glass thermometers, in which the expansion or contraction of a liquid, as mercury, alcohol, etc., indicates change in temperature. 2. Bourdon tube thermometers, in which the expansion or contraction of a liquid or gas in a tube actuates a Bourdon spring, which moves an indicating pointer or a recording pencil in accordance with the change in temperature. 3. Resistance thermometers in which change in temperature changes the electrical resistance of a calibrated wire. 4. Thermocouple pyrometers, in which a change of temperature sets up electromotive force at the junction of two dissimilar wires. 5. Radiation pyrometers, which absorb radiation of all wave lengths from the body whose temperature is being measured, the temperature of which is determined by the temperature attained by the absorber in the pyrometer. 6. Optical pyrometers, which determine temperature by comparing the luminosity of the body whose temperature is being measured with the luminosity of a calibrated source. 7. Pyrometric cones, comprising a series of cones of refractory material, graded according to temperature. These are used by noting which one of the series softens and bends when exposed to the temperature being investigated. The ranges of these several methods of measuring temperature are given in Table 1.

Table 1.—Range of Various Methods of Measuring Temperature

Type of Instrument	Range, deg. F.	Limits of Error, deg. F.
LIQUID-IN-GLASS THERMOMETER		
Ordinary glass, mercury-filled	−35 to 750	0.5 to 7
Corning normal, Corning Borosilicate, Jena 16III, Jena 59III, mercury- and nitrogen-filled	−35 to 925	0.5 to 7
BOURDON TUBE THERMOMETER		
Liquid-filled type		
Alcohol-filled	−50 to 300	2 to 10
Mercury-filled	−38 to 1000	2 to 10
Vapor-pressure type		
Alcohol-filled	200 to 400	2 to 10
Ether-filled	100 to 300	2 to 10
Sulphur-dioxide-filled	20 to 250	2 to 10
Aniline	400 to 700	2 to 10
Gas-filled type		
Nitrogen-filled	−60 to 1000	2 to 10
RESISTANCE THERMOMETER	−400 to 1800	Depends on indicating instrument
THERMOCOUPLE PYROMETERS		
Base metal	300 to 2000	Depends on indicating instrument
Rare metal	300 to 2800	2 to 20.
RADIATION PYROMETER	1000 and up	20 to 30 for black body conditions
OPTICAL PYROMETER	1500 and up	15 to 35 for black body conditions
PYROMETRIC CONES	1100 to 3600	15 to 30 (approx.) in best makes.

1. TEMPERATURE SCALES

The ideal temperature scale, known as the thermodynamic scale, was defined by Kelvin as follows: " The absolute values of two temperatures are to one another in the proportion of the heat taken in to the heat rejected in a reversible thermodynamic engine, working with a source and a refrigerator at the higher and lower temperatures, respectively." The interval between the freezing and boiling points of pure water being as 100° C., gives the size of the unit degree. The scales in practical use are supposedly in

general agreement with the thermodynamic scale or differ from it by small known quantities.

The high temperature scale from 100° C. to about 1500° C. is defined by the expansion of nitrogen in the constant-volume gas thermometer. This scale varies from the thermodynamic scale by about 0.05° at 200° C., 0.3° at 600° C. and 1° at 1200° C. This thermometer is employed only as a fundamental standard instrument to determine the number of so-called fixed points defined by the melting or boiling points of various chemical compounds and elements. Table 2 gives the fixed points utilized by the U. S. Bureau of Standards.

The limit of accuracy of the gas thermometer is 1550° C. Temperature scales beyond this point must be extrapolated by means of the Stefan-Boltzman or Wien-Planck radiation laws. The latter is $J = c_1\lambda^{-5}(e^k - 1)^{-1}$, where J = energy corresponding to wave length λ; T = absolute temperature of radiating black body; e = base of the natural system of logarithms; c_1 = constant depending on apparatus, distance from the source, etc.; c_2 = universal radiation constant = 14,330; $k = c_2/\lambda T$. The Bureau of Standards (Technologic Paper No. 170) has provisionally adopted the following means of defining the temperature scale: Interval −40 to +450° C. by the platinum resistance thermometer calibrated in ice, steam and sulphur vapor; from 450° to 1100° C. by fixed points of copper, antimony and zinc; above 1100° C. extrapolation of Wien's law, using the melting point of gold = 1063° C. and c_2 = 14,350. (See Optical Pyrometry, p. 3–12.)

Table 2.—Standard Fixed Points for Thermometer Calibration

(U. S. Bureau of Standards, 1933)

Material	Temperature		Change of State.* Note 1	Atmosphere	Material	Temperature		Change of State.* Note 1.	Atmosphere
	Deg. C.	Deg. F.				Deg. C.	Deg. F.		
Oxygen	−182.97	−297.35	B	......	Sulphur	444.6	832.3	B	......
Carbon dioxide	−78.5	−109.3	S	......	Antimony	630.5	1166.9	M or F	*Note 2
Mercury	−38.87	−37.97	M or F	......	Aluminum	660.1	1220 2	M or F	Note 2
Water	0.00	32.00	M or F	Air	Silver	960.5	1760.9	M or F	Note 2
Sodium sulphate	32.38	90.28	T	......	Gold	1063.0	1945.4	M or F	......
Water	100.00	212.00	B	......	Copper	1083.0	1981.4	M or F	Note 2
Naphthalene	217.96	424.33	B	......	Nickel	1455.	2651.	M or F	Vacuo
Tin	231.9	449.4	M or F	......	Palladium	1555.	2831.	M or F	*Note 3
Benzophenone	305.9	582.6	B	......	Platinum	1773.	3223.	M or F	......
Cadmium	320.9	609.6	M or F	......	Iridium	2454.	4449.	M or F	......
Lead	327.4	621.3	M or F	......	Tungsten	3400.	6152.	M	Vacuo
Zinc	419.5	787.1	M or F	......	......	......	......	......	......

* Note 1. B = Boiling point; M = Melting point; F = Freezing point; T = Transition point; S = Sublimation point. Note 2. Protected from O_2. Note 3. Protected from O_2 and H_2.

2. THERMOMETRY

Liquid-in-Glass Thermometers

THE FAHRENHEIT THERMOMETER generally is used in English-speaking countries and the Centigrade, or Celsius, thermometer in countries that use the metric system. In many scientific treatises in English, however, the Centigrade temperatures also are used, either with or without their Fahrenheit equivalents. The Réaumur thermometer is slightly used in Continental Europe.

In the Fahrenheit thermometer the freezing-point of water is taken at 32°, and the boiling-point of water at mean atmospheric pressure at the sea-level, 14.7 lb. per sq in., is taken at 212°, the distance between these two points being divided into 180°. In the Centigrade and Réaumur thermometers the freezing-point is taken at 0°. The boiling-point is 100° in the Centigrade scale, and 80° in the Réaumur.

1 Fahrenheit degree	= 5/9 deg. Centigrade	= 4/9 deg. Réaumur.
1 Centigrade degree	= 9/5 deg. Fahrenheit	= 4/5 deg. Réaumur.
1 Réaumur degree	= 9/4 deg. Fahrenheit	= 5/4 deg. Centigrade.
Temperature Fahrenheit	= 9/5 × temp. C. + 32°	= 9/4 temp. R. + 32°.
Temperature Centigrade	= 5/9 (temp. F. −32°)	= 5/4 temp. R.
Temperature Réaumur	= 4/5 temp. C.	= 4/9 (temp. F. − 32°).

FORMULAS FOR TEMPERATURE CONVERSION.—To convert Fahrenheit temperatures to Centigrade temperatures, the Taylor Instrument Companies, Roches-

ter, N. Y., give the following formulas, which are based on the fact that the values of the readings on both Centigrade and Fahrenheit scales are the same at $-40°$.

To Convert Centigrade to Fahrenheit:

$$T_c = \{(T_f + 40) \times {}^5/_9\} - 40 = 0.556\ (T_f + 40) - 40.$$

To Convert Fahrenheit to Centigrade:

$$T_f = \{(T_c + 40) \times {}^9/_5\} - 40 = 1.8\ (T_c + 40) - 40,$$

where T_f and T_c are Fahrenheit and Centigrade temperatures, respectively.

TYPES OF LIQUID-IN-GLASS THERMOMETERS.—Thermometers may be classified by:

1. Expanding fluid. The fluids commonly used and their working range are as follows:

Fluid	Mercury	Alcohol	Toluol	Pentane
Working range, deg. F.	−30 to 925	−100 to 250	−150 to 120	−300 to 70

2. Form.—Etched-stem, either armored or unarmored; industrial, with metal bulb and scale case surrounding thermometer, either of the straight stem or angle stem types; enclosed scale in which the scale, of paper or glass, is enclosed in an outer tube surrounding the thermometer stem.

3. Glass.—Corning normal or Jena 16^{III} for ordinary and medium temperatures; Corning boro-silicate or Jena 59^{III} for high temperatures (above 750° F.).

4. Immersion.—Graduated for full immersion of thread of expanding fluid in stem, or for partial immersion. For high temperatures they usually are graduated for partial immersion.

RANGE AND ACCURACY OF THERMOMETERS.—The usual limits of error of good etched-stem thermometers totally immersed in a calibrating bath in comparison with standard thermometers are:

Working temp. range, °F.	−300 to 70	−35 to 32	32 to 300	212 to 600	600 to 925
Limits of error, °F.	1 to 2	0.5 to 1	1	2	6 to 7

If the thermometers are used in wells or if graduated for full immersion and only partly immersed, even though stem corrections are applied, the errors usually will be greater than those given above. Partial immersion thermometers usually will be less accurate than the limits above noted.

CORRECTION FOR EXPOSED STEM OF MERCURIAL THERMOMETER.—If a portion of the mercury column thermometer is exposed to atmospheric temperature, the bulb being at a different temperature, a correction K must be added to the observed temperature, unless the thermometer has been calibrated for the particular conditions under which it is used.

$$K = n\beta\ (T - t) \text{ for Centigrade thermometers,}$$
$$= 0.55555\ n\beta\ (T - t) \text{ for Fahrenheit thermometers,}$$

where n = number of degrees of exposed stem; β = apparent coefficient of expansion of mercury in the glass; T = observed temperature; t = mean temperature of exposed stem. For temperatures up to 212° F.; the Smithsonian Institution gives the following values of β:

Jena 16^{III} or Greiner or Friedrich resistance glass, $\beta = 0.000159$; Jena 59^{III}, $\beta = 0.000164$; for glass of unknown composition, take $\beta = 0.000155$.

CALIBRATION OF THERMOMETERS.—Thermometers may be calibrated by comparison with a standard or secondary standard thermometer, both being immersed in a bath which can be heated to a desired temperature range. A standard thermometer is one that has been calibrated by the U. S. Bureau of Standards. A secondary standard thermometer is one that has been calibrated against a standard thermometer. The thermometer may be calibrated degree by degree, or at wider intervals ranging as high as 50°, depending on accuracy desired and purpose for which the thermometer is to be used. The standard thermometer should be immersed in the bath to the same depth as when it was calibrated, and the other thermometer to the depth at which it will be used. Ample time should be allowed to permit the thermometer under test to attain the temperature of the bath, in order to compensate for difference in time lag between it and the standard.

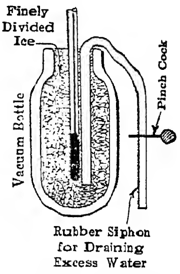


Fig. 1. Method of Determining Freezing Point

When calibrating by comparison with a standard thermometer, the temperature intervals can be made as small as desired if the bath has suitable control.

Another method of calibration is checking the fixed points, *i.e.*, freezing and boiling points on the thermometer.

Table 3.—Temperature Conversion—Centigrade to Fahrenheit

Values for Interpolation in Table										
Deg. C....	1	2	3	4	5	6	7	8	9	10
Deg. F....	1.8	3.6	5.4	7.2	9.0	10.8	12.6	14.4	16.2	18.0

Deg.→ C.↓	0	10	20	30	40	50	60	70	80	90
	Degrees, Fahrenheit									
−200	−328	−346	−364	−382	−400	−418	−436	−454	......	......
−100	−148	−166	−184	−202	−220	−238	−256	−274	−292	−310
−0	+32	+14	−4	−22	−40	−58	−76	−94	−112	−130
+0	+32	+50	+68	+86	+104	+122	+140	+158	+176	+194
100	212	230	248	266	284	302	320	338	356	374
200	392	410	428	446	464	482	500	518	536	554
300	572	590	608	626	644	662	680	698	716	734
400	752	770	788	806	824	842	860	878	896	914
500	932	950	968	986	1004	1022	1040	1058	1076	1094
600	1112	1130	1148	1166	1184	1202	1220	1238	1256	1274
700	1292	1310	1328	1346	1364	1382	1400	1418	1436	1454
800	1472	1490	1508	1526	1544	1562	1580	1598	1616	1634
900	1652	1670	1688	1706	1724	1742	1760	1778	1796	1814
1000	1832	1850	1868	1886	1904	1922	1940	1958	1976	1994
1100	2012	2030	2048	2066	2084	2102	2120	2138	2156	2174
1200	2192	2210	2228	2246	2264	2282	2300	2318	2336	2354
1300	2372	2390	2408	2426	2444	2462	2480	2498	2516	2534
1400	2552	2570	2588	2606	2624	2642	2660	2678	2696	2714
1500	2732	2750	2768	2786	2804	2822	2840	2858	2876	2894
1600	2912	2930	2948	2966	2984	3002	3020	3038	3056	3074
1700	3092	3110	3128	3146	3164	3182	3200	3218	3236	3254
1800	3272	3290	3308	3326	3344	3362	3380	3398	3416	3434
1900	3452	3470	3488	3506	3524	3542	3560	3578	3596	3614
2000	3632	3650	3668	3686	3704	3722	3740	3758	3776	3794
2100	3812	3830	3848	3866	3884	3902	3920	3938	3956	3974
2200	3992	4010	4028	4046	4064	4082	4100	4118	4136	4154
2300	4172	4190	4208	4226	4244	4262	4280	4298	4316	4334
2400	4352	4370	4388	4406	4424	4442	4460	4478	4496	4514
2500	4532	4550	4568	4586	4604	4622	4640	4658	4676	4694
2600	4712	4730	4748	4766	4784	4802	4820	4838	4856	4874
2700	4892	4910	4928	4946	4964	4982	5000	5018	5036	5054
2800	5072	5090	5108	5126	5144	5162	5180	5198	5216	5234
2900	5252	5270	5288	5306	5324	5342	5360	5378	5396	5414
3000	5432	5450	5468	5486	5504	5522	5540	5558	5576	5594
3100	5612	5630	5648	5666	5684	5702	5720	5738	5756	5774
3200	5792	5810	5828	5846	5864	5882	5900	5918	5936	5954
3300	5972	5990	6008	6026	6044	6062	6080	6098	6116	6134
3400	6152	6170	6188	6206	6224	6242.	6260	6278	6296	6314
3500	6332	6350	6368	6386	6404	6422	6440	6458	6476	6494
3600	6512	6530	6548	6566	6584	6602	6620	6638	6656	6674
3700	6692	6710	6728	6746	6764	6782	6800	6818	6836	6854
3800	6872	6890	6908	6926	6944	6962	6980	6998	7016	7034
3900	7052	7070	7088	7106	7124	7142	7160	7178	7196	7214

Examples: 1347° C. = 1340° C. + 7° C. = 2444° F. + 12.6° F. = 2456.6° F.
− 194° C. = − 190° C. + (− 4° C.) = − 310° F. + (− 7.2° F.) = − 317.2° F.
1852° F. = 1850° F. + 2° F. = 1010° C. + 1.11° C. = 1011.11° C.
− 226° F. = − 220° F. + (− 6° F.) = − 134.44° C. + (− 3.33° C.) = − 137.77° C.

Freezing Point is checked by surrounding the bulb with a mixture of water and ice, drawing off most of the water. The difference between the reading and the 32° mark on the thermometer is the error in location of the freezing point. In general, natural ice is satisfactory, but for a very high degree of accuracy ice made from distilled water should be used.

Boiling Point.—To determine boiling point, the thermometer is suspended in a bath so that it is entirely surrounded by the vapor of boiling distilled water at atmospheric pressure. Boiling should be continued until the water is free of air, which will be denoted by temperature indicated by the thermometer remaining constant, barometric pressure being unchanged. The thermometer should not project into the liquid. The difference between the thermometer reading and temperature as shown by the steam tables for the barometric pressure at which the test is made will be the error in the marking of the boiling point. Figs. 1 and 2 show types of apparatus recommended by the A.S.M.E.

Table 4.—Temperature Conversion—Fahrenheit to Centigrade

Values for Interpolation in Table										
Deg. F...	1	2	3	4	5	6	7	8	9	10
Deg. C...	0.555	1.111	1.666	2.222	2.777	3.333	3.888	4.444	5.000	5.555

Deg. → F ↓	0	10	20	30	40	50	60	70	80	90
	Degrees, Centigrade									
−400	−240.00	−245.56	−251.11	−256.67	−262.22	−267.78	−277.33	......	......	......
−300	−184.44	−190.00	−195.56	−201.11	−206.67	−212.22	−217.78	−223.23	−228.89	−234.44
−200	−128.89	−134.44	−140.00	−145.56	−151.11	−156.67	−162.22	−167.78	−173.33	−178.89
−100	−73.33	−78.89	−84.44	−90.00	−95.56	−101.11	−106.67	−112.22	−117.78	−123.33
−0	−17.78	−23.33	−28.89	−34.44	−40.00	−45.56	−51.11	−56.67	−62.22	−67.78
+0	−17.78	−12.22	−6.67	−1.11	+4.44	+10.00	+15.56	+21.11	+26.67	+32.22
100	37.78	43.33	48.89	54.44	60.00	65.56	71.11	76.67	82.22	87.78
200	93.33	98.89	104.44	110.00	115.56	121.11	126.67	132.22	137.78	143.33
300	148.89	154.44	160.00	165.56	171.11	176.67	182.22	187.78	193.33	198.89
400	204.44	210.00	215.56	221.11	226.67	232.22	237.78	243.33	248.89	254.44
500	260.00	265.56	271.11	276.67	282.22	287.78	293.33	298.89	304.44	310.00
600	315.56	321.11	326.67	332.22	337.78	343.33	348.89	354.44	360.00	365.56
700	371.11	376.67	382.22	387.78	393.33	398.89	404.44	410.00	415.56	421.11
800	426.67	432.22	437.78	443.33	448.89	454.44	460.00	465.56	471.11	476.67
900	482.22	487.78	493.33	498.89	504.44	510.00	515.56	521.11	526.67	532.22
1000	537.78	543.33	548.89	554.44	560.00	565.56	571.11	576.67	582.22	587.78
1100	593.33	598.89	604.44	610.00	615.56	621.11	626.67	632.22	637.78	643.33
1200	648.89	654.44	660.00	655.56	671.11	676.67	682.22	687.78	693.33	698.89
1300	704.44	710.00	715.56	721.11	726.67	732.22	737.78	743.33	748.89	754.44
1400	760.00	765.56	771.11	776.67	782.22	787.78	793.33	798.89	804.44	810.00
1500	815.56	821.11	826.67	832.22	837.78	843.33	848.89	854.44	860.00	865.56
1600	871.11	876.67	882.22	887.78	893.33	898.89	904.44	910.00	915.56	921.11
1700	926.67	932.22	937.78	943.33	948.89	954.44	960.00	965.56	971.11	976.67
1800	982.22	987.78	993.33	998.89	1004.4	1010.0	1015.6	1021.1	1026.7	1032.2
1900	1037.8	1043.3	1048.9	1054.4	1060.0	1065.6	1071.1	1076.7	1082.2	1087.8
2000	1093.3	1098.9	1104.4	1110.0	1115.6	1121.1	1126.7	1132.2	1137.8	1143.3
2100	1148.9	1154.4	1160.0	1165.6	1171.1	1176.7	1182.2	1187.8	1193.3	1198.9
2200	1204.4	1210.0	1215.6	1221.1	1226.7	1232.2	1237.8	1243.3	1248.9	1254.4
2300	1260.0	1265.6	1271.1	1276.7	1282.2	1287.8	1293.3	1298.9	1304.4	1310.0
2400	1315.6	1321.1	1326.7	1332.2	1337.8	1343.3	1348.9	1354.4	1360.0	1365.6
2500	1371.1	1376.7	1382.2	1387.8	1393.3	1398.9	1404.4	1410.0	1415.6	1421.1
2600	1426.7	1432.2	1437.8	1443.3	1448.9	1454.4	1460.0	1465.6	1471.1	1476.7
2700	1482.2	1487.8	1493.3	1498.9	1504.4	1510.0	1515.6	1521.1	1526.7	1532.2
2800	1537.8	1543.3	1548.9	1554.4	1560.0	1565.6	1571.1	1576.7	1582.2	1587.8
2900	1593.3	1598.9	1604.4	1610.0	1615.6	1621.1	1626.7	1632.2	1637.8	1643.3
3000	1648.9	1654.4	1660.0	1665.6	1671.1	1676.7	1682.2	1687.8	1693.3	1698.9
3100	1704.4	1710.0	1715.6	1721.1	1726.7	1732.2	1737.8	1743.3	1748.9	1754.4
3200	1760.0	1765.6	1771.1	1776.7	1782.2	1787.8	1793.3	1798.9	1804.4	1810.0
3300	1815.6	1821.1	1826.7	1832.2	1837.8	1843.3	1848.9	1854.4	1860.0	1865.6
3400	1871.1	1876.7	1882.2	1887.8	1893.3	1898.9	1904.4	1910.0	1915.6	1921.1
3500	1926.7	1932.2	1937.8	1943.3	1948.9	1954.4	1960.0	1965.6	1971.1	1976.7
3600	1982.2	1987.8	1993.3	1998.9	2004.4	2010.0	2015.6	2021.1	2026.7	2032.2
3700	2037.8	2043.3	2048.9	2054.4	2060.0	2065.6	2071.1	2076.7	2082.2	2087.8
3800	2093.3	2098.9	2104.4	2110.0	2115.6	2121.1	2126.7	2132.2	2137.8	2143.3
3900	2148.9	2154.4	2160.0	2165.6	2171.1	2176.7	2182.2	2187.8	2193.3	2198.9
4000	2204.4	2210.0	2215.6	2221.1	2226.7	2232.2	2237.8	2243.3	2248.9	2254.4
4100	2260.0	2265.6	2271.1	2276.7	2282.2	2287.8	2293.3	2298.9	2304.4	2310.0
4200	2315.6	2321.1	2326.7	2332.2	2337.8	2343.3	2348.9	2354.4	2360.0	2365.6
4300	2371.1	2376.7	2382.2	2387.8	2393.3	2398.9	2404.4	2410.0	2415.6	2421.1
4400	2426.7	2432.2	2437.8	2443.3	2448.9	2454.4	2460.0	2465.6	2471.1	2476.7
4500	2482.2	2487.8	2493.3	2498.9	2504.4	2510.0	2515.6	2521.1	2526.7	2532.2
4600	2537.8	2543.3	2548.9	2554.4	2560.0	2565.6	2571.1	2576.7	2582.2	2587.8
4700	2593.3	2598.9	2604.4	2610.0	2615.6	2621.1	2626.7	2632.2	2637.8	2643.3
4800	2648.9	2654.4	2660.0	2665.6	2671.1	2676.7	2682.2	2687.8	2693.3	2698.9
4900	2704.4	2710.0	2715.6	2721.1	2726.7	2732.2	2737.8	2743.3	2748.9	2754.4
5000	2760.0	2765.6	2771.1	2776.7	2782.2	2787.8	2793.3	2798.9	2804.4	2810.0
5100	2815.6	2821.1	2826.7	2832.2	2837.8	2843.3	2848.9	2854.4	2860.0	2865.6
5200	2871.1	2876.7	2882.2	2887.8	2893.3	2898.9	2904.4	2910.0	2915.6	2921.1
5300	2926.7	2932.2	2937.8	2943.3	2948.9	2954.4	2960.0	2965.6	2971.1	2976.7
5400	2982.2	2987.8	2993.3	2998.9	3004.4	3010.0	3015.6	3021.1	3026.7	3032.2
5500	3037.8	3043.3	3048.9	3054.4	3060.0	3065.6	3071.1	3076.7	3082.2	3087.8
5600	3093.3	3098.9	3104.4	3110.0	3115.6	3121.1	3126.7	3132.2	3137.8	3143.3
5700	3148.9	3154.4	3160.0	3165.6	3171.1	3176.7	3182.2	3187.8	3193.3	3198.9
5800	3204.4	3210.0	3215.6	3221.1	3226.7	3232.2	3237.8	3243.3	3248.9	3254.4
5900	3260.0	3265.6	3271.1	3276.7	3282.2	3287.8	3293.3	3298.9	3304.4	3310.0

Boiler Code Committee for determination of freezing and boiling points, respectively.

For temperatures up to 200° F. a water bath is satisfactory for comparison of thermometers. For temperatures from 200° to 600° F. an oil bath, using mineral or cylinder oil with high flash point, should be used. All baths should be stirred during tests to

maintain uniformity of temperature. For design of a satisfactory type of oil bath, see A.S.M.E. Power Test Code, Instruments and Apparatus, Part 3, p. 25, 1931. For tem-

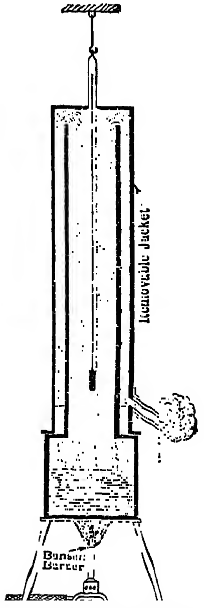


FIG. 2. Method of Determining Boiling Point

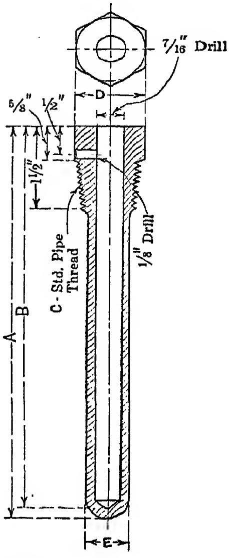


FIG. 3. Thermometer Well

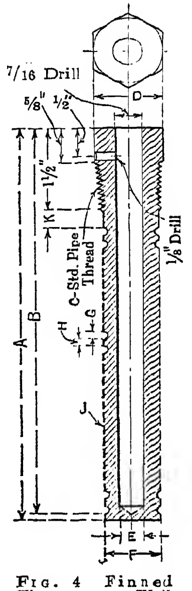


FIG. 4 Finned Thermometer Well

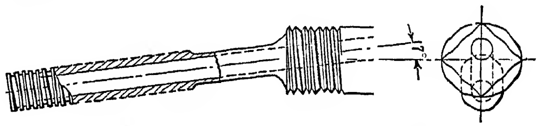


FIG. 5. Thermometer Well for Vertical Pipe

peratures between 300° and 900° F. molten salts, as Lavite, a mixture of barium and sodium chlorides, should be used. The type of bath should be similar to that used for oil.

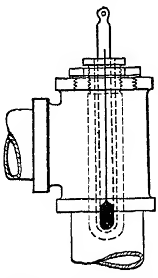


FIG. 6. Method of Inserting Well in Pipe

Time Lag.—A thermometer does not immediately indicate the temperature of the substance in which it is immersed. The time required to attain that temperature is known as the time lag and varies in different thermometers depending on a number of different conditions. In the measurement of temperatures of gases or liquids flowing in pipes the time lag may be of considerable importance, and the thermometer should be calibrated in respect to this feature. For a full discussion of time lag, see Thermometric Lag, U.S. Bureau of Standards Reprint No. 185.

THERMOMETER WELLS.—Whenever possible, a thermometer should be in direct contact with the substance whose temperature is being measured. When necessary to use a thermometer well, the well should not project any more than is necessary, in order to prevent heat transfer. All exposed parts of the well should be insulated, as should the wall of the pipe or other container for some distance on either side of the well, if such insulation will not affect the temperature of the substance being measured. The construction of the well should be such that it has the smallest possible metallic

connection, consistent with strength, with the wall of the pipe or other container, in order to reduce heat flow along the well. The well should have sufficient surface to enable it to absorb heat from the substance being measured as rapidly as it is lost from the exposed end.

The types of wells to be used to measure temperature of a fluid are as follows: Liquid and saturated vapor, plain well; gas and superheated vapor, finned well. Details of the construction of these wells are shown in Figs. 3 and 4, and in Table 5. Fig. 5 shows a finned well for a vertical pipe. Fig. 6 shows a method of inserting a well in pipes under 4 in. diam. For filling the wells mercury is advisable up to 500° F.; solder from 500 to 1000° F.; and tin from 600 to 1800° F.

Plain carbon steel (S max. = 0.05%) is suitable for wells for temperatures up to 750° F. For temperatures over 750° F. a heat-resisting steel should be used. The A.S.M.E. Boiler Code Committee gives (1931) the analyses of several steels that, with a proper factor of safety, may be used. Among these analyses are: 1. C, 0.49; Mn, 0.50; Si, 0.26; Cr, 1.62; Al, 1.00. 2. C, 0.50; Mn, 0.74; Si, 0.26; Cr, 2.32; V, 0.22. 3. C, 0.49; Mn, 0.24; Si, 0.26; Cr, 1.55; W, 2.40; V, 0.21. 4. C, 0.26; Mn, 0.65; Si, 3.03; Ni, 24.13; Cr, 18.23.

Table 5.—Dimensions of Plain and Finned Thermometer Wells

See Figs. 3 and 4. All dimensions in inches.

	Plain and Finned Wells					Finned Wells Only				
Pipe Size, in.	*A*	*B*	*C**	*D*	*E*	*F*	*G*	*H*	*J*†	*K*
4–7	4	3 7/8	1/2	1	0.525	11/16	3/32	1/16	15	5/32
8 and up	7	6 7/8	3/4	1 1/8	5/8	7/8	5/32	3/32	21	1/4

* Standard pipe-size thread. † Number of fins and grooves.

Bourdon Tube Thermometers

Bourdon tube thermometers are: 1. Liquid-filled, the liquid completely filling the bulb, capillary and spring. 2. Vapor pressure, the liquid in the bulb having a free surface. 3. Gas filled, the gas completely filling the bulb, capillary and spring. Bourdon tube thermometers may be used as indicating and recording instruments and also as distant-reading instruments.

LIQUID-FILLED THERMOMETERS are: 1. Non-compensated; 2. Compensated for change in temperature of spring only; 3. Compensated for change in temperature of spring and capillary. The liquids used usually are mercury for temperature ranges of −38° to 1000° F., and alcohol for temperature ranges of −50° to 300° F. The maximum length of capillary without compensation is 25 ft.; 10 ft. is the recommended maximum length. This type of thermometer uses bulbs smaller than those in the gas-filled and vapor-pressure types, and its accuracy is unaffected by barometric pressure changes.

VAPOR-PRESSURE THERMOMETERS depend for their action on the pressure inside the thermometer as determined by the temperature of the free surface of the liquid. If the capillary and pressure spring contain only vapor, error due to variation of temperature in tube and spring is absent. If the capillary and spring are filled with liquid no error will result from temperature change around the spring and along the capillary if the bulb temperature is higher.

The following liquids are used in vapor-pressure thermometers:

Liquid	Working Range, deg. F.	Liquid	Working Range, deg. F.
Ethyl alcohol	200 to 400	Aniline	400 to 700
Benzine	200 to 500	Sulphur dioxide	20 to 250
Water	200 to 600	Ether	100 to 300
Toluene	250 to 500	Methyl ether	−20 to 200

A vapor-pressure thermometer is not subject to error due to variation in temperature of the capillary if installed under conditions for which specified. The capillary may be of any length up to 200 ft. As a recording instrument it has the disadvantage of using a chart with a non-uniform scale which cannot be integrated with an ordinary planimeter. Also, its accuracy is affected by the elevation of the bulb above or below the Bourdon spring and by changes in barometric pressure. The error may amount to as much as would obtain between sea level conditions and elevations of 10,000 ft. and may range from 0.8% to 2.6%, depending on range of the thermometer, the part of the range that is used and the liquid in the instrument. The greatest error occurs with temperatures near the lower end of the range of the instrument. With a vapor-pressure thermometer,

temperature of the bulb usually must be above or below temperature at the spring or along the capillary.

GAS-FILLED THERMOMETERS chiefly utilize nitrogen as the working gas. The temperature range is −60° to 1000° F. The length of capillary that may be used is unlimited, and installations with capillaries several hundred feet long are not uncommon. Gas-filled thermometers are subject to errors if the capillary is at a different temperature in service than that at which it was calibrated.

3. PYROMETRY

The following division of pyrometric methods is given by Burgess (High Temperature Measurements): 1. The gas pyrometer, which utilizes the measurement of change of pressure of a constant volume of gas. See Bourdon Tube Thermometers, p. 3–09. 2. Calorimetric pyrometer, based on the elevation of temperature of a known volume of water by a mass of metal, at the temperature to be ascertained, dropped into the water. It is useful for intermittent researches in industrial establishments. Platinum is used in the laboratory, and nickel in the industrial plant. This method is practically obsolete (1935). 3. Radiation pyrometer, depending on the heat radiated by warm bodies. Its indications are influenced by the variable emissive power of the various substances. It is convenient for the determination of very high temperatures, as those of the electric arc, the sun, and very hot furnaces. It is useful for measuring the temperature of moving objects, as billets, etc., particularly when an automatic record is desired. 4. Optical pyrometer, using either the photometric measurement of radiation of a given wave length of a definite portion of the visible spectrum, or the disappearance of a bright filament against an incandescent background. It also is influenced, but to a lesser extent than radiation pyrometers, by variable emissivity. It is principally used in industry to determine temperature of bodies difficult of access, *i.e.*, liquid flowing metal, metal in a furnace, moving billets, etc. The optical pyrometer cannot be made recording. It is the best instrument for measuring temperatures above 3000° F. 5. Electrical resistance pyrometer, which utilizes variations in the electrical resistance of platinum or other metals with

Table 6.—Type of Pyrometers for Various Industrial Operations

(Compiled by Brown Instrument Co., Philadelphia, 1932)

Service or Operation	Temp. Range, deg. F.	Thermocouple	Protecting Tube	Type of Instrument
Steel hardening				
Carbon steel.......	to 1800	(*a*) Ni-Cr, Ni-Al	Ni-Cr	Thermo-electric potentiometer
High speed........	to 2500	(*b*) Platinum	Sillimanite* and outer nickel tube	"
Blast furnaces				
Checkerwork.......	1000–1600	(*a*) Ni-Cr, Ni-Al	Ni-Cr	"
Glass plants				
Melting tanks......	2500–2800	(*b*) Platinum	Sillimanite with outer silica block	"
Ovens and lehrs....	800–1400	Iron, constantan	Calorized iron	"
Ceramic kilns........	2200–2800	(*b*) Platinum	Silimanite with outer silica block	"
Boilers				
Furnace walls......	2000–2500	....	....	Radiation, optical
Last pass and stack	300–800	Iron, constantan	....	Thermo-electric
Brass, copper and bronze melting...	1700–2300	(*a*) Ni-Cr, Ni-Al	Bare couple	"
Iron and steel melting	2700–3200	....	....	Radiation, optical
Cement plants				
Kilns............	2800–3000	....	....	"
Kiln stack.........	1200–1600	(*a*) Ni-Cr, Ni-Al	Ni-Cr	Thermo-electric potentiometer
Oil refinery				
Furnaces..........	up to 1800	(*a*) Ni-Cr, Ni-Al	Ni-Cr	"
Hot oil............	750–1300	Iron, constantan	Steel	"
Oil vapor..........	150–600	Iron, constantan	Bronze, steel	"
Coke oven and producer gas plants..	1200–1600	(*a*) Ni-Cr, Ni-Al	Ni-Cr or high Cr-Fe alloy	"

(*a*) Positive wire: nickel, 90%; chromium, 10%. Negative wire: nickel, 98%; aluminum, 2%
(*b*) Positive wire: pure platinum. Negative wire: platinum, 90%; rhodium, 10%.
* Mullite.

temperature. It is correct up to 1800° F. but requires fragile and delicate parts. 6. Thermo-electric pyrometer, utilizing the electro-motive forces developed by the difference in temperature of two similar thermo-electric junctions opposed to one another. It is extensively used in industrial works for the measurement of temperatures up to 2700° F. 7. Contraction pyrometer (Wedgwood) which utilizes the permanent contraction taken by clayey materials when subject to more or less high temperatures. It is used only in a few pottery works. 8. Fusible cones, utilizing the unequal fusibility of earthenware cones of varied composition. These give only discontinuous indications. They are in general use in pottery works and similar industries.

The type of instrument to be used in any particular case depends on the temperature and on the conditions of operation. With thermo-electric pyrometers, the couple and its protecting tube must be chosen with regard to the temperature, the character of the gases present in the furnace and the time lag that is permissible. Table 6 will serve as a guide for the selection of the pyrometer suitable for different industrial conditions.

Pyrometers

RADIATION PYROMETER.—The intensity of the radiant energy emitted by a body is an indication of the temperature of that body. The intensity of radiation depends both upon the temperature of the body and upon the material composing it. If two similar hot bodies are at the same temperature, and the first is found to radiate energy at twice the rate of the second, the first is said to have twice the emissive power of the second. A material with the highest possible emissivity is known as a " black body." Black-body conditions are obtained by heating uniformly a hollow enclosure and measuring the radiation from a small opening in the walls. The numerical value 1 is usually applied to the emissivity of a black body and all other materials thus have an emissivity of less than 1. (See also Radiation, p. 3–31.)

In the radiation pyrometer the emissions from a radiant body are focused to fall upon the hot junction of a small thermocouple in the pyrometer, and the temperature to which this junction rises is approximately proportional to the rate at which the energy falls upon it. According to the Stefan-Boltzman law, this temperature is proportional to the 4th power of the absolute temperature of the source of radiation. The electro-motive force generated by this temperature rise is measured by a galvanometer or potentiometer, as in the thermo-electric pyrometer, which is calibrated to determine the relation between the electro-motive force so developed and the temperature of the radiant body. The energy waves radiating from the body are focused by a concave mirror upon the thermo-electric couple. Dust accumulations upon this mirror may reduce its reflection coefficient to such an extent that errors of 100° to 200° C. may be developed, and precautions should be taken to prevent dust entering the instrument. The readings generally are independent of the distance of the instrument from the source of radiation, providing that the image must more than cover the disc or black spot in the instrument, although this condition is not always realized. The radiation pyrometer tends to give a reading lower than the true reading as the sighting distance is increased or the size of the radiant body is decreased. It is, therefore, desirable to locate the instrument as closely as possible to the body whose temperature is to be measured.

Radiation pyrometers are calibrated to read correctly when sighted on a black body.

Table 7.—True Temperature Corresponding to Apparent Temperature Measured by Radiation Pyrometers when Sighted upon Materials in the Open.

(U. S. Bureau of Standards, Technologic Paper 170)

Observed Temperature, deg. F.	True Temperature, deg. F.				
	Molten Iron	Molten Copper	Copper Oxide	Iron Oxide	Nickel Oxide
1110	....	2065	1330	1165	1310
1200	....	2210	1425	....	1390
1290	....	2355	1525	1355	1470
1380	....	....	1635	....	1555
1470	2190	....	1735	1545	1645
1560	2320	....	1830	....	1725
1650	2445	....	1940	1735	1805
1740	2570	....	2040	....	1885
1830	2685	....	2140	1920	1965
1920	2820	....	....	....	2050
2010	2930	....	....	2110	2130
2100	3055	....	....	....	2210
2190	3180	....	....	2300	2290

Most furnaces approximate black body conditions. Materials in the furnace, and at the same temperature, cannot be distinguished from the surroundings, and may be considered black bodies. If the coefficient of actual emissivity of the material is 0.45, its reflection coefficient will be 0.55, and the total emissivity of the body will be 1.0, or that of a black body. That is 55% of the energy radiated from the body will be reflected from the surrounding walls, which are at the same temperature. If, however, the material be removed from the furnace, it then will emit but 45% of the radiant energy of a black body at the same temperature.

Materials in the open, whose temperature is to be measured, therefore, should have a correction applied to the observed temperature to convert it to true temperatures. Table 7 shows true temperatures corresponding to apparent temperatures observed with a radiation pyrometer and sighted on materials in the open.

For details of construction of radiation pyrometers, consult catalogs of Thwing Instrument Co., Philadelphia; Brown Instrument Co., Philadelphia; Taylor Instrument Co., Rochester, N. Y.; Leeds & Northrup Co., Philadelphia.

Optical Pyrometer.—The optical pyrometer is similar to the radiation pyrometer in that it measures the energy radiated from incandescent bodies, but measures the light energy instead of the heat energy. A source of light of constant brightness in the pyrometer is compared with the luminosity of the body whose temperature is to be measured. In one type, the pyrometer is focused on the source of energy, and then wedges of absorbing glass are adjusted until the lights from the two sources appear of the same intensity upon the field of the instrument. A color screen causes the light to be monochromatic (usually red). A scale which reads directly in degrees of temperature, attached to the absorbing wedges, is calibrated to show the temperature corresponding to different positions of the wedges. Another type of optical pyrometer compares the brightness of the filament of an electric lamp in the pyrometer with the brightness of the body whose temperature is being measured. The current through the filament is adjusted, until the filament disappears when viewed against the radiant body. The instrument is so calibrated that the current required for the lamp is a direct reading of the temperature of the body in question.

Optical pyrometers are calibrated to read correctly when sighted on a black body. Objects in the furnace which are practically indistinguishable when heated, approximate black body conditions, and the temperature measurement is not seriously in error. Such error as exists will give a reading higher than the true reading, if the furnace walls are brighter than the material and lower than the true reading if the material is brighter than the walls. The pyrometer should be sighted through a small peep hole when approximate temperature uniformity is reached, or through a deep wedge-shaped cavity or hole in the material itself.

The optical pyrometer sighted on glowing material in the open reads too low. Corrections are necessary for materials whose temperature is measured under such conditions. Table 8 gives the corrections for various industrial materials sighted on in the open, when measured by optical pyrometers. The optical pyrometer will not give satisfactory readings when sighted through flames or smoke. The presence of flames will increase

Table 8.—True Temperature Corresponding to Observed Temperature Measured by Optical Pyrometers Using Red Light when Sighted on Materials in the Open

Observed Temperature, deg. F.	True Temperature, deg. F.						
	Molten Copper	Molten Iron	Solid Iron Oxide	Solid Nickel Oxide	Nichrome or Chromel	Molten Slag	Bright Platinum
1290	....	....	1290	1295	1295	....	1380
1470	....	....	1475	1475	1480	....	1580
1650	....	....	1655	1660	1660	....	1780
1740	1990	....	1745	1750	1755	....	1885
1830	2100	....	1840	1845	1850	....	1980
1920	2215	....	1930	1935	1945	....	2090
2010	2330	2160	2020	2030	2040	....	2195
2100	2445	2260	2115	2125	2140	....	2300
2190	2560	2365	2210	2220	2235	....	2405
2280	2680	2470	....	2310	....	....	2505
2370	2800	2570	....	2410	....	....	2615
2550	....	2775	....	....	....	2650	2830
2730	....	2985	....	....	....	2850	3045
2910	....	3195	....	....	....	3040	....
3090	....	3410	....	....	....	3235	....
3180	....	3515	....	....	....	3325	....

the temperature readings, while smoke will cause the readings to be several hundred degrees low.

For details of the construction of optical pyrometers consult catalogs of Taylor Instrument Co., Rochester, N. Y.; Scientific Materials Co., Pittsburgh; Leeds & Northrup Co., Philadelphia; Shore Instrument Co., New York.

RESISTANCE PYROMETERS are made of platinum for temperatures up to about 1800° F., and of nickel for temperatures up to about 600° F. The variation in the electrical resistance of these metals with changes of temperature may be determined with a Wheatstone bridge or a galvanometer. These pyrometers are largely used for the measurement of low temperatures, and to some extent for high temperatures. They are of value in laboratory work for precision measurements. An important use is the accurate measurement of steam temperature. Resistance pyrometers are made for industrial use, and are applied to high temperatures where higher accuracy is desired than is possible with a base-metal thermocouple if the temperatures are not high enough to give the required sensitivity with a noble metal thermocouple. The thermocouple usually is preferred, due to its greater simplicity and lower first cost.

THERMO-ELECTRIC PYROMETERS.—The thermo-electric pyrometer consists essentially of a thermocouple of two different metals or alloys. The wires composing it are fused together at one end to form the measuring (hot) junction which is exposed to the temperature to be measured. The other ends, called the reference (cold) junction, are maintained at a constant temperature, as room temperature. The electro-motive force, induced by the difference in temperature between the two ends, is proportional to the temperature difference and may be measured by an indicating instrument, as the millivoltmeter or a potentiometer. These, together with lead wires connecting the reference junctions to the indicating instrument, complete the pyrometer.

The original thermo-electric pyrometer, as developed by Le Chatelier, employed a thermocouple consisting of one wire of platinum and one of an alloy of platinum 90%, and rhodium 10%. This couple may be used with temperatures up to about 2700° F. with an accuracy of about 5° F. Other combinations known as base metal couples are available and are of sufficient accuracy for industrial use. Up to 675° F. a copper-constantan couple may be used with extreme precision, or to within ±10° F. up to 900° F. Up to 1600° F. iron-constantan or nichrome-constantan couples may be used. Below 2000° F. alloys of chromium and nickel and of aluminum and nickel, known as chromel-alumel and nichrome-alumel, may be used in continuous service. For temperatures above 2700° F. the optical or radiation pyrometer should be used.

Thermo-electric pyrometers are calibrated for a certain reference junction temperature. Any variation in this temperature will vary the electro-motive forces in the couple, and therefore give a false indication of furnace temperature. If a thermo-electric couple is used at a different temperature than that for which it was calibrated, the true temperature of the measuring junction is $T + tK$, where T = observed temperature, t = difference between the temperature at which the reference junction was calibrated and the temperature at which it was used; K = a constant whose value is given in Table 9. These corrections may be read directly by setting the pointer of the galvanometer to read the reference junction temperature when the couple is disconnected. In the Bristol thermo-electric pyrometer a bi-metallic spring connected to one of the control springs of the moving coil performs this adjustment automatically. The reference junction must be located at the indicating instrument, or connected to it by compensating lead wires which will minimize reference junction errors. In Leeds & Northrup pyrometers, automatic compensation of reference junction is obtained by use of a nickel coil. For description see Pyrometry Symposium of A.I.M. & M.E., 1921, p. 206.

Compensating lead wires usually are of the same materials as those composing the couple. This, in effect, transfers the reference junction to the terminals of the indicating

Table 9.—Reference Junction Correction Factors, Deg. F.

(Based on calibration with t = 32° F.)

Engelhard, "Le Chatelier"		Johnson-Matthey, "Le Chatelier"		Copper-constantan		Iron-constantan		Chromel-alumel	
Temp.	K	Temp.	K	Temp.	K	Temp.	K	Temp.	K
510–840	0.65	480–750	0.60	0–120	1.00	0–210	1.00	0–1470	1.00
840–1200	.60	750–1020	.55	120–175	0.95	210–1110	.95	1470–2010	1.05
1200–1830	.55	1020–1650	.50	175–230	.90	1110–1830	.85	....	......
1830–2640	.50	1650–2640	.45	230–300	.85	....	.....	....	......
....	.....	....	.....	300–390	.80	....	.....	....	......
....	.....	....	.....	390–520	.75	....	.....	....	......
....	.....	....	.....	520–660	.70	....	.....	....	......

instrument or to a point where the temperature is fairly constant. If the reference junctions be buried 10 ft. below ground, temperature will be constant to within 5° F. throughout the year. For platinum-rhodium couples, compensating leads of these materials are expensive. The Bristol Co. has developed lead wires consisting of copper and of a copper-nickel alloy for this type of couple, the copper being connected to the platinum-rhodium wire of the couple and the copper-nickel wire to the platinum wire. The reference junction is located at the indicator end of the compensating leads, which, while they do not compensate individually, taken together will compensate to within 9° F. for a variation of 360° F. in the lead couple-wire junction.

PROTECTION OF THERMOCOUPLES.—Thermocouples, especially the platinum-rhodium type, are sensitive to rough handling and should be protected if there is danger of their coming in contact with materials in the furnace. While the use of a protecting tube introduces a certain time lag in the reading of the instrument, this is unimportant unless instantaneous values of temperature are required. The following properties are desirable in thermocouple protecting tubes: Low porosity to gases; low volatility; ability to withstand high temperature and sudden changes in temperature; ability to withstand mechanical shocks and stresses; high rigidity or viscosity; ability to resist corrosion due to molten metals or furnace gases. If rapid changes of temperature are to be measured, high thermal conductivity is desirable, although low conductivity usually is required so that the flow of heat along the tube shall be as small as possible. Table 6 shows the type of thermocouple and protecting tube for different services.

For the properties of the various metals used in protecting tubes, see Technologic Paper No. 170, U. S. Bureau of Standards, p. 91.

TEMPERATURE RECORDERS.—A record of temperature in furnaces, baths, etc., frequently is of great importance in industry. This can be obtained with Bourdon tube thermometers, and with pyrometers in which change of temperature is indicated by a change in potential of an electric current, as thermo-electric, resistance and radiation pyrometers. In these instruments a pen or other marking device that traces a record of fluctuations in temperature on a chart moved at a uniform rate by any suitable means, is substituted for the indicating pointer. The type of chart depends on the type of instrument used, the class of service, the temperature range, and other considerations. Catalogs of makers should be consulted, and their advice sought before making a selection.

Temperature recorders are widely used for central control of furnaces, as in heat treating, where accurate maintenance of predetermined temperatures is essential. Some types of instrument will record the temperature of several sources of heat at one time. They also may be arranged with a system of colored signal lamps to call to the attention of the attendant any deviation of temperature beyond the prescribed limits.

TEMPERATURE CONTROL is closely allied with temperature recording. It is accomplished by a modification of temperature recorders, whereby fluctuation of temperature causes the mechanism to open or close an electric circuit which operates a relay. This relay, in turn, controls a heavier current that operates the necessary valves, switches, dampers, etc., to increase or diminish the heat supplied to the apparatus whose temperature is being controlled.

Other types of temperature controller utilize vapor pressure or compressed air for actuating valves, switches, dampers, etc., for varying the supply of heat. The Bristol Co. builds an instrument in which air pressure is varied upon two sides of a diaphragm through suitable mechanism attached to the indicating arm. This variation in pressure operates a pilot valve, which in turn controls air pressure to actuate the heat supply valve or switch. In the type of temperature controller built by the Fulton-Sylphon Co. the vapor pressure system is used. The capillary connected to the heat-sensitive bulb is attached at its other end to a sensitive bellows which expands or contracts with changes in vapor pressure. This expansion and contraction is utilized to operate pilot valves or other means of controlling the heat supply. For light work the bellows acts directly on the control valve.

In certain types of controllers, a temperature record is also made. Other types can be arranged to control temperature at predetermined times; for instance, to supply heat to lead baths a couple of hours before starting time in the morning, so that they will be at the proper temperature when the men arrive. They also are used to shut off the source of heat at a predetermined time, as when a material is to be subjected to a given temperature for a given length of time.

Temperature controllers are quite sensitive, and will control temperatures to within 1/2 deg. F. in a 250-deg. range, or to within 2° F. in a 1000-deg. range.

The applications and arrangement of these instruments are so numerous and varied that extended treatment of the subject is not possible here. Makers should be consulted in regard to the selection of the type of instrument and its application to a given service.

MEASUREMENT OF TEMPERATURE BY PHOTO-ELECTRIC TUBE.—(*Ind. & Engg. Chem.*, Dec., 1931). The photo-electric tube is sensitive to light from visible spectrum and the near infra red-rays. As the temperature of a body changes, its brightness also changes. Changes in light falling on the tube change the flow of current in it, and hence it can be used for both temperature measurement and temperature control. The current developed by the tube is amplified to operate control relays in the latter case. The tube is focussed on the image of the furnace wall or other surface to be measured, in the plane of a diaphragm in front of the tube which allows only the center portion of image to pass through. See Fig. 7. It is important that the tube always observes the same area of hot body. The apparatus is calibrated by determining the temperature of the furnace or body by means of pyrometer or otherwise, adjusting the variable diaphragm in the tube housing until the correct temperature is indicated on the scale of the tube. But one calibration is necessary, since for monochromatic radiation the logarithm of the energy radiated is proportional to the reciprocal of the temperature. This method of measuring temperature can be applied to high temperatures and to atmospheres deleterious to thermocouples or pyrometers.

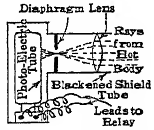


Fig. 7. Temperature Measurement b Photo-electric Tube

MEASUREMENT OF SURFACE TEMPERATURES (M. W. Boyer, J. Buss, F. W. Adams and R. H. Kean, *Ind. & Engg. Chem.*, July, Aug., 1926). A series of experiments showed the error in the measurement of surface temperatures by several different methods. The experiments were made on a steam chest cover ranging from 221° F. to 305° F. The methods used were: *A*. Thermometer under pad; *B*. Thermocouple under pad; *C*. Thermocouple of separate wires making contact through the metal surface itself; *D*. Thermocouple soldered to surface; *E*. Thermocouple embedded in surface; *F*. Compensated thermocouple held against surface. The following results were obtained:

Method	*A*	*B*	*C*	*D*		
Max. error, deg., C.	−17.8	−8.3	−8.3	−3.6		1.1
Min. error, deg., C.	−11.4	−5.8	−1.9	−2.7	−1.1	
Calculated error, %*	18	9	7	4	1	

* Based on heat loss by radiation and convection, as calculated, from body at 270° F. to room at 68° F.

The compensated couple is recommended for measurement of surface temperatures of either stationary or moving bodies. It consists of a means of applying sufficient heat to the exposed side of the thermocouple to prevent any heat loss through it. A bare couple had such large heat loss from the couple that it never attained the surface temperature. Fig. 8 diagrammatically represents the method of compensating. If the two couples show a difference of temperature, heat is flowing from the measuring to the regulating couple. In such event, heat is supplied by the heating coil until both couples show the same temperature. The temperature of either couple then is the temperature of the surface.

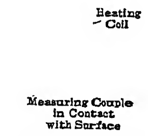


Fig. 8. Compensated Thermocouple

Pyrometric Cones

A pyrometric cone is a three-sided pyramid of mineral mixtures, the mixtures being so graded that they are affected by heat at different temperatures. A standard pyrometric cone, depending on its position in the series is either 3 in. high by $^5/_8$ in. wide at the base, or $^{25}/_{32}$ in. high by $^3/_8$ in. wide at the base. When heated under standard conditions, it bends at a definite known temperature. Pyrometric cones are used principally in clay, pottery and allied industries to determine the heat conditions of kilns, furnaces, etc. The softening or fusion is not altogether a matter of temperature, the element of time and character of surrounding gases also having an effect. Table 10 shows fusion points of pyrometric cones according to the original scale and according to the 1926 scale. The figures for the latter scale were obtained at the U. S. Bureau of Standards by Fairchild and Peters (*Jour.* Am. Ceramic Soc., vol. ix, p. 701, 1926).

4. TEMPERATURE DETERMINATION BY COLOR

COLOR-TEMPERATURE SCALES.—The statement that the experienced eye unaided can judge the temperature of a body by its color should be accepted with caution. Skilled observers may vary 100° F. or more in their estimation of relatively low temperatures by color, and beyond 2200° F. it is practically impossible to make estimations with any certainty whatever. (Bull. No. 2, U. S. Bureau of Standards, 1905.) This

Table 10.—Fusion Points of Pyrometric Cones

Cone No.	End Point, Original Scale		End Point, 1926 Scale. Heated in air at rate of 150° C. (270° F.) per hr.		Cone No.	End Point, Original Scale		End Point, 1926 Scale. Heated in air at rate of 150° C. (270° F.) per hr.	
	Deg. C.	Deg. F.	Deg. C.	Deg. F.		Deg. C.	Deg. F.	Deg. C.	Deg. F.
05	1050	1922	1040	1904	18	1490	2714	1490	2714
04	1070	1958	1060	1940	19	1510	2750	1520	2768
03	1090	1994	1115	2039	20	1530	2786	1530	2786
02	1110	2030	1125	2057	23	1590	2894	1580	2876
01	1130	2066	1145	2093	26	1650	3002	1595	2903
1	1150	2102	1160	2120	27	1670	3038	1605	2921
2	1170	2138	1165	2129	28	1690	3074	1615	2939
3	1190	2174	1170	2138	29	1710	3110	1640	2984
4	1210	2210	1190	2174	30	1730	3146	1650	3002
5	1230	2246	1205	2201	31	1750	3182	1680	3056
6	1250	2282	1230	2246	32	1770	3218	1700	3092
7	1270	2318	1250	2282	33	1790	3254	1745	3173
8	1290	2354	1260	2300	34	1810	3290	1760	3200
9	1310	2390	1285	2345	35	1830	3326	1785	3245
10	1330	2426	1305	2381	36	1850	3362	1810	3290
11	1350	2462	1325	2417	37	1870	3398	1820	3308
12	1370	2498	1335	2435	38	1890	3434	1835	3335
13	1390	2534	1350	2462	39	1910	3470	1865*	3389
14	1410	2570	1400	2552	40	1930	3506	1885*	3425
15	1430	2606	1435	2615	41	1950	3542	1970*	3578
16	1450	2642	1465	2669	42	1970	3578	2015*	3659
17	1470	2678	1475	2687	....	....	....	....	....

* Heated at the rate 600° C. (1080° F.) per hr.

statement is borne out by Table 11, which presents three widely-used color-temperature scales, formulated by H. M. Howe, F. W. Taylor and The Halcomb Steel Co., respectively. It will be observed that there is a variation of 360° F. in the temperature ascribed to cherry red, and similar discrepancies throughout the scales. Temperature determinations by color are unreliable and should be used only for the crudest class of work.

Table 11.—Color-Temperature Scales

	Howe		Taylor		Halcomb	
	Deg. C.	Deg. F.	Deg. C.	Deg. F.	Deg. C.	Deg. F.
Red, visible in dark	470	878	....	....	400	752
Red, visible in twilight	....	....	....	....	474	885
Red, visible in daylight	475	887	....	....	525	975
Red, visible in sunlight	....	....	....	....	581	1077
Dull red	550–625	1022–1157	....	....	....	....
Dark red, blood red	....	....	556	1050	700	1292
Dark cherry red	....	....	635	1175	800	1472
Medium cherry red	....	....	670	1250	....	....
Full cherry red	700	1292	746	1375	900	1652
Light red, bright cherry red	850	1562	843*	1550*	1000	1832
Orange, free scaling heat	....	....	899	1650	....	....
Orange-red	....	....	....	....	1100	2012
Light orange	....	....	941	1725	....	....
Yellow	950–1000	1742–1832	996	1825	....	....
Orange-yellow	....	....	....	....	1200	2192
Light yellow	1050	1922	1079	1975	....	....
Yellow-white	....	....	....	....	1300	2372
White	1150	2102	1205	2200	....	....
White welding heat	....	....	....	....	1500	2732
Dazzling white (bluish white)	....	....	....	....	1600	2912

* Heat at which scale forms and adheres on iron and steel.

Temper Colors and Temperatures.—The Halcomb Steel Co. (1908) gives the following heats and temper colors of steel:

Deg. C.	Deg. F.	Colors	Deg. C.	Deg. F.	Colors	Deg. C.	Deg. F.	Colors
221.1	430	Very pale yellow.	254.4	490	Yellow-brown.	282.2	540	Full purple.
226.7	440	Light yellow.	260.0	500	Brown-yellow.	287.8	550	Dark purple.
232.2	450	Pale straw-yellow.	265.6	510	Spotted red-brown.	293.3	560	Full blue.
237.8	460	Straw-yellow.	271.1	520	Brown-purple.	298.9	570	Dark blue.
243.3	470	Deep straw-yellow.	276.7	530	Light purple.	315.6	600	Very dark blue.
248.9	480	Dark yellow.						

5. ABSOLUTE TEMPERATURE—ABSOLUTE ZERO

The absolute zero of a gas is a theoretical consequence of the law of expansion by heat, assuming that it is possible to continue the cooling of a perfect gas until its volume is diminished to nothing.

The volume of a perfect gas increases $^1/_{273.1}$ of its volume at 0° C. for every increase of temperature of 1° C., and decreases $^1/_{273.1}$ of its volume at 0° C. for every decrease of temperature of 1° C. At −273.1° C. the volume would then be reduced to nothing. This point, −273.1° C. = − 459.6° F., or 491.6° F. below the temperature of melting ice, is called the absolute zero, and absolute temperatures are measured on either the Centigrade or the Fahrenheit scale, from this zero. The freezing-point, 32° F., corresponds to 491.6° F., absolute. If p_0 = pressure, and v_0 = volume of a perfect gas at 32° F. = 491.6° absolute, $= T_0$, and p = pressure and v = volume of the same weight of gas at any other absolute temperature T, then

$$\frac{pv}{p_0 v_0} = \frac{T}{T_0} = \frac{t + 459.6}{491.6}; \quad \frac{pv}{T} = \frac{\quad}{T_0} \quad R.$$

. cubic foot of dry air at 32° F. at the sea level (barometer 29.921 in. of mercury) weighs 0.080728 lb. The volume of one pound is 1/0.080728 = 12.387 cu. ft. The pressure is 2116.3 lb. per sq. ft.

$$R = \frac{p_0 v_0}{T_0} = \frac{2116.3 \times 12.387}{491.6} = \frac{26{,}214}{491.6} = 53.32.$$

QUANTITATIVE MEASUREMENT OF HEAT

; **OF HEAT.**—Several units for measuring the quantity of heat in a body are in use. The relations between them are given in Table 1.

The British Thermal Unit (B.t.u.) generally used in engineering work in the United States and Great Britain is $^1/_{180}$ of the heat required to raise the temperature of 1 lb. of water from 32° F. to 212° F. It formerly was defined as the quantity of heat required to raise 1 lb. of water from 62° to 63° F., but the former definition is now generally accepted.

The Kilogram-calorie or large calorie (kg.-cal.) is the heat required to raise the temperature of 1 kilogram of water from 14.5° to 15.5° C., or $^1/_{100}$ of the heat required to raise 1 kilogram of water from 0° C. to 100° C.

The Gram-calorie, small calorie, or 15° calorie (gm.-cal.), generally used in scientific work, is the heat required to raise 1 gram of water from 14.5° to 15.5° C.

The Mean Calorie (Mean cal.) is $^1/_{100}$ of the heat required to raise the temperature of

Table 1.—Relation of the Various Units of Heat

	B.t.u.	Kg.-cal.	Gram-cal.	Mean cal.	Ostwald cal.	Lb.-cal.
1 B.t.u. =	1.0	0.252	252.0	251.93	2.5193	0.55555
1 kilogram-calorie =	3.968	1.0	1000.	999.76	9.9976	2.2044
1 gram-calorie =	0.003968	0.001	1.0	0.999658	0.00996	0.002204
1 mean calorie =	0.003969	0.00100024	1.00024	1.0	0.0099991	0.00220499
1 Ost. calorie =	0.3969	1.00024	100.024	100.	1.0	0.220499
1 lb.-calorie =	1.8	0.4536	453.6	453.474	4.53474	1.0

Table 2.—Mechanical Equivalent of Heat

	Ft.-lb.	Kg.-meters	Joules		B.t.u. $\times 10^{4}$	Gm.-cal.	Watt-hr. $\times 10^4$	
			Int.	Abs.†			Int.	Abs.
1 Ft.-lb. =	....	0.13826	1.3563	1.3563	12.844	0.32367	3.7649	3.7662
1 Kg.-meter =	7.2330	....	9.80998	9.80665*	92.90	2.3410	27.232	27.241
1 Joule, Int. =	0.73781	0.10200	....	1.00034	94.76	0.2388	2.7778	2.7787
1 Joule, abs.† =	0.73756	0.10197	0.99966	....	94.73	0.23872	2.7769	2.7778
1 B.t.u.‡ =	778.57	107.64	1055.3	1055.6	....	252.0	2931.3	2932.3
1 Gm.-cal.‡ =	3.0896	0.42716	4.1876	4.1890	39.683	....	11.632	11.636
1 Watt-hr., Int. =	2656.1	367.220	3602.4	3601.2	34115.0	859.7	....	9996.70
1 Watt-hr., abs. =	2655.2	367.097	3601.2	3600.0*	34103.0	859.4	1000.32	....

* Exact value, by definition. † 1 absolute joule = 10 ergs. ‡ Mean value.

1 gram of water from 0° to 100° C., the latent heats of fusion and boiling not being considered.

The Ostwald Calorie (Ost. cal.), frequently used in electro-chemistry, is the heat required to raise the temperature of 1 gram of water from 0° to 100° C.

The Pound Calorie (lb.-cal.) is the heat required to raise the temperature of 1 lb. of water 1° C.

The Mechanical Equivalent of Heat is the number of foot-pounds of mechanical energy equivalent to one British thermal unit, heat and mechanical energy being mutually convertible. The U. S. Bureau of Standards has determined that 1 B.t.u. = 778.57 ft.-lb., which is 1/180 of the change in total heat along the saturated liquid water line from 32° to 212° F. Table 2 gives the mechanical equivalent of heat as expressed in various units.

SPECIFIC HEAT

THERMAL CAPACITY.—The thermal capacity of a body between two temperatures T_0 and T_1 is the quantity of heat required to raise the temperature from T_0 to T_1.

SPECIFIC HEAT.—The specific heat of any substance is the ratio of the heat required to raise the temperature of a unit weight of that substance 1 deg. and the quantity of heat required to raise the temperature of the same weight of water 1 deg., usually from 62° to 63° F.

Table 1.—Specific Heat of the Chemical Elements

(For gases see Table 2; for liquids see Table 3)

Element	Temp., deg. F.	Specific Heat	Element	Temp., deg. F.	Specific Heat	Element	Temp., deg. F.	Specific Heat
Aluminum.....	32	0.2089	Gold.......	32–212	0.0316	Rubidium......	32	0.0802
	932	.2739	Indium.....	32–212	.0570	Ruthenium.....	32–212	.0611
	61–579	.2250	Iodine......	48–208	.0541	Selenium.......	−306 to +64	.068
Antimony.....	59	.0489	Iridium.....	54–212	.0323			
	212	.0503	Iron*.......	99	.1092	Silicon.........	135	.1833
	392	.0520		1832–2192	.1989		450	.2029
Arsenic (gray)..	32–212	.0822		32–1112	.1396		32–212	.0559
" (black)	32–212	.0861		32–1472	.1597		212	.05663
Barium........	−121 to +68	.068		32–1832	.1557		63–945	.05987
			(cast)...	68–212	.1189	Sodium..	−301 to +68	.253
Bismuth.......	68–212	.0302	(wrought)	59–212	.1152			
Boron.........	32–212	.307	Lanthanum..	32–212	.0448	Sulphur.	−306 to +64	.137
Bromine (solid)	−108 to −4	.0843	Lead........	61–493	.0319			
			Lithium......	212	1.0407	" (rhombic)	32–129	.1728
" (fluid)	55–113	.107	Magnesium..	68–212	0.2492	" (liquid)	246–297	.235
Cadmium......	212	.0570	Manganese...	68–212	.1211	Tantalum......	−301 to +68	.033
Cæsium.......	32–79	.0482	Mercury.....	−121 to +68	.032			
Calcium.......	32–358	.170					2552	.043
Carbon........	52	.160	"	212	.03284	Tellurium..	59–212	.0483
	1789	.467	Molybdenum.	68–212	.0647	Thallium...	68–212	.0326
	3146	.50	Nickel.....	64–212	.109	Thorium...	32–212	.0276
Cerium........	32–212	.0448		212	.1128	Tin (cast)..	70–228	.0551
Chlorine (liquid)	32–75	.2262		932	.1299	" (fluid).	482	.05799
Chromium.....	32	.1039		1832	.1608	" (").	2012	.0758
	212	.1121	Osmium.....	66–208	.0311	Titanium..	32–212	.1125
	1112	.1872	Palladium....	32–2309	.0714	Tungsten..	32–212	.0336
Cobalt.	932	.1452	Phos. (red)...	32–124	.1829		1832	.0337
	1832	.204	(yellow)	55–97	.202	Uranium..	32–208	.028
Copper..	59–450	.0951	Platinum....	68–212	.0319	Vanadium.	32–212	.1153
	212	.0942	"	68–2372	.0359	Zinc......	68–212	.0931
	1652	.1259	Potassium....	−301 to +68	.170		212	.0951
Gallium.......	54–235	.080					572	.1040
Germanium....	32–212	.0737	Rhodium....	50–207	.0580	Zirconium.....	32–212	.0660

* Dr.-Ing. P. Oberhoffer, in *Zeit.* des Vereines Deutscher Ingenieure (*Eng. Digest*, Sept., 1908), describes some experiments on the specific heat of nearly pure iron. The following mean specific heats were obtained:

Temp., deg. F.	500	600	800	1000	1200	1300
Specific Heat	0.1228	0.1266	0.1324	0.1388	0.1462	0.1601
Temp., deg. F.	1500	1800	2100	2400	2700	
Specific Heat	0.1698	0.1682	0.1667	0.1662	0.1666	

The specific heat increases steadily between 500° and 1200° F. Then it increases rapidly to 1400° F., after which it remains nearly constant.

DETERMINATION OF SPECIFIC HEAT.—A hot body of known weight and temperature, immersed in a mass of liquid of known specific heat, weight and temperature, will raise the temperature of the liquid until both liquid and body are at the same temperature. If C_1 and C_2 = specific heat; w_1 and w_2 = weight; t_1 and t_2 = temperature, of the hot body and the liquid respectively; and if T = final temperature of both, then, $C_1 = C_2w_2(T - t_1) \div w_1(t_1 - T)$.

Another method consists in determining the amount of electrical energy required to raise the temperature of a unit weight of the substance 1 deg. in 1 min., converting the result into heat units on the basis of 1 watt = 0.05685 B.t.u. per min.

SPECIFIC HEATS OF VARIOUS SUBSTANCES.—The specific heats of the chemical elements and of the more commonly-used gases, liquids and solids of engineering are given in Tables 1 to 6. Except where otherwise noted, these tables are based upon the physical tables published by the Smithsonian Institution.

The specific heat of many substances varies with temperature and probably is greater when the substance is liquid than when solid. In a number of cases the specific heat at

Table 2.—Specific Heat of Gases and Vapors

Substance	Temperature Range, deg. F.	Specific Heat at Constant Pressure	Mean Ratio of	Substance	Temperature Range, deg. F.	Specific Heat at Constant Pressure	Mean Ratio of C_p/C_v*
Acetone, C_3H_6O	79–230	0.3468		Chloroform, $CHCl_3$	72–172	0.1489	1.150
Air	-22 to +50	.2377	1.4011	Ether, $C_4H_{10}O$	156–435	.4797	1.029
	68–824	.2366	2.333	Hydrochloric acid, HCl	55–212	.1940	1.389
	68–1472	.2430	.399	Hydrogen	70–212	3.4100	1.419
Alcohol, C_2H_5OH	226–428	.4534	.133	Hydrogen sulphide, H_2S	68–403	0.2451	1.324
" CH_3OH	214–433	.4580	.256	Methane, CH_4	64–406	.5929	1.316
Ammonia	73–212	.5202	.3172	Nitrogen	68–824	.2419	1.405
Argon	68–194	.1233	.667	Nitric oxide, NO	55–342	.2317	1.394
Benzene, C_6H_6	95–356	.3325	.403	Nitrous oxide, N_2O	61–405	.2262	1.3111
Blast-furnace gas	....	.2277		Oxygen	55–405	.2175	1.3977
Bromine	181–442	.0555		Sulphur dioxide, SO_2	61–396	.1544	1.256
Carbon dioxide	52–417	.2169	.3003	Water Vapor	32	.4655	1.274
Carbon monoxide	79–388	.2426	.395		212	.421	1.33
Carbon disulphide	187–374	.1596	205		356	.51	1.305
Chlorine	61–649	.1125	.336				

* C_p and C_v = specific heat at constant pressure and constant volume, respectively.

Table 3.—Specific Heat of Liquids

Substance	Temperature, deg. F.	Specific Heat	Substance	Temperature deg. F.	Specific Heat
Alcohol (ethyl)	104	0.648	Petroleum	69.8–136.4	0.511
" (methyl)	59–68	.601	Potassium hydrate,		
Aniline	59	.514	KOH + 30 H_2O	64.4	.876
Benzole, C_6H_6	104	.423	Sea water, sp. gr. 1.0043	63.5	.980
Calcium chloride, $CaCl_2$,			" " sp. gr. 1.0235	63.5	.938
sp. gr. 1.14	104	.787	" " sp. gr. 1.0463	63.5	.903
sp. gr. 1.26	104	.676	Sodium Hydrate,		
Copper sulphate,			NaOH + 50 H_2O	64.4	.942
$CuSO_4$ + 50 H_2O	53.6–59	.848	Sodium chloride,		
Ethyl ether	32	.529	NaCl + 10 H_2O	64.4	.791
Glycerine	59–122	.576	Sodium chloride,		
Naphthalene, $C_{10}H_8$	176–185	.396	NaCl + 200 H_2O	64.4	.978
Nitrobenzole	57.2	.350	Toluol, C_6H_8	149	.490
Oils: Castor	....	.434	Water: See Table 5.	....	....
Citron	42	.438	Zinc sulphate,		
Olive	44	.471	$ZnSO_4$ + 50 H_2O	68–125.6	.842
Turpentine	32	.411			

Table 4.—Specific Heat of Sodium Chloride Solutions

Temp., deg. F.	Parts NaCl by Weight in 100 Parts of Solution							
	2	4	6	8	10	12	16	20
32	0.966	0.944	0.923	0.904	0.885	0.869	0.840	0.814
60	.971	.949	.929	.909	.891	.874	.844	.817
100	.980	.958	.936	.916	.899	.881	.851	.823
140	.986	964	.942	.922	.903	.885	.855	.827

several temperatures is given. Where a range of temperature is specified, the mean specific heat for that range is given.

INSTANTANEOUS SPECIFIC HEAT.—For the instantaneous specific heat of carbon dioxide and other substances at constant pressure and constant volume, see p. 3–75.

Table 5.—Specific Heat of Water

(From Goodenough's Properties of Steam and Ammonia)

Temp., deg. F.	Specific Heat	Temp., deg. F.	Specific Heat	Temp., deg. F.	Specific Heat	Temp., deg. F.	Specific Heat
20	1.0210	130	0.9978	240	1.0104	350	1.044
30	1.0104	140	.9984	250	1.0125	360	1.048
40	1.0048	150	.9990	260	1.0148	370	1.053
50	1.0015	160	.9998	270	1.0173	380	1.057
60	0.9995	170	1.0007	280	1.020	390	1.062
70	.9982	180	1.0017	290	1.023	400	1.067
80	.9975	190	1.0028	300	1.026	450	1.095
90	.9971	200	1.0039	310	1.029	500	1.130
100	.9970	210	1.0052	320	1.033	550	1.200
110	.9971	220	1.0068	330	1.036	600	1.362
120	.9974	230	1.0085	340	1.040	650	1.793

Table 6.—Specific Heat of Various Solids

(For Specific Heat of Solid Chemical Elements, see Table 1)

Substance	Temperature, deg. F.	Specific Heat	Substance	Temperature, deg F.	Specific Heat
Alloys:			Hematite, Fe_2O_3	59–210	0.1645
Bell Metal	59–208	0.0858	Ice	0 to −108	.463
Brass (red)	32	.0899	India Rubber	?–212	.481
" (yellow)	32	.0883	Kaolin	68–208	.224
Cu 80, Sn 20	57–208	.0862	Lava	77–212	.197
Cu 88.7, Al 11.3	68–212	.1043	Limestone	59–212	.216
German silver	32–212	.0946	Magnetite	64–113	.156
Sb 37.1, Pb 62.9	50–208	.0388	Marble	32–212	.21
Bi 63.8, Sn 36.2	68–210	.0400	Mica	68	.10
Asbestos	68–208	.195	Paraffin	95–104	.622
Borax (fused)	51–208	.2382	" (fluid)	140–145	.712
Brick	32–212	.22	Porcelain	32–212	.22
Concrete	32–212	.156	Pyrites (copper)	59–210	.1291
Cork	32–212	.485	Quartz, SiO_2	54–212	.188
Corundum	42–208	.1976	" sand	68–208	.191
Dolomite	68–208	.222	Rock salt	55–113	.219
Earth	32–212	.44	Sandstone	32–212	.25
Galena, PbS	32–212	.0466	Talc	68–208	.2092
Glass (crown)	50–122	.161	Vulcanite	68–212	.3312
" (flint)	50–122	.117	Wood:		
Gneiss	63–210	.196	Fir	32–212	.65
Granite	54–212	.192	Oak	32–212	.57
Graphite	32–212	.201	Pine	32–212	.67

CHANGE OF STATE—LATENT HEAT

LATENT HEAT is the quantity of heat required to change the state, as solid or liquid, of a body without rise of temperature.

Latent Heat of Fusion is the quantity of heat required, at the fusion temperature, to change a body from the solid to the liquid state, without change of temperature. When the body changes from the liquid to the solid state this same amount of heat is rejected to the atmosphere or other surrounding bodies. See Tables 7 and 8 for latent heats of various materials.

Latent Heat of Vaporization is the quantity of heat required to change a liquid, at the boiling point, to a vapor under constant pressure, without change of temperature. When the substance is changed from the vapor to the liquid state, under the same conditions of temperature and pressure, this same quantity of heat will be rejected from the substance. See Table 8 for latent heats of vaporization of various substances.

TOTAL HEAT OF EVAPORATION (ENTHALPY) is the sum of the heat which disappears in evaporating one pound of a given substance at a given temperature (or latent

Table 1.—Melting and Boiling Points, Degrees F., of the Chemical Elements

(From Smithsonian Tables)

Element	Melting Point	Boiling Point	Element	Melting Point	Boiling Point	Element	Melting Point	Boiling Point
Aluminum	1217.66	3272.	Indium	311.	......	Rhodium	3542.	......
Antimony	1166.0	2624.	Iodine	236.3	>392.	Rubidium	100.4	1284.8
Argon	−306.4	−303.	Iridium †	4262.?	......	Ruthenium †	4442.	......
Arsenic	1562.	680.*	Iron	2786.	4442.	Samarium	2372.–2552.	......
Barium	1562.	......	Krypton	−272.2	−241.	Selenium	422.6–428.0	1274.
Beryllium	2336.	......	Lanthanum †	1490.?	......	Silicon	2588.	......
Bismuth	519.8	2606.	Lead	620.6±4.5	2777.	Silver	1760.9	3551.
Boron †	3992.–4532.?	......	Lithium	366.8	2552.	Sodium	207.5	1382.
Bromine	−18.86	142.	Magnesium	1203.8	2048.	Sulphur ‡	235. / 246.6 / 224.2	832.5
Cadmium	609.62	1432.4	Manganese	2246.0	3452.	Tantalum	5252.	......
Cæsium	78.8	1238.	Mercury	−37.98	674.6	Tellurium	845.6	2534.
Calcium	1490.	......	Molybdenum	4595.	6548.	Thallium	575.6	2336.
Carbon	>6332.*	6512.	Neodymium †	1544.?	......	Thorium	>3092.	......
Cerium	1184.	......	Neon	−423.?	−398.2	Tin	449.4±.4	4118.
Chlorine	−150.7	−28.5	Nickel	2645.	......	Titanium	3263.	......
Chromium	2939.	3992.	Niobium	3092.?	......	Tungsten	6152.	10,526.
Cobalt	2696.	......	Nitrogen	−347.8	−319.	Uranium	<3362.	......
Copper	1981.4±5.4	4190.	Osmium †	4892.?	......	Vanadium	3128.	......
Fluorine	−369.4	−304.6	Oxygen	−360.4	−296.9	Xenon	−220.	−164.4
Gallium	86.18	......	Palladium	2820.±9	......	Yttrium	2714.	......
Germanium	1756.4	......	Phosphorus	111.6	550.4	Zinc	786.9	1706.
Gold	1945.4	......	Platinum	3191.±9	7070.	Zirconium †	3092.?	......
Helium	<−455.8	−448.6	Potassium	144.1	1313.6			
Hydrogen	−434.2	−422.6	Praseodium	1724.	......			
			Radium	1292.	......			

* Sublimes. † Value uncertain. Temperatures above the melting point of platinum may be 100° F. in error. ‡ Melting point temperatures are for various forms of sulphur.

Table 2.—Melting Points of Alloys, Degrees F.

Alloys of Tin, Lead and Bismuth

	Percent									
Lead	32.0	25.8	25.0	43.0	33.3	10.7	50.0	35.8	20.0	70.9
Tin	15.5	19.8	15.0	14.0	33.3	23.1	33.0	52.1	60.0	9.1
Bismuth	52.5	54.4	60.0	43.0	33.3	66.2	17.0	12.1	20.0	20.0
Solidifies at	204.8	213.8	257.0	262.4	293.0	298.4	321.8	357.8	359.6	453.2

Low-melting-point Alloy

	Percent						
Cadmium	10.8	10.2	14.8	13.1	6.2	7.1	6.7
Tin	14.2	14.3	7.0	13.8	9.4	....	....
Lead	24.9	25.1	26.0	24.3	34.4	39.7	43.4
Bismuth	50.1	50.4	52.2	48.8	50.0	53.2	49.9
Solidifies at	149.9	153.5	155.3	155.3	169.7	193.1	203.0

Table 3.—Melting and Boiling Points, Degrees F., of Inorganic Compounds

Compound	Chemical Formula	Melting Point	Boiling Point
Aluminum oxide	Al_2O_3	3668.	....
Ammonia	NH_3	−103.	28.3
Ammonium sulphate		284.	....
Borax		1041.8	....
Calcium chloride	$CaCl_2$	1425.2	....
Carbon tetrachloride	CCl_4	−11.2	159.2
Hydrochloric acid	HCl	−168.3	−117.
Hydrofluoric acid	HFl	−134.14	−33.8
Mercurous chloride		842±	....
Mercuric chloride	$HgCl_2$	539.6	581.
Nitric acid	HNO_3	−43.6	186.8
Potassium chlorate	$KClO_3$	701.6	....
Sodium chloride	NaCl	1472.	....
" carbonate	Na_2CO_3	1565.6	....
" sulphate	Na_2SO_4	1623.2	....
Sulphur dioxide	SO_2	−104.8	14.
Sulphuric acid	H_2SO_4	40.7	640.4
Zinc chloride	$ZnCl_2$	689.	1310.
" sulphate		122.	....

heat of evaporation) and of the heat required to raise its temperature, before evaporation, from some fixed temperature up to the temperature of evaporation. The latter part of the total heat is called the sensible heat. Total heat of evaporation is known as Enthalpy.

For the total heat, latent heat, etc., of steam at different pressures, see table of the

Table 4.—Melting and Boiling Points of Organic Compounds

Compound	Chemical Formula	Melting Point, deg. F.	Boiling Point, deg. F.
Acetylene	C_2H_4	−113.8	−118.8
Acetic acid	$C_2H_4O_2$	62.0	244.4
Alcohol, ethyl	C_2H_6O	−173.2	172.4
" methyl	CH_4O	−142.6	150.8
Aniline		17.6	363.0
Beeswax		143.6	
Benzine	C_6H_6	41.8	176.3
Benzoic acid	$C_7H_6O_2$	249.8	480.2
Camphor		348.8	408.2
Carbolic acid	C_6H_6O	109.4	359.6
Carbon disulphide	CS_2	−166.	115.
" tetrachloride	CCl_4	−22.	170.
Chloroform	$CHCl_3$	−85.	142.1
Ether, ethyl	$C_4H_{10}O$	−180.4	94.3
Gasoline			158–194
Glycerine	$C_3H_8O_3$	68.	554.
Naphthalene	$C_6H_4 \cdot C_4H_4$	176.	424.4
Nitrobenzine	$C_6H_5O_2N$	41.	411.8
Olive oil		68±	572±
Oxalic acid		374.	
Paraffin wax (soft)		100–125	662–734
" " (hard)		125–133	734–806
Spermaceti		113±	
Sugar (cane)	$C_{12}H_{22}O_{11}$	320.	
Tallow (beef)		80–100	
" (mutton)		90–142	
Tartaric acid	$C_4H_6O_6$	338.	
Toluene	C_7H_8	−133.6	230.5

Table 5.—Boiling Points of Water at Various Barometric Pressures

(Condensed from Goodenough's Properties of Steam and Ammonia)

Pressure, in. of Mercury	Temperature, deg. F.	Pressure, in. of Mercury	Temperature, deg. F.	Pressure, in. of Mercury	Temperature, deg. F.	Pressure, in. of Mercury	Temperature, deg. F.	Pressure, in. of Mercury	Temperature, deg. F.
20.0	192.37	22.8	198.61	25.6	204.24	28.2	209.03	29.6	211.45
20.2	192.84	23.0	199.03	25.8	204.62	28.3	209.20	29.7	211.62
20.4	193.31	23.2	199.45	26.0	205.00	28.4	209.38	29.8	211.79
20.6	193.77	23.4	199.86	26.2	205.38	28.5	209.56	29.9	211.96
20.8	194.23	23.6	200.27	26.4	205.76	28.6	209.73	30.0	212.13
21.0	194.68	23.8	200.69	26.6	206.13	28.7	209.91	30.1	212.30
21.2	195.13	24.0	201.09	26.8	206.50	28.8	210.08	30.2	212.47
21.4	195.58	24.2	201.49	27.0	206.87	28.9	210.25	30.3	212.63
21.6	196.02	24.4	201.89	27.2	207.23	29.0	210.43	30.4	212.80
21.8	196.46	24.6	202.29	27.4	207.60	29.1	210.60	30.5	212.97
22.0	196.90	24.8	202.69	27.6	270.96	29.2	210.77	30.6	213.13
22.2	197.33	25.0	203.08	27.8	208.32	29.3	210.94	30.7	213.30
22.4	197.76	25.2	203.47	28.0	208.67	29.4	211.11	30.8	213.46
22.6	198.19	25.4	203.86	28.1	208.85	29.5	211.28	30.9	213.63

Table 6.—Increase in Temperature of Boiling by Salts in Solution

Salt	Rise in Boiling Point, deg. C.									
	1	3	5	7	10	20	40	60	80	100
	Number of Parts of Salt Added to 100 Parts of Water									
$CaCl_2$	6.0	16.5	25.0	32.0	41.5	69.0	137.5	222.0	314.0	......
KOH	4.7	13.6	20.5	26.4	34.5	57.5	92.5	121.7	152.6	185.0
KCl	9.2	23.4	36.2	48.4	......	......	......	......	......	......
$MgSO_4+7H_2O$	41.5	138.0	262.0	......	......	......	......	......	......	......
NaOH	4.3	11.3	17.0	22.4	30.0	51.0	93.5	150.8	230.0	345.0
NaCl	6.6	17.2	25.5	33.5	......	......	......	......	......	......
NH_4Cl	6.5	19.0	29.7	39.6	56.2	......	......	......	......	......
NH_4NO_3	10.0	30.0	52.0	74.0	108.0	248.0	682.0	1370.0	2400.0	4099.0
$C_4H_6O_6$	17.0	52.0	87.0	123.0	177.0	374.0	980.0	3774.0	......	......

Properties of Saturated Steam. For tables of total heat, latent heat, and other properties of steams of ether, alcohol, acetone, chloroform, chloride of carbon, and bisulphide of carbon, see Röntgen's Thermodynamics (Dubois's translation). Tables of the properties of ammonia, carbon dioxide, sulphur dioxide and other refrigerants are given in Section 10, Refrigeration. See pp. 10–14 to 10–20.

RISE OF BOILING POINT OF SALT SOLUTIONS.—The boiling point of salt solutions is raised as the strength of the solution increases. Table 6 shows the number of parts of various salts that must be added to 100 parts of water, to raise the boiling point the number of degrees given.

Table 7.—Latent Heats of Vaporization

Substance	Boiling Point, deg. F.	Latent Heat, B.t.u. per lb.	Substance	Boiling Point, deg. F.	Latent Heat, B.t.u. per lb.
Acetic acid	244.4	152.82	Ethyl iodide	159.8	84.6
Alcohol (ethyl)	172.4	369.0	Heptane	194.	140.0
" (methyl)	150.8	480.6	Hexane	158.	142.6
Aniline	363.0	198.0	Iodine	....	42.3
Benzine	176.3	167.2	Mercury	674.6	117.0
Bromine	141.8	82.1	Octane	266.0	126.0
Carbon disulphide	115.0	150.8	Pentane	86.	154.4
Chloroform	142.1	105.3	Sulphur	600.8	651.6
Ether (ethyl)	94.3	159.1	Toluol	230.5	154.8
Ethyl bromide	100.7	108.7	Turpentine	318.8	133.3

Table 8.—Latent Heats of Fusion

Substance	Melting Point, deg. F.	Latent Heat, B.t.u. per lb.	Substance	Melting Point, deg. F.	Latent Heat, B.t.u. per lb.
ALLOYS:			Ice		143.33
Pb Sn			(from sea water)	17.6	97.2
30.5 69.5	361.4	30.6	Lead	620.6	9.65
36.9 63.1	354.2	27.9	Mercury	−38.2	5.07
63.7 36.3	351.5	20.88	Naphthalene	175.8	64.11
77.8 22.2	349.7	17.17	Nickel	2615	8.35
1 9	456.8	50.4*	Palladium	2813	65.34
Aluminum	1216.4	138.24	Phosphorus	111.6	8.94
Benzol	41.7	55.08	Platinum	3191	48.96
Bromine	18.9	29.16	Potassium	143.6	28.26
Bismuth	514.4	22.75	Paraffin	126.3	63.18
Cadmium	609.3	24.59	Silver	1761.8	37.92
Calcium chloride + $6H_2O$	85.3	73.26	Sodium	206.6	57.06
Copper	1981.4	75.6	Spermaceti	111.0	66.56
Iron, gray cast		41.4	Sulphur	239	16.87
" white		59.4	Tin	449.6	25.2
		90	Wax	143.2	76.14
Iodine		21.08	Zinc	786.2	50.63

* Total heat from 32° F.

Table 9.—Freezing Mixtures

Ingredient	Parts	Mixed with	Parts	Temp. before Mixture, deg. F.	Temp. of Mixture, deg. F.	Lowering of Temp., deg. F.
Calcium chloride (crystals)	250	Water	100	51.4	11.1	40.3
Sodium hyposulphite (crystals)	110	"	100	51.2	11.0	40.2
Potassium sulphate	10	Snow	100	30.2	28.6	1.6
Sodium carbonate (crystals)	20		100	30.2	28.4	1.8
Calcium chloride	30		100	30.2	12.4	17.8
Ammonium chloride	25		100	30.2	4.3	25.9
Ammonium nitrate	45		100	30.2	1.9	28.3
Sodium chloride	33		100	30.2	− 6.3	36.5
Sulphuric acid + water (H_2SO_4, 66.1%)	1		1.097	30.2	−34.6	64.8
" " " "	1		13.08	30.2	3.2	27.
Alcohol at 39.2°	77		73	32.	−22.	54.

EXPANSION BY HEAT

EXPANSION OF AIR.—In the Centigrade scale the coefficient of expansion of air per degree is 0.003665 = $^1/_{273}$; that is, the pressure being constant, the volume of a perfect gas increases $^1/_{273}$ of its volume at 0° C. for every increase in temperature of 1°. In

Table 1.—Expansion of Solids

Substance	t° F.*	Coefficient of Linear Expansion, per deg. F.†
Aluminum.	*M*	0.00001233
	1112	.00001750
Antimony	*M*	.00000587
Bismuth	*M*	.00000731
Brass, cast	*M*	.00001042
" wire	*M*	.00001072
Bronze (3 Cu, 1 Sn).	*M*	.00001024
" (86.3 Cu, 9.7 Sn, 4 Zn)	104	.00000990
Cadmium	*M*	.00001755
Carbon, diamond	104	.00000066
" gas carbon	104	.00000300
" graphite	104	.00000437
" anthracite	104	.00001154
Caoutchouc	*M*	.00003650
		.00003811
Cobalt	104	.00000687
Constantan.	39–84	.00000846
Copper	*M*	.00000926
Ebonite	77–95	.00004677
Fluorspar, (	*M*	.00001083
German silver	*M*	.00001020
Gold	*M*	.00000817
Gold-platinum (2 Au, 1 Pt)	*M*	.00000846
Gold-copper (2 Au, 1 Cu)	*M*	.00000862
Glass, tube	*M*	.00000463
" plate	*M*	.00000495
" crown	*M*	.00000498
" flint	*M*	.00000438
" Jena 16III.	*M*	.00000450
" " 59III.	*M*	.00000322
" quartz	61–1832	.00000032
Gutta-percha	68	.00011016
Ice	−4 to +30	.00002833
Indium	104	.00002317
Iridium	*M*	.00000489
Iron, soft	104	.00000672
" cast	104	.00000589
" wrought	0–212	.00000633
" steel	104	.00000734
" " annealed	*M*	.00000608
Lead	*M*	.00001516
Lead-tin (2 Pb, 1 Sn).	*M*	.00001393
Magnalium	54–102	.00001322
Magnesium	104	.00001450
Manganin		.00001005
Marble	59–212	.00000650
Nickel	104	.00000566
Osmium	104	.00000365
Palladium	104	.00000653
Paraffin	32–61	.00005923
	61–100	.00007238
	100–120	.00026501
Phosphorus	32–104	0.00006961
Platinum	104	.00000499
Platinum-iridium (10 Pt + 1 Ir)	104	.00000491
Platinum-silver (1 Pt + 2 Ag)	*M*	.00000846
Porcelain	68–1454	.00000229
" Bayeux	1832–2552	.00000307
Potassium	32–122	.00004607
Quartz (‖ to axis)	32–176	.00000443
" (⊥ to axis)	32–176	.00000743
Rhodium	104	.00000472
Rock salt	104	.00002244
Rubber, hard	32	.00003839
Ruthenium	104	.00000535
Selenium	*M*	.00003669
Silicon	104	.00000424
Silver	104	.00001067
Sodium	32–194	.00012554
Speculum metal	*M*	.00001074
Sulphur	*M*	.00006556
Tellurium	*M*	.00002048
Thallium	104	.00001678
Tin	*M*	.00001275
Topaz, ‖ to lesser horiz axis	*M*	.00000462
Topaz, ‖ to greater horiz. axis	*M*	.00000464
Topaz, ‖ to vertical axis	*M*	.00000262
Type metal	62–489	.00001084
Vulcanite	32–64	.00003533
Wedgwood ware	*M*	.00000494
Wood, ash ‡	*M*	.00000528
" " ¶	35.6–93.2	....
" beech ‡	35.6–93.2	.00000143
¶	35.6–93.2	.00003411
chestnut ‡	35.6–93.2	.00000361
" ¶	35.6–93.2	.00001806
elm ‡	35.6–93.2	.00000314
¶	35.6–93.2	.00002461
mahogany ‡	35.6–93.2	.00000201
	35.6–93.2	.00002244
maple ‡	35.6–93.2	.00000354
" ¶	35.6–93.2	.00002689
oak ‡	35.6–93.2	.00000273
" ¶	35.6–93.2	.00003022
pine ‡	35.6–93.2	.00000301
" " ¶	35.6–93.2	.00001894
" walnut ‡	35.6–93.2	.00000366
" " ¶	35.6–93.2	.00002689
Wax, white	50–79	.00012778
	79–88	.00017333
	88–109	.00027000
	109–135	.00084594
Zinc	*M*	.00001653

* *M* = Mean coefficient, 32°–212°. † Cubical expansion may be taken as (3 × linear expansion). Coefficient of expansion per deg. C. = coefficient per deg. F × $^9/_5$. ‡ Parallel to fiber. ¶ Across fiber.

Fahrenheit units it increases $1/_{491.6} = 0.002034$ of its volume at 32° F. for every increase of 1° F.

EXPANSION OF SOLIDS, LIQUIDS AND GASES.—The coefficients of expansion of various solids, liquids and gases are given in Tables 1 to 3. These tables are based on the tables compiled by the Smithsonian Institution, Washington, from various sources.

EXPANSION OF STEEL AT HIGH TEMPERATURES. (Charpy and Grenet, *Comptes Rendus*, 1902.)—Coefficients of expansion (for 1° C.) of annealed carbon and nickel steels at temperatures at which there is no transformation of the steel are given in Table 4. The results seem to show that iron and carbide of iron have appreciably the same coefficient of expansion.

Table 2.—Coefficients of Cubical Expansion of Liquids, per Degree F.

Liquid	$C \times$	Liquid	$C \times 10^3$
Acetic acid	0.595	Glycerine	0.281
Acetone	.826	Hydrochloric acid (33.2% solution)	.253
Alcohol, amyl	.501	Mercury	.101
" ethyl	.622	Olive oil	.401
" methyl	.666	Pentane	.893
Benzine	.687	Potassium chloride (24.3% solution)	.196
Bromine	.629	Phenol	.606
Calcium chloride (5.8% solution)	.139	Petroleum (density 0.8467)	.531
" " (40.9% solution)	.254	Sodium chloride (20.6% solution)	.230
Carbon disulphide	.677	" sulphate (24% solution)	.228
" tetrachloride	.687	Sulphuric acid (100%)	.310
Chloroform	.707	Turpentine	.541
Ether	.920	Water	See p. 2–03

Table 3.—Coefficients of Expansion of Gases, per Degree F.

Pressures given are in cm. of mercury

Substance	Constant Volume		Constant Pressure	
	Pressure	Coefficient	Pressure	Coefficient
Air.	75.2	0.00203667	76.	0.00203944
32°–212°	100.1	.00204133	100.1	.00204044
	200.	.00205016	257.	.00205167
Argon	51.7	.00203778		
Carbon dioxide	76.0	.00204756	76.0	.00206111
32°–212°	51.8	.00205400	51.8	.00205961
3	99.8	.00207011	99.8	.00207833
Carbon monoxide	76.	.00203706	76.	.00203833
Helium	56.7	.00203611		
Hydrogen, 32°–212°	76.4	.00202800	100.	.00203333
Nitrogen, 32°–212°	100.2	.00204133	.....	
Nitrous oxide	76.	.00204222	76.	.00206611
Oxygen	75.9	.00203783		
Sulphur dioxide	76.	.00213611	76.	.00216833
Water vapor, 32°–392°			76.	.00218778

Table 4.—Coefficients of Expansion of Steel at High Temperatures per Degree C.

Composition of Steels				Mean Coefficients of Expansion from			Coefficients between	
C	Mn	Si	P	1.5° to 200°	200° to 500°	500° to 650°		
0.03	0.01	0.03	0.013	11.8×10^{-6}	14.3×10^{-6}	17.0×10^{-6}	880° & 950°	24.5×10^{-6}
0.25	0.04	0.05	0.010	11.5	14.5	17.5	800° & 950°	23.3
0.64	0.12	0.14	0.009	12.1	14.1	16.5	720° & 950°	23.3
0.93	0.10	0.05	0.005	11.6	14.9	16.0	" "	27.5
1.23	0.10	0.08	0.005	11.9	14.3	16.5	" "	33.8
1.50	0.04	0.09	0.010	11.5	14.9	16.5	" "	36.7
3.50	0.03	0.07	0.005	11.2	14.2	18.0	" "	33.3

Nickel steels			Mean Coefficients of Expansion from				
Ni	C	Mn	15° to 100°	100° to 200°	200° to 400°	400° to 600°	600° to 900°
26.9	0.35	0.30	11.0×10^{-6}	18.0×10^{-6}	18.7×10^{-6}	22.0×10^{-6}	23.0×10^{-6}
28.9	0.35	0.36	10.0	21.5	19.0	20.0	22.7
30.1	0.35	0.34	9.5	14.0	19.5	19.0	21.3
34.7	0.36	0.36	2.0	2.5	11.75	19.5	20.7
36.1	0.39	0.39	1.5	1.5	11.75	17.0	20.3
32.8	0.29	0.66	8.0	14.0	18.0	21.5	22.3
35.8	0.31	0.69	2.5	2.5	12.5	18.75	19.3
37.4	0.30	0.69	2.5	1.5	8.5	19.75	18.3
25.4	1.01	0.79	12.5	18.5	19.75	21.0	35.0
29.4	0.99	0.89	11.0	12.5	19.0	20.5	31.7
34.5	0.97	0.84	3.0	3.5	13.0	18.75	26.7

INVAR, an alloy of iron with 36% Ni, has a smaller coefficient of expansion with the ordinary atmospheric changes of temperature than any other metal or alloy known. This alloy, sold under the name of Invar, is used for scientific instruments, pendulums of clocks, steel tape-measures for accurate survey work, etc. The Bureau of Standards found its coefficient of expansion to range from 0.000000374 to 0.000000440 for 1° C., or about $^1/_{28}$ of that of steel. For all surveys except in the most precise geodetic work a tape of invar may be used without correction for temperature. (*Eng. News*, Aug. 13, 1908.)

PLATINITE, an alloy of iron with 42% Ni, has the same coefficient of expansion and contraction at atmospheric temperatures as has glass. It can, therefore, be used for the manufacture of armored glass, that is, a plate of glass into which a network of steel wire has been rolled, and which is used for fireproofing, etc. It also can be used instead of platinum for the electric connections passing through the glass plugs in the base of incandescent electric lights. (Stoughton's Metallurgy of Steel.)

HEAT TRANSMISSION

By W. J. King

References. Heat Transmission, by W. H. McAdams, McGraw-Hill, N. Y., 1933. The Calculation of Heat Transmission, by M. Fishenden and O. A. Saunders, H. M. Stationery Office, London, 1932. The Basic Laws and Data of Heat Transmission, by W. J. King, *Mech. Engg.*, March-Aug. inclusive, 1932.

THE FUNDAMENTAL HEAT TRANSFER PROCESSES are Conduction, Convection and Radiation. Evaporation and Condensation are special forms or combinations of conduction and convection.

Conduction is the transmission of heat by molecular vibration from one part of a body to another or from one body to another body in direct contact with it.

Convection is the transfer of heat between a fluid and a surface by the circulation or mixing of the fluid. In free or natural convection the fluid motion is caused by gravity forces due to difference in density between the hotter and cooler parts; in forced convection the motion is produced artificially, as by a pump, blower, or other external forces not connected with the temperature of the fluid.

Radiation is the transmission of heat in the form of radiant energy or wave motion in the ether, from one body to another across an intervening space. This term sometimes is loosely used to denote dissipation of heat from the outer surface of a furnace or pipe, which usually includes both radiation and convection.

Heat may be transmitted by any one of the three processes acting alone, or by combinations acting in series or in parallel, as in the case of most practical applications. Although it is sometimes convenient to deal only with the combined or overall heat transfer, as from one fluid to another across a dividing wall, this sort of treatment does not bring out the effects of the individual components, and the data are not of such general utility as when these components are treated separately. The present section is therefore devoted to the basic individual processes listed above. Whenever the total or overall heat transmission is desired, care should be taken to include all of the processes acting in conjunction, as shown in the section on Heat Transfer Applications on page 3-34.

1. CONDUCTION

CONDUCTIVITY.—In conduction problems, the fundamental unit is the *thermal conductivity* K, which is a specific property of a substance. In engineering units, conductivity is defined as the rate of heat flow in B.t.u. per hour through a cross-sectional area of 1 sq. ft. for a temperature difference of 1 deg. F. across a thickness or length of 1 in. in the direction of the heat flow. In any case, if q = heat flow, B.t.u. per hr.; A = area,

Table 1.—Conversion Factors for K

Unit of Heat	Unit of Time	Unit of Area	Unit of Temperature Diff.	Unit of Thickness	Divide K by
B.t.u.	hour	sq. ft.	deg. F.	inch	1
B.t.u.	hour	sq. ft.	deg. F.	foot	12
Gram-calorie	second	sq. cm.	deg. C.	cm.	2903
Kg.-calorie	hour	sq. meter	deg. C.	meter	8.064
Watt-seconds	second	sq. cm.	deg. C.	cm.	694
Kw.-hr.	hour	sq. ft.	deg. F.	inch	3415

sq. ft.; L = thickness, in.; = temperature at hot end of path; t_2 = temperature at cold end, deg. F., then

$K = \cdot$

L

Factors for obtaining values of K in various other common units are given in Table 1.

In the case of radial flow through a thick-walled cylinder, A should be taken as the logarithmic mean of the outer and inner areas; $A = (A_2 - A_1)/\log_e (A_2/A_1)$.

It is sometimes convenient to obtain a heat transfer coefficient h in B.t.u. per hr. per sq. ft. per deg. F., for a given thickness of a solid material, such as the metal wall of a pipe. This is done by dividing the conductivity by the thickness, $h = K/L$ . . . . . [3].

Conductivity of Gases

THE THERMAL CONDUCTIVITY OF ANY GAS may be calculated with fair accuracy by the formula

$$K = bZc_v \qquad [4]$$

where K = B.t.u. per hr. per sq. ft. per deg. F. per in.; Z = viscosity in centipoises; c_v = specific heat at constant volume; b = a constant depending on the number of atoms in the molecule as follows: monatomic gases, $b = 71$; diatomic gases, $b = 55.2$; triatomic gases, $b = 49.4$; complex gases, $b = 38$. For all gases K increases with the temperature, but is practically independent of the pressure, except at very low or high pressures. This formula is based on the kinetic theory and has been verified experimentally.

If the values of K in Table 2 are used to compute the heat flow across a gas layer, it must be remembered that a considerable amount of heat also may be transferred by convection and radiation as parallel processes.

The Conductivity of Air at any temperature T, deg. F. absolute, may be obtained from the formula

$$K = (0.01 + 1.076 \times 10^{-6}T)\sqrt{T}/(1 + 211/T) \qquad [5]$$

Conductivity of Liquids and Solutions

The following empirical formula is given by J. F. Dowine-Smith (*Ind. and Engg. Chem.*, vol. xxii, 1930, p. 1246) for the value of K for pure liquids

$$K = 2.35S^{2.15}C^{1.55}M^{0.192}/Z^{0.12} \qquad [6]$$

where S = specific gravity; C = specific heat; M = molecular weight; Z = viscosity, centipoises.

The conductivity of most organic liquids decreases slightly with increasing temperature. For water, K increases to a maximum at 270° F., and then decreases until its value at 570° F. is the same as at 40° F. The conductivity of aqueous solutions generally is lower than that of water by an amount roughly proportional to the concentration of the solute.

Conductivity of Metals and Alloys

According to the Weidemann-Franz-Lorenz Law (See *Jour.* Inst. of Metals, vol. xxxix, 1928, p. 337) the thermal conductivity K, B.t.u. per hr. per sq. ft. per deg. F. per inch, of any metal or alloy can be obtained from the relation $K\omega = 58\ T$, where ω = electrical resistivity, ohms per circular-mil-ft.; T = temperature in deg. F., absolute. Except

Table 2.—Thermal Conductivity, K, of Gases

(In B.t.u. per hr. per sq. ft. per inch thickness per deg. F. difference in temperature)

Gas	Temperature, deg. F.	K	Gas	Temperature, deg. F.	K
Air	32	0.163	Hydrogen	32	1.130
"	212	0.211	"	212	1.435
Ammonia	32	0.141	Methane	32	0.210
"	212	0.206	Neon	32	0.031
Argon	32	0.110	Nitrogen	32	0.163
Carbon dioxide	32	0.097	"	212	0.210
" "	212	0.133	Nitrous oxide	32	0.100
Carbon monoxide	32	0.155	Nitric oxide	32	0.144
Chlorine	32	0.053	Oxygen	32	0.165
Ethane	32	0.126	"	212	0.215
Ethylene	32	0.114	Steam	212	0.162
Helium	32	0.970	"	572	0.254
			Sulphur dioxide	32	0.056

at very low temperatures, this relation has been found to hold fairly closely for pure metals and approximately for alloys. For pure metals any variation of conductivity with temperature generally is small, although in some cases there is an appreciable decrease at higher temperatures. For most alloys, the value of K increases, and for the ferrous metals it decreases with rising temperatures.

The addition of a small amount of another metal, or an impurity, usually will result in a sharp drop in conductivity of a pure metal, particularly when a solid solution is formed. Practically all of the alloy steels have lower conductivities than wrought iron. When the percentage of added constituents (C, Co, Cr, Mn, Mo, Ni, Si, V, W, etc.) is small, the value of K usually is around 280, with lower values as the percentage increases.

MISCELLANEOUS SOLID MATERIALS.—In general the conductivities of non-metallic solid materials increase very considerably with density, temperature, and moisture

Table 3.—Thermal Conductivity, K, of Liquids and Aqueous Solutions

(In B.t.u. per hr. per sq. ft. per 1 in. thickness per deg. F. temperature difference)

Liquid	Temperature, deg. F.		Liquid		Temperature, deg. F.	K
Acetic acid	68	1.20	Oil, turpentine		55	0.88
Acetone	68	1.24	Pentane (n)		68	0.94
Alcohol (methyl)	68	1.44	Sulphur dioxide		68	2.34
" (ethyl)	68	1.26	Toluene		68	1.05
" (amyl)	68	1.02	Vaseline		68	1.27
Aniline	32	1.25	Water		32	3.85
Benzene	68	1.18			140	4.54
Carbon dioxide	68	1.45			270	4.76
Carbon disulphide	68	1.11			420	4.55
Chloroform	54	0.84				
Ether	68	0.95	Solutions	Specific Gravity		
Glycerine	68	1.98				
Kerosene	68	1.05	$CuSO_4$	1.160	40	3.42
Oil, castor	68	1.24	KCl	1.026	55	3.37
" lubricating	68	1.16	NaCl	1.178	40	3.34
" olive	39	1.21	H_2SO_4	1.054	69	3.67
" petroleum	55	1.03	$ZnSO_4$	1.134	40	3.50

Table 4.—Thermal Conductivity, K, of Metals and Alloys.

(In B.t.u. per hr. per sq. ft. per 1 in. thick per deg. F. temperature difference)

Metal	Temp., deg. F.	K	Metal	Temp., deg. F.	K
Aluminum	32	1475	Manganin	32	144
"	400	1560	"	212	182
Antimony	32	128	Mercury	32	58
"	212	115	"	120	55
Bismuth	64	56	Molybdenum	32	1000
"	212	47		212	965
Brass, yellow	68	770	Monel metal	90	242
" "	400	1010	Nichrome	90	95
" red	32	715	Nickel	32	408
" "	212	836		400	390
Cadmium	32	648	Plat[illegible]	32	482
"	212	624		212	500
Constantan	68	162	+ 10% Ir.	63	215
"	212	186	+ 10% Rh.	63	212
Copper	32	2680	Platinoid	64	174
"	400	2605	Potassium	68	675
Gold	32	2070	Rhodium	63	610
"	212	2030	Silver	32	2850
Iron, pure	64	468		400	2600
" "	212	438	Sodium	32	950
" cast	32	350	"	212	835
" "	400	220	Tantalum	32	376
" wrought	32	423	"	3092	505
" "	400	360	Tin	32	450
Steel (1% C.)	32	310	"	212	425
" "	400	300	Tungsten	64	1381
Lead	64	241	"	2912	723
"	212	235	Zinc	32	775
Magnesium	32	1090		212	750
	400	1015			

content. Conductivities given in Table 5 are representative values for conditions of normal use.

Table 5.—Thermal Conductivity, K, of Various Solids

In B.t.u. per hr. per sq. ft. per 1 in. thick per deg. F. temperature difference. For conductivity of heat insulators, see p. 3-71; of building materials, see p. 3-61.

Material	K	Material	K	Material	K
Carbon, graphite	35.0	Glass, flint	5.8	Sand, dry	2.3
Carborundum	1.46	" pyrex	7.5	" moist	8.0
Cardboard	1.8	Granite	15.0	Sandstone	12.0
Celluloid	1.45	Ice at 32° F.	15.0	Sawdust	0.35
Chalk	5.8	Lime	0.84	Slate, ⊥ to cleavage	9.8
Coke, powdered	1.27	Mica	4.0	" ‖ to cleavage	17.4
Concrete, cinder	2.4	Paper	1.0	Snow, fresh	0.75
" stone	8.0	Porcelain	10.0	" old	3.5
Cotton	0.4	Quartz, ⊥ to axis	47.0	Soil, dry	1.0
Earth's crust	11.6	" ‖ to axis	82.0	" moist	10.0
Fiber, sheet	2.0	Rock salt	48.0	Sulphur	1.8
Firebrick	9.0	Rubber, solid	1.2	Vulcanite	2.5
Glass, window	5.5	" sponge	0.35	Wool	0.25
	6.0				

2. CONVECTION

Free or Natural Convection

Cases treated under this heading are for the conditions of still air or a body immersed in an unagitated fluid. Coefficients refer only to the convection component of the total heat transfer, and do not include radiation, which, in the case of gases, generally should be computed separately as an additional component.

THE CONVECTION COEFFICIENT h, sometimes called the *film coefficient*, is expressed in B.t.u. per hr. per sq. ft. of surface per deg. difference in temperature between the surface and the main body of the fluid. In free convection, the value of the coefficient h is affected very little by the nature of the surface, but depends principally upon the size, temperature difference and physical properties of the fluid.

Free Convection in Air.—For vertical plane surfaces in air,

$$h = 0.30(\theta/L)^{1/4} \qquad [7]$$

where h = convection coefficient, B.t.u. per hr. per sq. ft. per deg. F.; θ = temperature difference, deg. F.; L = height, ft. The convection from a horizontal surface facing upward is about 30% greater, and facing downward 30% less, than in the vertical position.

The convection coefficient for horizontal or vertical pipes in air is given by

$$h = 0.42(\theta/d)^{1/4} \qquad [8]$$

where d = pipe diameter, in., other notation being as before.

Free Convection in Liquids.—In this case the convection coefficient represents the total heat transfer, since radiation is not involved. Ordinarily, the effects of the shape or size of the body are not significant, except that for fine wires the coefficients will be considerably higher than those given below.

Convection in Water.—For bodies of ordinary size in water

$$h = 0.165(t_w + 30)\sqrt{\theta} \qquad [9]$$

where t_w = water temperature deg. F., other notations being as before. This formula does not apply when the surface temperature is above the boiling point. Values of h for various values of t_w and θ are given in Table 6.

Table 6.—Heat Transfer Cofficients for Surfaces in Unagitated Water

B.t.u. per hr. per sq. ft. per deg. F.

Water Temperature, t_w, deg. F.	Temperature Difference θ, deg. F.									
	10	20	30	40	50	60	70	80	90	100
40	37	52	63	73	82	90	96	103	110	116
60	47	66	81	94	105	115	124	133	141	149
80	57	81	100	115	128	140	152	162	172	182
100	68	96	117	136	152	166	179	192	203	214
120	78	110	136	156	175	192	207	221	235	....
140	89	125	154	177	198	217	235	....	....	....
160	99	140	172	198	222	....	....	....	....	....
180	110	155	190	....	....	....	....	....	....	....

Convection in Oils.—If Z = viscosity in centipoises at the average of the surface and oil temperatures, the convection coefficient may be obtained from $h = 17\theta^{1/4}/Z^{0.4}$ . . . . [10].

Forced Convection

When the fluid is circulated artificially over the heat transfer surface, the value of the coefficient h is governed by the velocity and physical properties of the fluid, and by the size, shape, arrangement and nature of the surface. In general, roughening the surface, or anything that promotes turbulence in the fluid flow, will increase the heat transfer coefficient. In liquids, the convection coefficient represents the total heat exchange. This also generally is true of gases, as regards the exchange of heat between the surface and the gas, but with exposed surfaces the additional effect of radiation to or from the surroundings may be relatively significant, particularly at low velocities and high temperatures. The effect of humidity generally is negligible, except when the temperature of the surface is below the dew-point of the gas. Condensation then will occur as an additional process, governed by the vapor pressure difference.

Gas Film Coefficients

In the following formulas, which represent the best available data from various sources, h = convection coefficient, B.t.u. per hr. per sq. ft. per deg. F. temperature difference; v = gas velocity, ft. per sec.; G = mass velocity, lb. per sec. per sq. ft. of cross-sectional area = (linear velocity $v \times$ density, lb. per cu. ft.); c_p = specific heat at constant pressure; Z = viscosity, centipoises; t = average gas temperature, deg. F.; d = pipe diam., in.

For any gas flowing inside a tube, $h = 36\, c_p\, G^{0.8}(Z/d)^{0.2}$. . . . . [11]

For air flowing at right-angles across a single rod or pipe,

$$h = 8.2\, G^{0.58}(1 + 0.000576\, t)/d^{0.42} \quad [12]$$

In Tube Banks, the coefficient increases with the number of rows, and is somewhat higher when the tubes are staggered than when they are arranged in line with the flow. **For air flowing across staggered tubes 4 or more rows deep,**

$$h^* = \{1.7(t + 460)^{0.3}\, G^{2/3}\}/d^{1/3} \quad [13]$$

For smooth plane surfaces in ordinary atmospheric air at velocities $v < 15$ ft. per sec.,

$$h = 0.8 + 0.22\, v \quad [14]$$

At higher velocities,

$$h = 0.56\, v^{3/4}. \quad [15]$$

Coefficients for rough surfaces such as brick, concrete and stucco are from 20 to 50% higher than for smooth surfaces.

The curves, Fig. 1, give coefficients for air at atmospheric pressures and temperatures in the vicinity of 100 deg. F. For other conditions, equivalent velocity * may be obtained from the expression $v = 14.3\, G$, where G = mass velocity, lb. per sec. per sq. ft. of flow area. The formulas and curves for air also may be used for similar gases such as N_2, O_2 and CO; in case of flue gases the value of h usually is about 25% higher. Curves marked Cross Flow are representative of the data for finned tubing, although the latter vary considerably with size and spacing of fins, etc. Table 7 shows the effect of tube diameter upon the above coefficients.

Liquid Film Coefficients

LIQUIDS FLOWING IN PIPES.—For the heating of liquids in *turbulent flow* in pipes, Sherwood and Petrie (*Ind. and Engg. Chem.*, vol. xxiv, 1932, p. 736) give an equation which may be reduced to the form

$$h = 4.37 K^{0.6}\, G^{0.8}\, c^{0.4}/d^{0.2}\, Z^{0.4}. \quad [16]$$

where h = heat transfer coefficient, B.t.u. per hr. per sq. ft. per deg. F. difference between the pipe and liquid temperatures; K = thermal conductivity of the liquid, B.t.u. per hr. per sq. ft. per deg. F. per in.; G = mass velocity, lb. per sq. ft. per sec. = (linear velocity, ft.

Table 7.—Effect of Tube Diameter upon Forced Convection Coefficients

Dia	0.1	0.25	0.5	0.75	1	1.5			
	Multiply coefficient for 1-in. tube by								
Flow Inside Tube	1.59	1.32	1.15	1.06	1.0	0.92	0.87	0.76	0.66
Cross Flow, Single Tube	2.60	1.77	1.33	1.13	1.0	0.84	0.75	0.56	0.42
Tube banks	2.15	1.58	1.25	1.10	1.0	0.87	0.79	0.63	0.50

* In the case of tube banks the value of G or v should be taken as the maximum velocity between the tubes, *i.e.*, referred to the minimum free area rather than the face area of the bank.

Table 8.—Heat Transfer Coefficients for Water Flowing in a 1-in. Pipe

B.t.u. per hr. per sq. ft. per deg. F.

Water Temperature, deg. F.	Velocity, ft. per sec.								
	0.5	1	2	3	4	5	6	8	10
40	145	252	438	607	764	915	1060	1330	1585
60	166	288	501	694	873	1045	1210	1520	1810
80	186	324	564	780	982	1175	1360	1710	2035
100	207	360	626	867	1090	1305	1510	1900	2260
120	228	396	689	954	1200	1435	1665	2085	2490
140	248	432	750	1040	1310	1570	1815	2280	2720
160	269	468	815	1130	1420	1700	1965	2470	2940
180	290	504	876	1215	1530	1830	2120	2660	3170
200	311	540	9[illegible]0	1300	1635	1960	2270	2850	3400

per sec. × density, lb. per cu. ft.); c = specific heat; d = inside diameter of tube, in.; Z = viscosity of liquid, centipoises. Fluid properties were taken at the average temperature of the main body of the liquid. This formula represents tests on water, acetone, benzine, kerosene and n-butyl alcohol. It is in good agreement with the results of other investigators and should be applicable to any fluid. Table 8 gives values of h for water flowing in a clean 1-in. pipe; for other pipe diameters multiply h by factors in Table 7.

Corrosion, Scale or Dirt on the pipe surface will reduce the coefficient considerably, in extreme cases by as much as 50%. Effect of pipe length generally is negligible when length > 20 diameters; for shorter pipes the coefficient may be appreciably higher.

COEFFICIENTS FOR COOLING.—When the fluid properties do not vary very widely over the temperature range, formula [16] will apply to cooling as well as to heating of liquids. With petroleum oils, however, the variation of viscosity is so rapid that it is difficult to obtain a formula that is valid for all conditions. Most of the data on heating of oils in turbulent flow are represented with reasonable accuracy by

$$h = 70\, v/Z^{0.63} \quad . \quad . \quad . \quad . \quad . \quad [17]$$

where v = linear velocity, ft. per sec.; Z = viscosity, centipoises, at the average of inlet and outlet temperatures. If the oil is being cooled, the coefficient is usually about 25% lower.

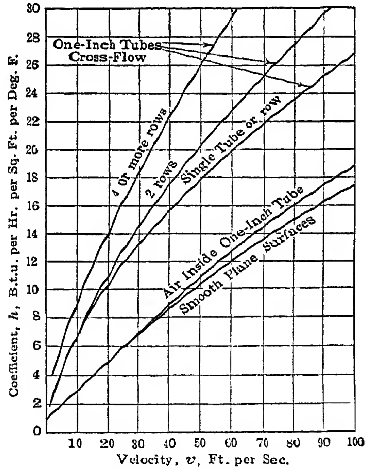
STREAMLINE OR VISCOUS FLOW.—For the heating of any liquid in streamline or viscous flow in pipes, the approximate value of the coefficient may be obtained from

$$h = 15\,(GcK^2/dL)^{1/3} \quad . \quad . \quad . \quad . \quad . \quad [18]$$

or for petroleum oils,

$$h = 43\,(v/dL)^{1/3} \quad . \quad . \quad . \quad . \quad . \quad . \quad [19]$$

where L = heated length of pipe, ft., and coefficient h is based on the arithmetic mean temperature difference between pipe and liquid, other notation being as before. A few data indicate that when the oil is being cooled the coefficient will be roughly 30% lower.



. 1. Coefficient of Heat Transfer for Air at Atmospheric Pressure and Moderate Temperature

FLOW IN CHANNELS OR ANNULAR SPACES.—For fluids flowing in channels or annular spaces it is customary to take d as the hydraulic diameter (4 × cross-sectional area ÷ perimeter), or, with concentric pipes, as the difference between the diameters.

3. RADIATION OF HEAT

Radiant energy may be regarded as a form of wave motion in the ether, which is manifested in various forms, as radio waves, light, heat and x-rays, depending upon the wave lengths. In the range of short wave lengths, known as the visible spectrum, radiant heat and light are identical physically. In the longer wave lengths, which are associated with lower temperatures, the radiation is invisible but it still follows the general laws of optics as regards propagation and reflection; *i.e.*, it travels in straight lines with the speed

of light, the intensity at any point varies inversely as the square of the distance from the source, and for a polished surface the angle of reflection is equal to the angle of incidence. On the other hand, it is important to note that this long-wave radiation from sources at temperatures below incandescence may be emitted, absorbed, reflected or transmitted to a very different degree than in the case of short waves from luminous sources. For example, ordinary window glass will transmit about 90% of the solar radiation falling upon it, but will almost completely absorb radiation from a source at a temperature below 1000° F. Also, at ordinary temperature, a white surface may be as good a radiator or absorber as a black one, whereas absorption of solar radiation increases with the darkness of the color.

DEFINITIONS.—**A Black Body** is a body that absorbs all the radiant energy falling upon it. Such a body also radiates energy at the maximum rate possible by virtue of its temperature.

The Emissivity of a body is the ratio of its radiating power to that of a black body.

The Absorptivity of a body is the fraction of the radiant energy falling upon it that is absorbed.

The Reflectivity of a body is the fraction of the radiant energy falling upon it that is reflected.

THE STEFAN-BOLTZMANN LAW, which has been verified experimentally, states that the total radiation from a black body is proportional to the fourth power of its absolute temperature. This may be expressed in the form

$$q = 0.174 \times 10^{-8} A T^4 \qquad [20]$$

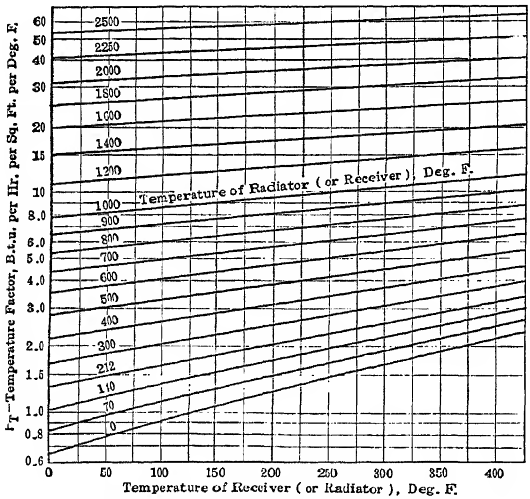


FIG. 2. Values of F_T (Multiply F_T by emissivity factor e' to obtain radiant heat transfer coefficient h_r)

where q = total radiation, B.t.u. per hr.; A = area, sq. ft.; T = absolute temperature, deg. F. For ordinary non-black bodies emissivity e represents the fraction of this amount of heat actually radiated, so that

Values of e are given in Tables 9 and 10. These expressions represent only the emission from a radiating body, and do not take into account the simultaneous absorption from the surroundings. What usually is desired in practical applications is the net exchange of radiation between a body and its surroundings, or between two bodies. In general, this depends, in a rather complex way, upon the solid geometry of the case (see H. C. Hottel, *Mech. Engg.*, vol. lii, 1930, p. 699), but for certain simple cases it can be calculated by convenient formulas.

In the following cases the net radiation is

$$q/A = 0.174\, e' \left[(T_1/100)^4 - (T_2/100)^4 \right] \text{ B.t.u. per hr. per sq. ft.} \qquad [21]$$

where T_1 = the higher, and T_2 = the lower absolute temperature, and e' = emissivity factor. It frequently is convenient to express this in the form of a radiant heat transfer coefficient, h_r, by dividing by the temperature difference. Thus, let F_T be the **temperature factor** defined by the expression

$$F_T =$$

The coefficient is then $h_r = e'F_T$ B.t.u. per hr. per sq. ft. per deg. F. Values of F_T for ordinary Fahrenheit temperatures are given in Fig. 2.

Case 1.—Radiation or absorption by a relatively small body in an enclosure. In this case $e' = e$, the emissivity of the body. As the size of the body approaches that of the enclosure, the emissivity factor approaches the value given for Case 2.

Case 2.—Two parallel planes whose areas are large compared to their distance apart. This involves the emissivities e_1 and e_2, of both surfaces, and

$$e' = 1/[(1/e_1) + (1/e_2) - 1] \quad . \; . \; . \; . \; . \; . \; . \; . \; . \; [23]$$

Case 3.—Concentric cylinders or spheres. If the heat transfer rate is referred to the area A_1 of the enclosed body, then

$$e' = \frac{1}{\text{[illegible]}} \quad . \; . \; . \; . \; . \; . \; . \; . \; . \; . \; [24]$$

Radiation in Gases and Flames.—In general, radiation or absorption by dry air, oxygen, nitrogen or hydrogen is inappreciable. In the case of carbon dioxide or water vapor the effect may be considerable, particularly for thick gas layers, high concentrations and high temperatures. For the calculation of radiation from gases and flames, see Haslam and Hottel, *Trans.* A.S.M.E. FSP-50-3, 1928, and Wilson, Lobo and Hottel, *Ind. and Engg. Chem.*, vol. 24, 1932, p. 486.

Emissivity and Absorptivity

KIRCHHOFF'S LAW states that the emissivity of any body is equal to its absorptivity for the same kind of radiation, *i.e.*, the same wave lengths. For practical purposes this law may be approximated by the statement that the absorptivity of any material is equal to its emissivity when the radiation is invisible. It should be noted, however, that a material or surface may have a low absorptivity for solar radiation and yet have a high emissivity at ordinary temperatures, as in the case of white paint.

EMISSIVITY OF METALLIC SURFACES.—The emissivity of a metallic surface depends to a marked extent upon the degree of oxidation. Values of e for bright metal surfaces are given in Table 9. The figures given in Table 10 show the normal variation of e for moderately to badly oxidized surfaces. For slightly oxidized or tarnished surfaces values of e intermediate between those of Table 9 and Table 10 should be used.

Table 9.—Emissivity e of Polished Metal Surfaces

Metal	Temperature		
	70° F.	1000° F.	3000° F.*
Aluminum	0.05	0.075	......
Brass	0.05	0.06	......
Copper	0.04	0.08	0.15
Gold	0.03	0.05	......
Iron, cast or wrought	0.20	0.25	0.28
Lead	0.08	......	......
Monel metal	0.07	0.10	......
Nickel	0.06	0.10	......
Platinum	0.036	0.10	0.20
Silver	0.025	0.035	......
Steel	0.20	0.25	0.28
Tin	0.08	......	......
Tungsten	0.03	0.09	0.25
Zinc	0.10	......	......

* Or molten, if melting point is below 3000° F.

Table 10.—Emissivity e of Oxidized Metals at Temperatures below 1500° F.

Metal	e	Metal	e
Aluminum	0.10–0.20	Iron and steel	0.60–0.90
Brass	0.25–0.60	Monel metal	0.40–0.50
Copper	0.55–0.75	Nickel	0.40–0.60

The Emissivity of Aluminum or Bronze Paints varies from 0.3 to 0.6, depending on age and amount of lacquer.

EMISSIVITY OF NON-METALLIC MATERIALS.—A careful study of the results of tests on several hundred different materials reported by about 30 investigators indicates that practically all non-metallic materials, such as porcelain, glass, rubber, paper, cloth, refractories, building materials, enamels and paints *of any finish or color* have emissivities between 0.85 and 0.95. In view of the lack of agreement in many cases, a value of $e = 0.9$ is recommended for all such materials.

SOLAR RADIATION.—On a clear day, the average solar radiation at noon upon a surface facing the sun is about $q = 320$ B.t.u. per hr. per sq. ft. Of this, the amount absorbed will be equal to aq, if a = absorptivity of the surface. For solar radiation the value of a depends primarily on the color, although it is influenced somewhat by the nature

and roughness of the surface. Approximate values of a for a few substances are given in Table 11.

Table 11.—Absorptivity a of Various Surfaces for Solar Radiation

Surface	a	Surface	a
Aluminum paint	0.35	Red brick or tile	0.65
Copper, polished	0.50	Silver, polished	0.07
Galvanized iron, new	0.65	Slate, gray	0.90
Lampblack	0.97	Steel, polished	0.45
Magnesium carbonate	0.02	White paper	0.25
Nickel, polished	0.40	Whitewash	0.25

4. HEAT TRANSFER TO BOILING LIQUIDS

Heat transfer coefficients from metal surfaces to boiling liquids are subject to extremely wide variations, depending chiefly on the nature of the surface, temperature difference, and nature and temperature of the liquid. Typical results obtained by various observers are given in Fig. 3. The value of h is affected so greatly by such factors as the amount of air dissolved in the liquid, the design of the apparatus and the presence of scale on the surfaces that it is not possible to obtain an adequate general correlation of the data.

Cryder and Gilliland (*Refrig. Engg.*, Feb. 1933, p. 78) give the results of tests on eleven different liquids, boiling at atmospheric pressure and heated by a horizontal brass cylinder 1 in. diameter, which are represented by the formula

$$h = 0.0617\theta^{2.4} K^{1.8} C^{0.4} S^{2.4}/Z^{3.2}, \qquad [25]$$

where h = heat transfer coefficient, B.t.u. per hr. per sq. ft. per deg. F.; θ = temperature difference, deg. F.; K = thermal conductivity of the liquid, B.t.u. per hr. per sq. ft. per deg. F. per 1 in.; C = specific heat; S = specific gravity; Z = viscosity, centipoises. This formula is of value mainly in indicating effects of temperature difference and fluid properties, as the constant term will vary with the type of heating surface. In boilers and evaporators values of h usually range from 400 to 2000 for water and from 50 to 500 for refrigerants and organic liquids, the higher values being associated with high heating rates.

5. CONDENSATION OF VAPORS

PURE VAPORS.—The following formulas, due to Nusselt, are in good agreement with the test data for the condensation of pure saturated vapors.

For a Vertical Surface of height L ft., $h = 16.3(r\rho^2K^3/ZL\theta)^{1/4}$; [26]

For a Horizontal Pipe of diameter d in., $h = 23.6(r\rho^2K^3/Zd\theta)^{1/4}$, [27]

where h = heat transfer coefficient, B.t.u. per hr. per sq. ft. per deg. F.; r = latent heat of condensation, B.t.u. per lb.; ρ = density of liquid, lb. per cu. ft.; K = thermal conductivity of the liquid, B.t.u. per hr. per sq. ft. per deg. F. per 1 in.; Z = viscosity of liquid, centipoises; θ = temperature difference between the vapor and the cooling surface, deg. F. For condensing superheated vapors, h has practically the same value if θ is taken as the difference between saturation temperature and the surface temperature. In the case of a bundle of horizontal pipes, d should be taken as the sum of the diameters in a vertical row.

EFFECT OF NON-CONDENSING GASES.—The presence of even a small amount of non-condensing gas, as air, in a vapor may have a marked effect in reducing the heat transfer coefficient. Othmer (*Ind. and Engg. Chem.*, vol. xxi, 1929, p. 576) found that the presence of 1.07% of air in steam reduced the coefficient to 55% of its value for pure steam. Under the conditions of good engineering practice, the average value of h is about 2000 for steam and 1000 for ammonia, bearing in mind that these are film coefficients on the vapor side only. Values of h for other vapors usually are lower; McAdams and Frost (*Ind. and Engg. Chem.*, vol. xiv, 1924, p. 13) obtained coefficients of about 300 for carbon tetrachloride and 350 for benzine.

6. HEAT TRANSFER APPLICATIONS

CONDUCTANCE. THERMAL RESISTANCE.—It has been shown above that it is possible to express heat transfer by any of the fundamental processes in terms of an individual heat transfer coefficient h, in B.t.u. per hr. per sq. ft. per deg. F. The heat flow then is given by expression $q = hA(t_1 - t_2)$ B.t.u. per hr., where A = area, sq. ft., and t_1 and t_2 = terminal temperatures, deg. F. For the purpose of combining several processes acting in parallel or in series, to obtain the total or overall heat transfer, it is convenient to define the *Conductance*, C, of each process as the product of the coefficient

and the corresponding area; $C = hA$. The reciprocal of the conductance is the *Thermal Resistance*, $R = 1/C$. Thermal conductance and resistance then may be combined or resolved in a manner analogous to the treatment of the corresponding terms in electrical circuits, as shown below.

PARALLEL PROCESSES.—For several processes acting through parallel paths, the conductances are additive. $C = C_1 + C_2 + C_3$, etc. If two or more processes act in conjunction through the same path, or dissipate heat from the same surface, the individual coefficients may be added directly, $h = h_1 + h_2$, etc.

For example, in computing heat loss from a surface in still air, radiation coefficient, h_r should be added to convection coefficient, h_c. In many practical cases, such as non-metallic surfaces or oxidized steel pipes, the emissivity e is about 0.9. Under these conditions the value of the combined coefficient $(h_c + h_r)$ may be obtained from Table 12. The coefficients for pipes are based on tests by R. H. Heilman (*Trans.* A.S.M.E., vol. xliv, 1922, p. 299), who obtained practically the same results with horizontal bare steel pipes as with canvas covered surfaces. Very similar figures have been reported by other observers, for vertical as well as horizontal pipes. Under the conditions of forced convection at moderate temperatures the radiation component usually is so small that the total heat transfer may be obtained directly from Fig. 1.

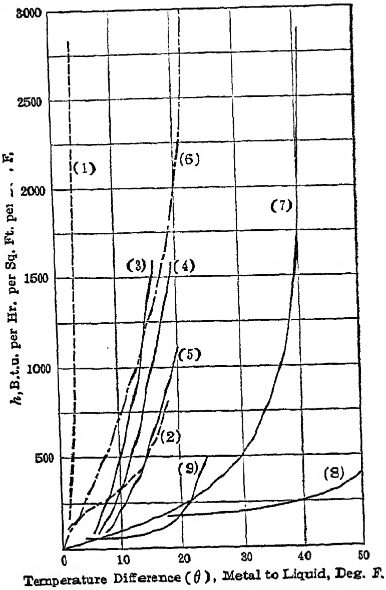


FIG. 3. Heat Transfer Coefficients, Metal to Boiling Liquids

(1), (2), Jakob and Fritz, water, 212° F., horizontal copper plate, (1) grooved, (2) polished chromium surface. (3), (4), (5), Linden and Montillon, water, (3) 212° F., (4) 195° F., (5) 180° F., inside inclined copper pipe 1 in. × 4 ft. (6), (7), (8), Dunn and Vincent, (6) water, 212° F., (7) methyl alcohol, 149° F., (8) toluene, 232° F., horizontal copper plate. (9), Young, methyl formate or ethyl ether, 60° F., vertical copper plate

SERIES OR OVERALL HEAT TRANSFER.—When heat flows through several paths in series, as from one fluid to another across a dividing wall, the total or overall resistance is equal to the sum of the individual resistances.

$$R = R_1 + R_2 + R_3 =$$

The heat flow in B.t.u. per hr. is then $q = (t_1 - t_2)/R$. In series flow the overall heat transfer coefficient usually is designated by the symbol U, as B.t.u. per hr. per sq. ft. per deg. overall temperature difference. If the cross-section of the heat path for all processes is the same, or if an average area is used in the case of thin-walled pipes, the reciprocals of the individual coefficients are additive. $1/U = 1/h_1 + 1/h_2 + 1/h_3$, etc. The value of h for a pipe wall is equal to the conductivity of the metal divided by the wall thickness in inches. In many practical cases, as in the exchange of heat between two fluids through a thin metal wall, the conduction coefficient of the metal is so large that its reciprocal may be left out of the equation for series flow, so that $1/U = 1/h_1 + 1/h_2$.

Table 12.—Combined Heat Transfer Coefficients $(h_c + h_r)$ for Surfaces in Still Air at Room Temperatures

(In B.t.u. per hr. per sq. ft. per deg. F.)

Pipe Size, in.	Outside Diam., in.	Temperature Difference, deg. F.								
		25	50	100	200	300	400	500	600	700
1/2	0.840	2.40	2.48	2.70	3.21	3.84	4.57	5.41	6.31	7.20
1	1.315	2.21	2.29	2.50	3.00	3.60	4.34	5.16	6.05	7.00
2	2.375	2.04	2.15	2.35	2.85	3.45	4.16	4.98	5.84	6.75
4	4.50	1.90	2.00	2.19	2.66	3.24	3.95	4.77	5.64	6.55
8	8.625	1.80	1.91	2.09	2.56	3.14	3.86	4.66	5.53	6.40
Plane Surface		1.69	1.75	2.00	2.51	3.06	3.73	4.42	5.28	6.22

In any case, the overall coefficient is always less than the smallest individual coefficient. For example, if the coefficient for air flowing over a copper tube is $h_1 = 10$ and the coefficient for water circulating through the tube is $h_2 = 500$, then $1/U = 1/10 + 1/500$, or $U = 9.8$. If the water velocity is increased until $h_2 = 1000$, the overall coefficient becomes $U = 9.9$, which shows that under such conditions the value of U is practically equal to the air film coefficient (Fig. 1). On the other hand, in the case of heat transfer from steam to water in condenser tubes or feedwater heaters, the water velocity is the limiting factor, as is shown in Table 13, based on Orrok's tests (*Trans.* A.S.M.E., vol. xxxii, 1910, p. 1139).

Table 13.—Heat Transfer from Steam to Water in Condenser Tubes

1-in. O.D., 18 B.W.G. Admiralty tubes; steam temperature, 130° F.; water temperature 55° F.

	Water Velocity, ft. per second					
	0.5	1	2	4	6	8
	Values of U, B.t.u. per hr. per sq. ft. per deg. F.					
New, clean tubes........	145	242	394	614	774	900
Old, dirty tubes.........	135	215	327	465	550	614

Many tests on surface condensers and heaters, reported in the literature, show values of U intermediate between the figures given above for old and new tubes.

OVERALL HEAT TRANSMISSION.—When heat flows from a medium at a constant temperature, t_1, to another medium at a constant temperature, t_2, deg. F., through a path having a constant or mean area, A, sq. ft., the rate of heat transfer in B.t.u. per hr. is $q = UA\theta$, where θ is the temperature difference $(t_1 - t_2)$. The value of the overall coefficient U may be computed from the individual coefficients, h_1, h_2, etc., as shown above. In heat exchangers where the temperature difference decreases from θ_a at one end of the apparatus to θ_b at the other end, an arithmetic mean $(\theta_a + \theta_b)/2$, may be used with little error if $\theta_a/\theta_b < 2$. Otherwise the logarithmic mean difference should be used as follows,

$$/2.3 \log_{10} (\theta_a/\theta_b) \quad . \quad . \quad . \quad . \quad [28]$$

This applies to any of the following cases: 1. A surface or fluid at a constant temperature exchanging heat with a flowing medium; 2. Two media flowing in opposite directions (counter flow); 3. Two media flowing in the same direction (parallel flow), provided the specific heats and heat transfer applications are sensibly constant along the surface. Methods for computing the mean temperature difference under other other conditions are given in McAdams' Heat Transmission.

EVAPORATORS AND EVAPORATION

By W. L. Badger

References.—Chemical Engineers' Handbook, Perry; McGraw-Hill, 1934; Elements of Chemical Engineering, Badger and McCabe; McGraw-Hill, 1931; Heat Transfer and Evaporation, Badger; Chemical Catalog Company, 1926; Evaporation, Webre and Robinson, Chemical Catalog Company, 1926; Evaporating, Condensing and Cooling Apparatus, Hausbrand, Wright and Heastie; Van Nostrand, 1933.

Evaporation may be carried out by the use of any suitable source of heat, but certain methods, because of their convenience and economy, are most practical. Evaporation by solar heat is practical in very few locations in the U. S., and is confined entirely to the manufacture of common salt and similar compounds. For evaporation in sprays and cooling towers, see Section 9. Evaporation by direct fire is a province of the designer of steam boilers. See Section 6. Where liquids are evaporated other than for making steam or by waste heat, the apparatus never has been standardized. Such apparatus may be designed on the basis of information in pp. 3-38 to 3-45. In most operations in practice where a solution is to be concentrated or water is to be distilled, some type of steam-heated apparatus almost invariably is used. Many types of construction are found, but certain constructions are so common as to be almost standard.

1. EVAPORATOR CONSTRUCTION

Steam-heated evaporators may be classified in three general types: Those with horizontal tubes, with inclined tubes, and with vertical tubes. The latter may be subdivided into evaporators with natural circulation and evaporators with forced circulation. Fig. 1 gives conventionalized illustrations of these types.

THE HORIZONTAL TUBE EVAPORATOR, Fig. 1-A, consists of a vertical cylindrical body, that may be from 3 to 15 ft. diam. and from 4 to 15 ft. high. Two rectangular

steam chests in the lower section contain tube sheets, between which the tubes are fastened. The tubes, usually 7/8 in. diam. in smaller evaporators, and 1 1/4 in. in larger ones, generally are fastened to the tube sheets by rubber packing rings and packing plates held down by studs. Connection for inlet and outlet of liquor may be at any convenient point. The vapor offtake always is in the center of the top. It may or may not be provided with internal or external foam catchers or entrainment separators. Steam for heating enters one steam chest, and condensate and non-condensible gases are removed from the other. This evaporator is used widely, although it is not the commonest type. It is primarily suitable for non-viscous liquids that do not deposit salt or scale during evaporation. It gradually is being replaced by other types.

THE STANDARD VERTICAL TUBE EVAPORATOR, Fig. 1-B, consists of a vertical cylindrical shell which may be closed with a flat bottom, a deep dish or a conical bottom. Vertical tubes are fastened between two horizontal tube sheets extending across the entire body near the bottom. The tubes always are held in place by rolling, and are from 1 1/2 to 4 in. diam. (2 and 2 1/2 in. are most common) and from 30 in. to 6 ft. long (5 ft. most common). A central downtake well generally is provided, of cross-sectional area of about 75% of the combined cross-sectional area of the tubes. The liquid is inside the tubes. Steam enters the space outside the tubes and between the tube sheets through suitable connections. Condensate is taken off the bottom tube sheet, and non-condensed gases usually are removed from the top tube sheet at a point opposite the steam inlet. During boiling, normal circulation is up through the tubes and back through the downtake. Connections for admitting feed and discharging thick liquor may be made where desired.

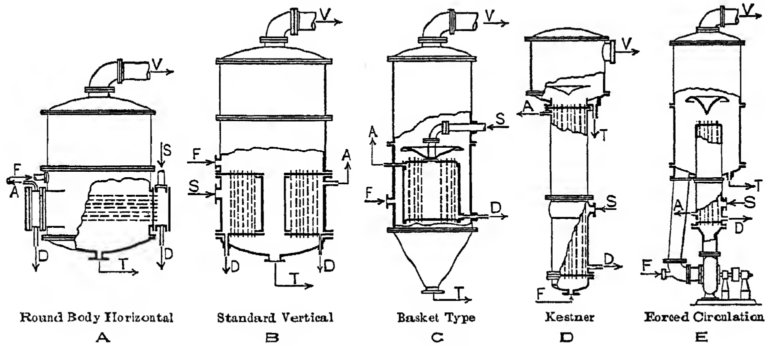


Fig. 1. Types of Evaporators

This evaporator is probably at present (1935) the most widely used of all types. It can be adapted to more different purposes and can be built in larger units than the horizontal tube evaporator. It is the only practical type for liquids that deposit salt or scale during boiling. It can carry liquids to relatively high viscosities. The construction is so common that it often is referred to as the standard evaporator.

THE BASKET-TYPE EVAPORATOR, Fig. 1-C, is a structural variation of the standard type. The downtake, instead of being a central well, is an annular ring; the heating element comprises a cylindrical drum or basket with tubes of the usual size rolled into the tube sheets. Steam generally is admitted to the top of the basket at the center. This type is quite common in the U. S. and is suitable for the same uses as the standard evaporator.

There may be many other minor variations of the standard vertical tube evaporator. The downtake may be at one side; it may consist of a number of small downtake tubes; or it may consist of downtake passages entirely external to the main body. The constructions, 1-B and 1-C, however, are the only important ones.

LONG-TUBE NATURAL CIRCULATION VERTICAL EVAPORATORS, Fig. 1-D, have tubes 15 to 20 ft. long. The liquor space is reduced to a very small chamber below the bottom tube sheet. There is a relatively small vapor head above the top tube sheet. In operation, the normal liquor level is only about 3 ft. above the bottom tube sheet, and the liquor being evaporated is carried to the top of the evaporator as a film along the tube wall. Immediately above the tubes, some type of entrainment separator deflects

the liquid into a channel, whence it is drawn off. In this type the liquor passes through the evaporator but once.

This type has been very little used in the U. S., but has been generally known abroad as the Kestner evaporator. Interest in this type is rapidly growing (1935) and it is easily possible that it may outstrip shorter tube types. It is not suitable for liquids that deposit salt or scale.

FORCED CIRCULATION EVAPORATORS, Fig. 1-E, consist of a bundle of tubes, usually $^7/_8$ in. diam. by 8 ft. long. Steam is introduced around these tubes near the bottom of the heating element, passes up behind an inner cylindrical baffle, and flows downward along the tubes. Non-condensed gases and condensate are taken off near the bottom tube sheet. A portion of the heating element projects into the main body of the evaporator. Liquid is pumped through the tubes at velocities of 6 to 12 ft. per second, usually by a low head centrifugal pump. The liquid issues from the tubes as a mixture of vapor and spray, striking a parabolic deflector that throws the liquid down into the lower part of the evaporator body, whence it returns to the circulating pump. Concentrated liquor usually is taken off from the body, and feed is introduced into the pump suction.

This evaporator is suitable for the concentration of liquids to extremely high viscosities, for liquids that tend to deposit salt or scale, for foamy liquids or cases where entrainment must be reduced to a minimum, or for cases where high-priced metals must be used for the heating surface. This evaporator, with nickel tubes, has become the standard in the U. S. for the evaporation of caustic soda.

Inclined Tube Evaporators have been popular at one time in both Europe and the U. S., but are not now offered by many of the well-known makers of evaporators. The long-tube natural-circulation type has all the advantages claimed for the inclined tubes, and requires less floor space.

Miscellaneous Types.—Many evaporators with special types of heating surfaces have been recommended at one time or another, especially for the preparation of water for boiler feed, in both stationary and marine plants. These usually have heating surfaces in the form of flat coils, hairpin tubes, or vertical coils. One of the better known types is the Griscom-Russell. It uses a number of vertical coils between two cast-iron headers, attached to a door that fits the side of a vertical cylindrical body. One advantage claimed for such evaporators, and for the Griscom-Russell evaporator in particular, is that when the surfaces become coated with hard scale, cold water may be turned into the coils and the resultant contraction cracks off the scale. These evaporators have been used only in power plant work.

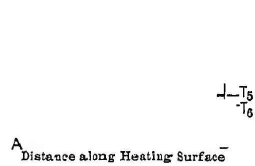


Fig. 2. Thermal Conditions in Evaporators

The purpose for which evaporators may be used and the types of liquids to be evaporated are so varied that it is impossible to make definite statements as to the principal field of usefulness of each type. In many industries the type of evaporator used is dictated by custom, rather than by sound engineering principles.

2. HEAT TRANSFER IN EVAPORATORS

The discussions of Heat Transfer (see p. 3-26) indicate that the only logical method of studying heat transfer coefficients would be to separate them into their separate film coefficients. Practically no work of this type has yet been done on boiling liquids.

Fig. 2 shows diagrammatically the thermal conditions in the evaporator in the most general cases. Temperatures are plotted on the Y axis and distance along the heating surface is plotted along the X axis. Steam enters somewhat superheated at temperature T_1 and is first cooled to saturation temperature T_2. The heating surfaces should be so proportioned that there is no appreciable pressure drop between steam inlet and condensate outlet and, therefore, condensation takes place throughout the heating surface at T_2. Before leaving the evaporator the condensate may be cooled to T_3. It is obvious that in practice these operations do not take place in three distinct stages, but are more or less simultaneous. For discussion, however, it is convenient to separate them as shown.

TEMPERATURE DIFFERENCES.—The liquid enters the evaporator somewhat below its boiling point at T_4. It then is heated to boiling point, T_5, at which temperature it evaporates throughout the body. In practically every case T_5 is higher than the boiling point of pure water, T_6, under the pressure existing in the body of the evaporator.

The true temperature difference between steam and liquid would be the area $T_1T_2T_3T_5T_4T_1$ divided by distance AB. In most cases in practice, however, the superheat temperature, the condensate temperature, and the feed temperature are neglected,

and the working temperature drop is considered to be the difference between the temperature of the steam and the temperature of the liquid.

It is much easier to measure the pressure in the vapor space of the evaporator than it is to measure temperature of the boiling liquid. It is customary, therefore, to calculate temperature T_6 from the pressure of the vapor space by means of steam tables, and the temperature difference $(T_2 - T_6)$ is termed the *apparent temperature difference.* Heat transfer coefficients based on this apparent temperature difference are called *apparent heat transfer coefficients.*

If elevation in boiling point of the solution in question is known, or can be determined, temperature T_5 is obtained by adding the known elevation in boiling points to the apparent boiling point T_6. When temperature T_5 thus obtained is subtracted from T_2, this gives a *temperature drop corrected for boiling point elevation*; the heat-transfer coefficients based on this temperature drop are called *coefficients corrected for elevation in boiling points.*

The boiling point T_6 exists only at the surface of the liquid. The lower layers of liquid are under a pressure greater than that of the vapor space and, therefore, must have a higher boiling point than the surface layers. Consequently, the true mean working temperature drop is less than either of the temperature drops described above. A correction for this effect of hydrostatic head, although important, cannot be made with certainty (due to rate of circulation of the liquid). It, therefore, ordinarily is not included in evaporator calculations.

Table 1.—Boiling Points of Sodium Chloride Solutions

Pressure, mm.	Grams NaCl per 100 grams of Water										
	0	4	8	12	16	20	24	28	32	36	Sat.
	Boiling Point, deg. F.										
760	212.0	213.1	214.2	215.6	217.0	218.6	220.3	222.1	223.9	225.7	227.7
540	195.3	196.4	197.6	198.9	200.2	201.6	203.2	205.0	206.8	208.6	210.2
380	179.1	180.1	181.3	182.5	183.8	185.2	186.8	188.4	190.2	192.0	193.6
240	159.3	160.2	161.2	162.3	163.4	164.8	166.3	167.9	169.7	171.5	172.4
160	142.9	143.8	144.8	145.8	146.8	148.2	149.6	151.1	152.8	154.6	155.5
100	124.8	125.6	126.5	127.4	128.5	129.7	131.1	132.8	134.4	136.1	136.9

Table 2.—Boiling Points of Calcium Chloride Solutions

Pressure, mm.	Grams $CaCl_2$ per 100 Grams of Water									
	0	10	30	50	70	90	110	130	150	Sat.
	Boiling Point, deg. F.									
760	212.	214.3	222.6	235.2	249.1	261.1	272.1	282.4	291.6	355.1
560	197.1	199.2	207.3	219.7	233.4	245.3	256.1	266.2	275.2	301.6
360	176.6	178.7	186.4	198.7	212.0	223.9	234.5	244.2	252.7	269.6
240	159.3	161.1	168.4	180.0	193.6	205.3	215.8	225.1	233.4	244.6
160	142.9	144.7	152.0	163.6	176.5	188.2	198.5	207.7	215.4	223.2
100	124.8	126.6	133.5	145.0	158.0	169.5	179.6	188.6	195.8	208.2

Table 3.—Boiling Points of Glycerol-water Solutions

Pressure, mm.	Percentage of Water in Solution										
	100	90	80	70	60	50	40	30	20	10	4.36
	Boiling Point, deg. F.										
760.	212	213.3	214.9	217.2	220.1	224.1	229.3	237.2	250.7	283.6	348.4
525.80	194	195.1	196.7	199.6	201.6	205.4	210.6	218.3	230.5	262.0	322.0
355.10	176	176.9	178.5	180.7	183.2	186.7	191.9	199.0	210.7	240.8	295.7
233.53	158	158.7	160.2	162.3	164.7	168.1	173.2	180.0	190.9	219.2	269.8
149.19	140	140.5	141.8	144.0	146.3	149.7	154.6	160.7	171.1	197.6	243.7
92.30	122	122.3	123.6	125.8	128.1	131.4	135.7	141.8	151.2	176.2	217.6

Table 4.—Boiling Points of Glycerol-water Solutions Saturated with Sodium Chloride

Pressure, mm.	Boiling Point of Water, deg. F.	Percentage of Water in Solution										
		100	90	80	70	60	50	40	30	20	10	4.36
		Boiling Point, deg. F.										
760.	212	277.7	228.4	229.6	232.0	234.7	237.6	242.2	249.6	264.2	301.6	354.7
525.80	194	208.8	209.3	210.6	213.0	215.2	218.3	222.8	229.3	243.7	278.8	328.5
355.10	176	189.9	190.2	191.7	193.8	196.0	199.0	203.0	209.7	222.8	255.9	301.8
233.53	158	171.1	171.1	172.6	174.7	177.1	180.0	183.6	189.9	203.0	232.9	275.4
149.19	140	152.2	151.7	153.7	155.8	158.0	160.3	164.1	170.1	181.6	210.0	249.0
92.30	122	133.7	133.0	134.6	136.8	138.9	141.1	144.7	150.1	161.1	189.0	222.4

It should be noted that in no case can the elevation in boiling points of a solution be calculated from known rules, except for solutions so dilute that they are of no practical importance. Further, the elevation in boiling point changes with changes in concentration and in pressure, and, therefore, must be determined by experiment. Many data on boiling points of pure substances are given in reference books, especially International Critical Tables, but for many commercial solutions these elevations are incorrect. Tables 1 to 4 represent the results of experiments by the author.

THE OVERALL HEAT TRANSFER COEFFICIENT in an evaporator is obviously the resultant of the steam film coefficient, the resistance of the metal wall, together with any scale it may carry, and the resistance of the liquid film. Since in most cases the liquid film is in natural or free convection, it practically is impossible to calculate overall heat transfer coefficients for any except forced circulation evaporators. In these the liquids inside the tubes may be considered as non-boiling through a considerable part of their length.

While there are many data in the literature on heat transfer coefficients in evaporators, they cover such a small portion of the entire field that the average engineer cannot predict heat transfer coefficients. To give an idea of variations that may be expected, a set of such determinations is reproduced in Fig. 3. This represents overall apparent coefficients between steam and boiling distilled water, in a vertical-tube basket-type evaporator with tubes 2 in. diam. by 30 in. long. The general shape of the curves is similar for other tube proportions. Note that the heat transfer coefficient increases with increasing temperature drop, due to more vigorous boiling at the higher temperature drops, with correspondingly more rapid circulation. Also note that for a given temperature drop the heat transfer coefficient increases as the boiling point increases. This is due largely to a decreased viscosity of water at higher temperatures, with consequent increase in rate of circulation. Changes of the same, or even greater, order of magnitude can be caused by a change in the type of liquid, depth of liquid, diameter of tubes, length of tubes, shape and size of the body, and many other factors. It is obvious that it is impossible to present in any summary a definite statement as to what heat transfer coefficient may be expected in a given case. The author has tests of evaporators showing overall heat transfer coefficients ranging from 4000 to 2 B.t.u. per sq. ft. per hr. per deg. F. In practice, unless data are available from plant tests on evaporators of the same type and size as the one under consideration, and operating under the same conditions on the same liquor, it is necessary to depend on the knowledge of the companies manufacturing commercial evaporators. In general, with ordinary non-viscous, non-scaling liquids, the horizontal-tube or the standard vertical-tube evaporator will have heat transfer coefficients between 200 and 500 B.t.u. per sq. ft. per hr. per deg. F. The long-tube natural-circulation evaporator and the forced-circulation evaporators may reach 1000 to 1200 B.t.u. in the same units.

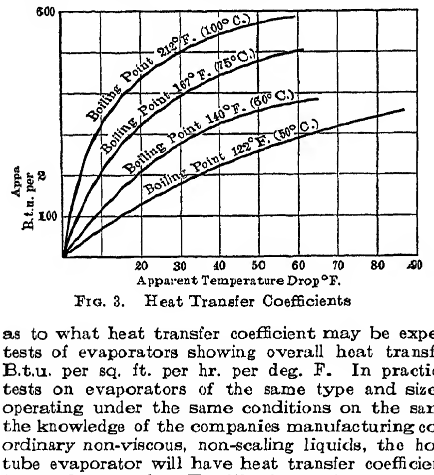


FIG. 3. Heat Transfer Coefficients

3. MULTIPLE-EFFECT EVAPORATION

A multiple-effect evaporator is a series of evaporators so connected that the vapor from one body is used as the heating steam in the next. To provide a working temperature drop in each body or effect, the pressure in the vapor space of each body must be lower than the preceding one. The individual bodies of a multiple-effect evaporator are similar in all respects to the construction of single-effect evaporators.

HEAT RELATIONS.—In Fig. 4, imagine each body to be fed with liquid heated to the boiling point for the particular body to which it is fed. Let the liquid being evaporated be either pure water or a dilute solution whose thermal properties are not appreciably different from those of pure water. Consider radiation losses to be negligible. Then if W_0 pounds of steam at pressure P_0 are fed to the heating surface of the first effect, and if condensate be assumed to leave at T_0, the saturation temperature corresponding to the pressure P_0, then W_0L_0 B.t.u. are delivered to the heating surface. L_0 = latent heat of steam at pressure P_0. The liquid is at pressure P_1 and has a boiling point of T_1. T_1

will not be greatly different from T_0 and, therefore, L_1, the latent heat of the liquid in the first effect, will be nearly the same as L_0. W_0 pounds of steam entering the first effect will evaporate W_1 pounds of water in the first effect, and W_1 and W_0 will be nearly equal.

When the vapor produced by the boiling liquid in the first effect enters the second effect it will condense at the pressure P_1 and will, therefore, have latent heat L_1 very nearly the same as L_0. The liquid in the second effect will have a latent heat of L_2, not greatly different from L_1. Therefore, W_2 also will be nearly equal to W_0. The same line of reasoning may be continued, from which it may be concluded that in an N-effect evaporator, one pound of steam will evaporate N pounds of water. In practice, this statement must be modified to take into consideration such details as heat used for heating of feed, changes in latent heats when there are very large temperature drops across the evaporator, heat losses by radiation, and similar effects. The principle remains unchanged, that increasing the number of effects increases the *economy* of operation.

RELATIVE CAPACITY OF SINGLE- AND MULTIPLE-EFFECT EVAPORATORS.—The pressure and temperature of steam for operating the evaporator, and the pressure and temperature that may be produced in a condenser, usually are fixed by conditions external to the evaporator. If T_0 is the saturation temperature of the steam available, and T_N is the saturation temperature corresponding to the pressure in the condenser, then the total temperature drop available for the operation of the evaporator is $T_0 - T_N$. If a single-effect evaporator is used, of a heating surface of A square feet and an overall coefficient U, the heat transmitted by this evaporator will be $UA\ (T_0 - T_N)$.

Suppose that a double-effect evaporator be operated with steam to the first effect at T_0 and that the pressure in the vapor space of the second effect corresponds to temperature T_N. The boiling point in the first effect (and consequently the temperature of the heating steam in the second effect) are represented by T_1. If both evaporators have the same surface A per effect, and if their heat transfer coefficients are U_2 and U_1, respectively, then the heat transferred through the first effect will be $U_1A\ (T_0 - T_1)$; the heat transferred through the second effect will be $U_2A\ (T_1 - T_N)$. If heating of feed and losses by radiation, etc., are neglected, it follows that $U_1A\ (T_0 - T_1)$ must be approximately equal to $U_2A\ (T_0 - T_1)$. That is, the evaporator will come to equilibrium with T_1 at such a value that the temperature drops in the two effects will be approximately inversely proportional to the heat transfer coefficients. Temperature T_1 cannot be set arbitrarily, or controlled mechanically, as it is solely the result of thermal equilibrium between the effects. If any operating condition changes so that U_2 decreases, then steam will not be condensed in the heating surface of the second effect as fast as it is generated in the first effect. As a result, pressure will build up in the first effect with consequent rise in temperature T_1. This will decrease the temperature drop across the first effect and increase it across the second effect until the evaporator has attained an equilibrium corresponding to the new conditions.

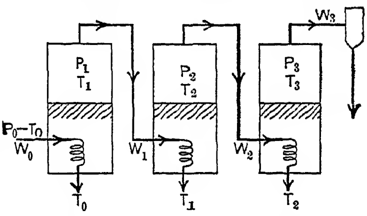


FIG. 4. Diagram of Multiple Effect

The above reasoning may be extended to any number of effects and the conclusion reached that, in a multiple-effect evaporator, the temperature distribution between effects represents a normal and automatically-attained thermal equilibrium. This cannot be altered mechanically, and will be such that the temperature drops across the various effects will be approximately inversely proportional to the heat transfer coefficients in those effects.

It will also appear from the above reasoning that, since $(T_0 - T_1)$ is only a part of the single-effect temperature drop, $(T_0 - T_N)$, A square feet of heating surface in the first effect of a double-effect evaporator will transmit much less heat than the same number of square feet in a single-effect evaporator working between the same terminal temperatures T_0 and T_N. Further, since the heat transmitted in the first effect is approximately equal to the heat transmitted in the second effect, it follows that even if the heat transfer coefficients of a double-effect evaporator were equal to the heat transfer coefficient in a single-effect evaporator, there would be needed A square feet in *each* effect of the double-effect evaporator to transmit the same total quantity of heat with the same terminal temperature drop as would be transferred by A square feet in a single-effect evaporator. This same reasoning can be continued to any number of effects, and leads to the conclusion that a multiple-effect evaporator, to have a given total evaporation, must have as much heating surface in *each* effect as the heating surface of a single-effect

evaporator operating under the same terminal temperatures. Further, Fig. 3 shows that as temperature drops decrease, heat transfer coefficients decrease. Consequently, in practice it usually is necessary to have more surface per effect in a multiple-effect evaporator than would be required in a single-effect evaporator to do the same work.

The first cost of an evaporator, and consequently fixed charges and maintenance, thus will increase in proportion to the number of effects. Another statement of the same conclusion is the capacity of the evaporator per square foot decreases in proportion to the number of effects. Consequently, although passing from single effect to multiple effect improves steam *economy*, it decreases *capacity*, and increases fixed charges. The economic number of effects for any particular case is obviously that number which shows the minimum total cost. The total cost is the result of adding steam costs and condenser water costs (which decrease with number of effects) to fixed charges and maintenance (which increase with number of effects). Such total cost curves usually show a marked minimum.

In many cases in present practice single- and double-effect evaporators are used. Triple- and quadruple-effects are very common and there are a few quintuple-effects. Evaporators with six effects or more are extremely rare.

4. CALCULATIONS FOR MULTIPLE-EFFECT EVAPORATORS

The most important results to be obtained from the preliminary calculations for multiple effect evaporators are: 1. Quantity of steam required; 2. Heating surface required; 3. Temperature in the various effects; 4. Water consumption of the condenser. As heat transfer coefficients must be determined by experience, and as they are not available to the average engineer, result (2) will be only approximate. The other results, however, will be quite accurate. These figures, especially item (1), are of the greatest importance in making preliminary decisions as to the number of effects, the arrangement of the evaporator, and its influence on the heat balance of the plant.

In Fig. 5, thin liquor is fed to the first effect, the liquor is passed from effect to effect in the direction of decreasing pressures and thick liquor is removed from the last effect. This is the commonest method of feeding evaporators.

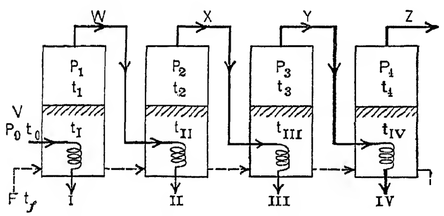


FIG. 5. Multiple-effect Calculations

In most evaporator installations, all the effects are of the same size and construction. This usually is a requisite condition in all such calculations. In order to make the calculations, approximate values of the heat transfer coefficients in different effects must be available. The most obvious method of attack would be to write heat balance equations across each effect. This, however, would involve the temperature in each effect, but, as heretofore noted, temperatures in the various effects are the results of the evaporator coming to an equilibrium determined by the relation between the heat transfer coefficients. This prevents writing a set of equations that can be solved directly, so that trial and error must be used.

EVAPORATOR CALCULATIONS.—So many evaporator arrangements and so many complicating conditions are possible that no general formulation can be made. One case will be presented with certain simplifying assumptions, and the method followed through for this case. It will be necessary to develop similar equations by methods that should be obvious for whatever arrangement may occur in practice.

In practically all cases in practice the liquid being evaporated has a higher boiling point than pure water under the same conditions. Hence the vapor leaving the liquid will be superheated. The amount of superheat contained in the vapor under ordinary circumstances will be so small a part of the total heat available from the steam that its transfer is accomplished in a very small fraction of the apparatus, and most of the heating surface will be transmitting heat from saturated steam. Therefore, it is usual to disregard the effect of superheat on the temperature drop in evaporators, but not necessarily to disregard it as it may affect heat balances.

Assumptions.—1. Condensate leaves the heating surface at the saturation temperature of the steam. 2. Radiation is negligible. 3. Superheat in the vapor does not affect the temperature difference in the next effect. 4. Secondary thermal effects such as heat

of concentration, are negligible. 5. No appreciable amount of solids separate from the liquid during evaporation. 6. Coefficients of heat transfer have been corrected for elevation in boiling point. 7. Boiling point elevations are known for the liquor in question at all concentrations and pressures. 8. Specific heat of the solution to be evaporated is 1.00 at all concentrations.

Notation.—Let I, II, III, IV = first, second, third and fourth effects respectively; F = lb. thin liquor fed per hr.; E = total evaporation, lb. per hr.; V = lb. steam used per hr.; W, X, Y, Z = evaporation in I, II, III and IV, respectively, lb.; A_1, A_2, A_3, A_4 = heating surface in I, II, III and IV, respectively, sq. ft.; t_0 = saturation temperature of steam to I, deg. F.; t_{I}, t_{II}, t_{III}, t_{IV}, = boiling point of the solution in the several effects, deg. F.; t_1, t_2, t_3, t_4 = saturation temperatures of vapor from the several effects, deg. F.; L_0, L_1, L_2, L_3, L_4 = latent heat of vaporization of steam from the solution at t_0, t_1, t_2, t_3, t_4, B.t.u.; h_0, h_1, h_2, h_3, h_4 = heat present as superheat in vapor from the various effects, B.t.u.; U_1, U_2, U_3, U_4 = overall heat transfer coefficients in the various effects, corrected for elevation of boiling point. t_f = temperature of feed, deg. F.; Δt_1, Δt_2, Δt_3, Δt_4 = net working temperature drops in the various effects, deg. F.

Heat Balance Equations are as follows:

Across I. $$V(L_0 + h_0) = F(t_{\text{I}} - t_f) + W(L_1 + h_1) \qquad [1]$$

Across II. $$W(L_1 + h_1) + (F - W)(t_{\text{I}} - t_{\text{II}}) = X(L_2 + h_2) \qquad [2]$$

Across III. $$X(L_2 + h_2) + (F - W - X)(t_{\text{II}} - t_{\text{III}}) = Y(L_3 + h_3) \qquad [3]$$

Across IV. $$Y(L_3 + h_3) + (F - W - X - Y)(t_{\text{III}} - t_{\text{IV}}) = Z(L_4 + h_4) \qquad [4]$$

Material Balance.—In addition to the foregoing, a material balance equation can be 'tten:

$$E = W + X + Y + Z \qquad [5]$$

Heating Surface and Heat Transfer.—Next a set of equations connecting the heating surface and the heat transfer in each effect may be written:

$$A_1 = \frac{V(L_0 + h_0)}{U_1\,(t_0 - t_{\text{I}})} = \frac{W(L_1 + h_1) + F(t_{\text{I}} - t_f)}{U_1(t_0 - t_1)} \qquad [6]$$

$$\frac{+\ h_1)}{-\ t_{\text{II}})} \qquad [7]$$

$$A_3 = \frac{X(L_2 + h_2)}{U_3(t_2 - t_{\text{III}})} \qquad [8]$$

$$A_4 = \frac{Y(L_3 + h_3)}{U_4(t_3 - t_{\text{IV}})} \qquad [9]$$

The steps in the solution are as follows: 1. An approximation is made of the values of W, X, Y and Z; from these the approximate concentrations in each effect is determined. 2. From this approximate concentration the elevation in boiling point in each effect is determined. 3. The pressure and temperature of steam to the first effect and the vacuum to be maintained in the last effect usually are available as fixed conditions in the problem. If not, values for them are assumed, and from these, total available temperature drop for the whole evaporator is determined. 4. From the total available temperature drop, the sum of the elevations in boiling point is subtracted and the remaining net or effective temperature drop is divided between the effects, approximately inversely to the heat transfer coefficients. This, with the approximate elevation in boiling point, gives the temperature of liquid and the saturation temperature of vapor for each effect. 5. On the basis of these assumed temperatures the heat balance equations are solved for V, W, X, Y, and Z. 6. On the basis of these values for evaporation in each effect, the heat transfer equations are solved for the surface in each effect. 7. If these surfaces are not sensibly equal (assuming that it is a condition of the problem that all evaporator bodies must be of the same size), the temperatures in the various bodies are readjusted and steps 4, 5 and 6 are repeated. This process is continued until a set of temperatures is found that will give results satisfying the condition that all heating surfaces are equal. Values for V and Z thus obtained ordinarily are carried to within a few percent. If final values for W, X, Y and Z result in concentrations so different from those assumed in step 1 that the preliminary elevations in boiling point determined in step 2 are incorrect by more than one or two degrees then the calculations must be repeated on the basis of the new values for elevations of boiling point. Although the method sounds tedious, it is usually possible with a little experience to make the second or third trial yield results with an accuracy ample for any preliminary calculations. The following example will illustrate the application of the method.

Example.—A quadruple-effect evaporator is to be fed with 50,000 lb. per hr. of 5% sodium chloride solution at 150° F. Steam to the first effect will be at 35 lb. per sq. in., gage. Vacuum on

the last effect will be 26 in. referred to a 30-in. barometer. The solution is to be concentrated to 25% solids. Required the approximate heating surface, the steam used, and the heat above 32° F. going to the condenser.

Assumptions.—1. Feed will be forward. 2. Radiation losses will be negligible. 3. All specific heats may be taken as 1.00. 4. There will be no appreciable heat of concentration. 5. All condensate will leave steam chests at saturation temperature. 6. All effects are to have the same heating surface. 7. Superheat in vapor due to elevation in boiling point will not affect temperature drop in next effect. 8. Coefficients, in B.t.u. per hr. per sq. ft. per deg. F., corrected for elevation in boiling point, will be: First-effect, 375; second effect, 350; third effect, 300; fourth effect, 200. 9. Elevation in boiling point of salt solution may be taken from Table 1. 10. No salt or scale will separate. 11. Steam to the first effect is dry and saturated.

Solution.—*Step* 1. The total evaporation is determined as follows:

	H_2O	NaCl	Total
Feed	47,500 pounds	2,500 pounds	50,000 pounds
Product	7,500 "	2,500 "	10,000 "
Evaporation	40,000 pounds		40,000 pounds

Assume that the evaporation will be approximately equal in all effects. Then the concentrations will be as follows:

	H_2O, lb.	NaCl, lb.	Total, lb.	Concentration
Feed to I	47,500	2,500	50,000	5.00%
Evaporation in I	10,000		10,000	
Feed to II	37,500	2,500	40,000	6.25%
Evaporation in II	10,000		10,000	
Feed to III	27,500	2,500	30,000	8.33%
Evaporation in III	10,000		10,000	
Feed to IV	17,500	2,500	20,000	12.50%
Evaporation in IV	10,000		10,000	
Product	7,500	2,500	10,000	25.00%

Step 2. The elevations will be approximately as follows.

Effect	First	Second	Third	Fourth
Concentration	6.25%	8.33%	12.50%	25.00%
Elevation, deg. F.	1	2	3	10

Step 3. Steam to first effect—35 lb. gage = 281° F.
Vacuum on last effect—26 in. = 125° F.
Total temperature drop = 156° F.

Step 4. Total available temperature drop = 156° F.
Sum of all boiling point elevations = 16° F.
Net temperature drop = 140° F.

After several trials it is found that the desired temperature drops are $\Delta t_1 = 35°$; $\Delta t_2 = 25°$; $\Delta t_3 = 31°$; $\Delta t_4 = 49°$. The conditions will be

		Latent heat, B.t.u.	Superheat, B.t.u.	$L + h$
Steam to I	281° F.	923.5	......	923.5
Δt_1 =	35			
Boiling point in I =	246			
Elevation in I =	1			
t_1 =	245	948.6	0.5	949.1
Δt_2 =	25			
Boiling point in II =	220			
Elevation in II =	2			
t_2 =	218	966.4	1.0	967.4
Boiling point in III =	187			
Elevation in III =	3			
		987.4	1.5	988.9
Boiling point in IV	135			
Elevation in IV =	10			
t_4 =	125	1022.2	5.0	1027.2

Step 5. Substituting in Equations [2], [3], [4] and [5] gives

$$949.1W + (50{,}000 - W)(246 - 220) = 967.4$$
$$967.4 + (50{,}000 - W - X)(220 - 187) = 988.9Y$$
$$988.9Y + (50{,}000 - W - X - Y)(187 - 135) = 1027.2Z$$
$$W + X + Y + Z = 40{,}000$$

A simultaneous solution of these equations gives W = 8690 lb.; X = 9650 lb.; Y = 10,500 lb., Z = 11,160 lb.

Step 6. Substituting in Equations [6], [7], [8], and [9], gives $A_1 = 1014$ sq. ft.; $A_2 = 972$ sq. ft.; $A_3 = 1007$ sq. ft.; $A_4 = 998$ sq. ft.; or an average of 1000 sq. ft. per effect.

Since the values for the coefficients are only approximate, and since a closer correspondenc

32° F. in the vapor going to the condenser is $11{,}160 \times (1027.2 + 125 - 32)$ 12,389,800 B.t.u.

If any of the assumptions made on page 3-44 in connection with this problem are not justified, the resulting modification of the above equations and methods will be obvious.

5. EXTRA STEAM

The phrase "extra steam" has a special meaning in connection with evaporator flow sheets. It means vapor withdrawn from any body of an evaporator for use elsewhere in the plant.

In a plant having a complicated steam flow sheet, especially as regards the use of process steam, the possibilities of a multiple-effect evaporator as a producer of low-pressure steam rarely is given sufficient consideration. For instance, if vapor is taken from the second effect of a multiple-effect evaporator for use as process steam, providing that the temperature of the second effect is sufficiently high for the purpose desired, this vapor may be considered as steam that has already evaporated twice its weight of water and is therefore that much more economical than steam from the mains. The removal of such quantities of vapor from a multiple-effect evaporator alters the evaporator balance somewhat, and makes the evaporator, as an evaporator, somewhat less economical. However, the effect on the heat balance of the plant as a whole is always highly favorable. If large quantities are withdrawn from any one body in comparison to the amount generated in that body, the temperature distribution over the evaporator may be too much disturbed. It may then be more practical to put additional heating surface in the body from which such vapor is withdrawn and all bodies ahead of it. If the amount withdrawn is 25% or less of the total amount generated in any given body, it usually still will be possible to make all bodies alike without having abnormal operating conditions. The advantages of this method of operation are fully understood only in the beet sugar industry. They deserve much more consideration in other industries than they have had heretofore.

6. EVAPORATOR OPERATION

OPERATING TEMPERATURES.—The pressure of the steam to the first effect of an evaporator usually is fixed by conditions already existing in the plant. When a new plant is to be planned, that will use steam both for generating power and for evaporating, rather extensive calculations are necessary to determine the point at which prime movers should exhaust to evaporators. Evaporator practice was developed when power generally was obtained from small non-condensing reciprocating engines, and a general impression resulted that most evaporators should be given steam at from 5 to 25 lb. gage. This is by no means necessary, and evaporators are in operation with steam pressures on the first effect anywhere from 125 lb. gage down to 22 in. vaccuum.

The vacuum on the last effect is not so easily determined. When evaporators operated with steam at about 5 lb. gage, a vacuum of 26 in. was about all that ever was expected, and has become quite customary in evaporator practice. It is, however, a mistake to assume that because power plant engineering has developed devices capable of producing much higher vacuums economically, these devices should be applied to evaporators and the highest possible operating vacuum used.

A vacuum sometimes is carried on an evaporator for the express purpose of boiling a heat-sensitive liquid at the lowest possible temperature. The effect of boiling point on the properties of the liquid in question then determines the vacuum to be used, and outweighs all other considerations. In the great majority of cases, however, vacuum is used merely to give a larger temperature drop across the evaporator. It would at once seem that slightly increasing the vacuum would make a relatively large difference in the total available temperature drop, and that therefore the highest possible vacuum should be carried on the last effect. Fig. 3 shows, however, that everything else being equal, the lower the boiling point, the smaller will be the heat transfer coefficient. This is because the viscosity of all liquids changes rapidly with temperature. Lowering the boiling point increases the viscosity; increasing the viscosity slows the circulation; slowing the circula-

tion gives lower heat transfer coefficients. In a multiple-effect evaporator where the temperature drops per effect are not too large, the vacuum on the last effect can be increased to the point where a lowering of the boiling point in the last effect decreases the heat transfer coefficient more than it increases the temperature drop, and, therefore, decreases the capacity of the evaporator. For this reason, very high vacuums are not desirable on evaporators, and the elaborate devices used in power plants have no place in evaporator practice.

It is not possible to make any definite statement regarding the optimum vacuum; this varies with the liquid, with type of evaporator, with number of effects, with total available temperature drop and other factors. In general, however, there are probably few evaporator installations where a vacuum of over 28 in. is justified, and in many installations the optimum vacuum probably is around 26 in. A determination of the optimum vacuum obviously will consider, not only the capacity of the evaporator, but the cost of producing the higher vacuum.

In fixing maximum and minimum temperatures in multiple-effect evaporators, consideration also must be given to the fact that, in natural circulation evaporators, the smaller the temperature drop per effect the lower will be the average coefficients, and therefore the larger the evaporator must be for a given purpose. When a natural circulation evaporator is operated on too small temperature drops, circulation stops, the liquid lies quiet and merely simmers on the surface, and the capacity is negligible. This minimum operating temperature drop varies with the type of evaporator and the characteristics of the liquid; it is usually in the neighborhood of 10 to 15° F. This limits the number of effects that can be included in any particular total temperature drop or, conversely, fixes the minimum total temperature difference over a given number of effects.

METHODS OF FEEDING.—Forward Feed. The simplest method of feeding a multiple-effect evaporator is to feed thin liquor into the first effect, pass it progressively from effect to effect, and withdraw it from the last effect. This method usually is known as forward feed. It involves only one pump with its suction under vacuum, and most of the control is by simple throttle valves. If the feed is very cold, however, larger and larger amounts of steam are condensed in the first effect to heat the liquid from the feed temperature to boiling point. Since steam so condensed yields no vapor to be used in succeeding effects, lowering the temperature of the feed increases the steam consumption per pound of evaporation.

Backward feed consists in feeding thin liquor to the last or lowest pressure effect, and withdrawing thick liquor from the first or highest pressure effect. This requires more feed pumps and somewhat more difficult control. If the feed is colder than the boiling point in the last effect, it is heated by steam that has already evaporated several times its weight of water, and in passing from effect to effect it also is heated each time by steam that already has done some work. Consequently, for cold feed, backward feed is the most economical. However, if the feed is hotter than the boiling point in the last effect it flashes down to the temperature of the last effect, and this vapor goes directly to the condenser.

In general, with low feed temperatures, backward feed should be used, and forward feed with high feed temperatures. There is a wide range of intermediate feed temperatures where the proper arrangements can be determined only by making the complete evaporator calculations for both methods.

Parallel feed usually means feeding directly into each body of a multiple-effect evaporator. This is done only when there is no thick liquor to be removed, as in the case of salt evaporators.

Mixed feed covers all other arrangements. In the particular case of liquid to be finished at high densities and high viscosities, it may be desirable to finish this liquid in the first effect in order to have the viscosity as low as possible, and yet other circumstances may call for a forward feed. In such a case a feed order II, III, IV, I, may be the most satisfactory solution. Similar considerations lead to the development of other special cases.

FOAM AND ENTRAINMENT.—These two terms should be used in different senses. Foam is the formation of a solid mat of stable bubbles. Entrainment is the carrying of either single bubbles or solid drops of liquid with the vapor. Both may occasion further difficulty or serious loss in evaporator operation.

Foam.—The causes of foam are not well understood, but it is known that in liquids having a tendency to foam, the condition is greatly aggravated by the presence of finely divided suspended matter. Operation of evaporators with foamy liquids is extremely difficult. However, low liquid levels or jets of live steam directed against the surface of the blanket may break down the foam. For a true foam, entrainment separators are of no use. Foam also may be broken down by causing it to impinge at very high veloci-

ties against the baffle. This probably explains the success of the Yaryan type natural circulation evaporator and of the forced circulation evaporator in handling foamy liquids.

Entrainment is the net result of the velocity at which particles of liquid are shot up from the boiling surface and the velocity of the vapor into which they are projected. Entrainment can be eliminated by baffles in the vapor space, by high liquid levels in natural circulation evaporators, or by keeping vapor velocities low. Vapor that leaves the body of an evaporator carrying liquid in suspension usually can be made to drop its entrainment only by causing the liquid particles to impinge against a solid surface. This is effected by entrainment separators where the steam either is turned around sharp corners at high velocity or given a circular motion. In either case, the momentum of the liquid particles throws them out of the stream of steam and against a surface where they can collect. The result of the few tests that have been made in practice is that entrainment separators are more effective as the velocity of the steam through them is increased, though this increase in velocity can be carried to a point where it causes serious pressure drops.

SCALE.—The formation of scale on the heating surface depends, not on the presence of solids of low solubility, but on the presence of solids with an inverted solubility curve, that is, solids whose solubility decreases as temperature increases. This means that in the film of liquid against the heating surface, where the highest temperatures exist, the solubility of the substance in question is the least. It therefore precipitates next the heating surface rather than in the bulk of the solution. Calcium sulphate is probably the most universal scale producing substance. Sodium sulphate and sodium carbonate are important when present in such amounts that the solution becomes saturated with them. Many other substances may form scale under certain conditions.

It rarely is possible, because of the cost involved, to pre-treat the solution to remove scale-forming substances. The problem is usually the one of so operating that scale formation shall be at a minimum. The theory of scale formation indicates that if the stagnant film be kept as thin as possible or be torn off as frequently as possible, scale formation will be at a minimum. This is proved in practice, where it is found that all factors which increase the velocity of circulation decrease the rate of scale formation. This is especially true in the forced circulation evaporator. It should be emphasized that such methods cannot, in general, prevent scale formation, but merely decrease the rate of scale formation, and, therefore, increase the time between cleanings.

Scale may be removed from evaporator tubes by any of the usual mechanical methods. Some scales can be dissolved in water; others can be dissolved by the proper chemical treatment. There is now (1935) on the market muriatic acid containing an inhibitor that does not affect its action on calcium carbonate scale or rust, but which does practically stop its action on metallic iron. Most evaporators safely can be boiled out with such a solution.

REMOVAL OF CONDENSED WATER AND NON-CONDENSED GASES.—Condensed water from the first effect is distilled water and should be returned directly to the boilers as boiler feed. Condensate from the later effects will be more or less contaminated by entrained liquid and, according to the type of substance involved and the amount of entrainment, may or may not be suitable for boiler feed make-up. Where it is not so used, condensate from one effect may be flashed down to the temperature of the next effect in a suitable receiver, combined with the condensate of the next effect and flashed down to the temperature of the following one. Unless the evaporator is very large, or the temperature drops across the individual effects very great, this method usually is too complicated to be worth while. Condensate may be removed by reciprocating pumps, self-priming centrifugal pumps, or traps. In some cases condensate pumps may be located far enough below the evaporators so that their suction connections are not under vacuum, and any type of pump then may be used.

The presence of non-condensed gases in steam gradually lowers the rate of heat transfer. Such non-condensed gases are always present, and provision must be made for their removal in any evaporator installation. In a multiple-effect evaporator, the non-condensed gases in all effects after the first will be increased by air dissolved in the solution being evaporated, by air drawn in through leaks or, in some cases, by gases generated as a result of chemical changes in the solution itself. This last, for instance, is the most important source of non-condensed gases in evaporating sugar-beet juices.

Where the amount of gas to be removed is small it is customary to have a connection from the far end of the steam chest to the vapor space of the same body. This passes the gases along from effect to effect, but, if their amount is small, a harmful concentration will not be reached even in the last effect. This method has the advantage that if the vent valves are open too wide, so that steam is vented also, this steam is not lost

but is used in succeeding effects. Where the amount of gas to be removed is large, it may be more practical to vent each effect directly to the condenser.

EVAPORATOR AUXILIARIES.—No special comments are necessary. Pumps for all purposes can be any type that is suitable for the liquor being handled, the heads against which they must work, and the possibility of vacuum on the suction side. Condensers and vacuum producing devices may be any type dictated by practice. Unless the vapor from the last effect is required for some other purpose, or unless an extremely high vacuum must be carried, surface condensers generally are too expensive, both in first cost and in water consumption, for evaporator practice. Evaporator practice regularly uses the direct-contact jet condenser, and since high vacuums seldom are required, a reciprocating vacuum pump with a Corliss type valve is at present (1935) more common than steam-jet ejectors.

7. EVAPORATION TO THE ATMOSPHERE *

EVAPORATION OF WATER FROM TANKS AND RESERVOIRS (Tech. Bull. 271, U. S. Dept. of Agriculture, Dec., 1931).—A series of experiments extending from 1923 to 1930 to determine the evaporation from tanks, reservoirs, open channels, etc., developed the formulas:

For Tanks, $E = (1.465 - 0.0186B)(0.44 + 0.118W)(e_s - e_d)$ . . . . [10]

For Reservoirs, $E = 0.771(1.465 - 0.0186B)(0.44 + 0.118W)(e_s - e_d)$ . . . [11]

where E = evaporation, in., per 24 hr.; B = mean barometer reading, in. of mercury at 30° F.; W = mean velocity of ground wind or water surface wind, mi. per hr.; e_s = mean vapor pressure of saturated vapor at temperature of water surface, in. of mercury; e_d = mean vapor pressure of saturated air at temperature of dew point, in. of mercury. Values of the factor $(1.465 - 0.0186B)$ for various altitudes are given in Table 5.

Table 5.—Value of Factor $(1.465 - 0.0186\,B)$ for Various Altitudes

Altitude above Sea-level, ft.	Barometer, in. of Hg	Factor	Altitude above Sea-level, ft.	Barometer, in. of Hg	Factor	Altitude above Sea-level, ft.	Barometer, in. of Hg	Factor	Altitude above Sea-level, ft.	Barometer, in. of Hg	Factor
0	29.90	0.91	4000	25.81	0.98	8,000	22.28	1.05	12,000	19.23	1.11
1000	28.82	.93	5000	24.88	1.00	9,000	21.47	1.07	13,000	18.53	1.12
2000	27.78	.95	6000	23.98	1.02	10,000	20.70	1.08	14,000	17.86	1.13
3000	26.78	.97	7000	23.11	1.04	11,000	19.95	1.09	15,000	17.22	1.14

DRYERS AND DRYING

By Francis E. Finch

1. CHARACTERISTICS OF DRYERS AND MATERIALS

A dryer is an apparatus for driving off moisture by the application of heat to the material, either by direct or indirect means. Thus, it differs from a dehydrator, such as a filter press or a centrifugal, which extract moisture by mechanical means; and from a calciner, which drives off volatiles, etc., by roasting at a high temperature. The calciner is used occasionally as a combined dryer and roaster, but the shell then is longer than is necessary for roasting alone.

CHARACTERISTICS OF DRYERS.—Where continuous drying is desired, it is necessary to separate the problem into, and to balance, three fundamental parts: 1. Application of heat to the material or to the air mixture which is to absorb the moisture. 2. Means for removing the water vapor, steam or mixture. This includes a study of the vapor pressures involved. 3. Conveying the material, in its wet, semi-dried and dried condition, into, through, and out of the apparatus, giving the material the proper time and contact with the heating- and moisture-absorbing elements.

The various methods of heating dryers are: *a.* By direct or indirect furnace gases, generated by coal, wood, oil, gas or other fuel; *b.* by steam, either exhaust or direct from the boiler. Steam is a most efficient means for rack or batch drying. Superheated steam or steam at high pressure has not been found efficient for continuous dryers, due to

* Staff contribution.

difficulties in mechanical construction and because superheated steam has low capacity for transferring heat; *c.* by warm air from a warm air furnace, bank of steam coils or one of the newer steam heaters.

The usual means of removing moisture driven off from the material is an exhaust fan; in some cases a stack is used. The fan impeller should have few blades, for if dust is drawn off with the moisture, it must pass through the fan unless a dust eliminator is used between dryer and fan. Stacks are satisfactory only with fairly high and uniform exhaust temperatures. With a stack, close regulation and rapid changes in draft conditions are hard to obtain. With either fan or stack, the temperature of the exhaust should be kept above the dew point, or condensation will occur in the exhaust system.

Conveying material into, through, and out of the dryer depends on the material to be dried and type of dryer selected. The thermal efficiency and capacity of the dryer depend greatly on the manner in which material travels through the drying machine.

At present (1935) several hundred different designs of drying machines are built. Therefore, the selection of a dryer for a given purpose involves the problems as above given, and also many other considerations. No certain type of dryer can be said to be the best, except when it is applied to one definite drying problem. The type, design and size of dryer selected for such definite problem must take into account the material handled, available sources of fuel or heat, space occupied, operating labor required, available power, cost of erection, upkeep or repair cost, and, last and most important, whether that type and size will give the desired product at the lowest cost. In considering the last, thermal efficiency, materials used in construction, and interest and depreciation on erected cost usually are the most important.

RELATION OF MATERIAL CHARACTERISTICS TO METHOD OF DRYING.—The varying physical and chemical properties of materials require different methods of drying. Therefore it is best to classify materials, as far as possible, into groups and have different types of dryers, each type having designs along the same general principles, for each group. To group the materials and select the proper type, the information needed is: 1. Amount of free moisture in material as fed to dryer. 2. Specific gravity of dry material. 3. Specific heat of material. 4. Size of feed to dryer. 5. Amount of free moisture allowable in material as discharged from dryer. 6. Maximum temperature to which material may be heated without undergoing undesirable chemical change or without danger of ignition. 7. Whether or not material will be injured by contact with products of combustion. 8. Action of the material as the free moisture is driven off. This usually is one of the physical properties of the material but occasionally is also a chemical one. These classified groups are:

Group A.—Materials which may be heated to high temperatures without injury by contact with products of combustion. Such materials are trap rock, limestone, sand, gravel, slag, some ores, concentrates and clays. The simplest dryer for such materials is a single revolving drum with lifters attached inside. The drum rests on supporting rollers, and has a furnace at one end and a stack or fan at the other. This type has the advantages of low first cost and accessibility for cleaning or repairs. The disadvantages are excessive fuel consumption and, if material is fed from stack to furnace end, large radiation losses and high cost of repairs to drum. If the feed is reversed, exhaust temperatures must be kept well above 212° F. to avoid recondensation and material again becoming wet. Drying to a low final moisture content is difficult if the gases travel parallel to the flow of material. Improvements on some designs of this type of dryer have eliminated some disadvantages.

Group B.—Materials which cannot come in contact with products of combustion, but which are not injured by high temperature. These are kaolin, talc rock, fuller's earth, china clay, some salts, etc. They may be dried by passing through a single drum dryer encased in brickwork. The heat is applied to the outside of the drum, and the products of combustion do not come in contact with the material. This machine is uneconomical to operate, due to the high temperature of escaping gases and high cost and frequency of repairs. Various types of patented indirect heat dryers, with the heat applied through inside tubes or other means, are much more economical to operate.

Group C.—Materials not injured by products of combustion, but which can stand only temperatures of 212° to 400° F. due to danger of ignition or chemical changes. Such materials are bituminous coal, lignite, petroleum coke, wood chips, sawdust, gypsum, various chemicals and fertilizer materials. These may be dried in the single drum and also the "bricked-in" type of dryer, but neither are satisfactory due to high operating costs, danger of ignition and explosions or chemical decomposition. Several designs of semi-direct heat types of dryers using two passes of the furnace gases inside of one revolving drum are more efficient and have lower operating costs.

Group D.—Organic materials which are used for food, as grains, corn germs breakfast

foods, starch foods, cotton seed; chemicals, as nitrate of ammonia, carbonate of potash, etc. These must be dried at low temperatures and some, such as food products, cannot be contaminated by products of combustion. For such materials, dryers of two types usually are used. Both types use live or exhaust steam as their heating medium. One type is a revolving drum with steam pipes attached inside of and revolving with the drum. The other type is a single shell dryer with steam pipes or other steam heater substituted for the furnace. Many arrangements of both types are available.

Group E comprises the largest in number of materials to be dried, and is the most varied. It contains those materials and products which cannot be agitated, due to their physical properties. Among them are ceramic wares, knitted and dyed products, very finely divided materials, extremely sticky materials, etc. For these, drying racks heated by steam or furnace, vacuum dryers, pan dryers and conveying chain dryers of many types are used. Most dryers of this type are designed for one particular product or installation.

Note: The above grouping of materials can be only general. Due to the many designs of dryers now built, a material may be dried economically in the type best suited to another group, modifications and improvements having been made in a certain design, enabling that one design to cover a broader field.

EFFICIENCY OF DRYERS.—The thermal efficiency of a dryer is the ratio of the theoretical heat required to the total heat supplied to it. When the dryer has its own separate furnace, the heat losses of the furnace are taken as part of the heat losses of the dryer. Heat losses in a dryer are: Radiation, 4 to 30%; in waste gases, 7 to 40%; in dried material, 3 to 25%. These losses may be kept near the minimum by proper design and operation.

A well-designed dryer for materials in Group A will have 70 to 82% efficiency; a single-shell dryer of good design, 45 to 60%; a "bricked-in" direct heat dryer, 50 to 65%. A properly designed indirect heat dryer for materials in group B will have, with good operation, as high as 67% efficiency, while improperly designed machines may run as low as 25%. Well designed and well operated dryers for materials in Group C will have 72 to 78% efficiency. For the various materials in Groups D and E, the best designed dryers will have efficiencies of about 60%, while some of poor design have efficiencies as low as 12%.

2. PERFORMANCE OF DRYERS

STEAM HEATED DRYERS.—Rotary Dryer with Pipes Attached to Inside of Drum.—Material, grain. Original moisture, 19%. Final moisture, 1.2%. Dried material per hour, 3820 lb. Water evaporated per hr., 985 lb. Steam consumed per hr., 1540 lb. at 10 lb. gage pressure. Power used, 18 Hp. for drum; 4 Hp. for fan. Temperature of feed, 65° F.; of discharged material, 214° F. Thermal efficiency, 83.4%, not including boiler efficiency.

Rotary Single-Shell Dryer with Bank of Steam Coils at discharge end, fan at feed end. Material, ammonium sulphate. Original moisture, 2.82%. Final moisture,

Table 1.—Performance of Various Types of Rotary Dryers

Type of Dryer	Single-shell. Direct Heat	Single-shell. Bricked-in. Direct Heat	Single-shell. Special Baffles. Direct Heat	Double-shell. Direct Heat	Double-shell. Indirect Heat	Single-shell. Bricked-in. Direct Heat
Material	Limestone	Coal	Zinc concentrates	Silica Sand	Kaolin	Talc Rock
Delivered weight, per hr., lb.	114,000	14,200	13,500	22,000	18,500	8,800
Moisture, initial, percent	5.2	8.6	12.5	11.5	19.8	7.9
Moisture, final, percent	0.3	1.2	3.2	0.6	1.3	0.8
Heat value of fuel, B.t.u.	14,000	12,100	13,200	13,500	13,800	13,200
Fuel consumed per hr., lb	1,260	248	245	360	636	205
Water evaporated per hr., lb.	5,892	1,150	1,434	2,709	4,267	689
Water evaporated per lb. of fuel, lb.	4.67	4.63	5.85	7.52	6.71	3.36
Fuel per ton of dried material, lb.	22.1	34.9	36.3	32.8	68.7	46.6
Heat lost in exhaust, percent	20.8	21.4	17.1	12.1	18.7	25.7
Heat lost in radiation, percent	19.3	19.4	20.4	11.3	18.9	23.3
Heat used to raise temp. of material, percent	20.5	15.5	12.9	14.3	6.8	12.1
Heat used to evaporate water, percent	39.4	43.7	49.6	62.3	55.6	28.9
Thermal efficiency, percent	59.9	59.2	62.5	76.6	62.4	41.0

0.08%. Dried material per hour, 12,100 lb. Water evaporated, 351.4 lb. Steam consumed per hour, 1060 lb. at 15 lb. gage pressure. Power used, 11 Hp. for drum; 4 Hp. for fan. Temperature of feed, 52° F.; of discharged material, 182° F. Thermal efficiency, 78.7% not including boiler efficiency.

Steam Tray Dryer.—Material, pigment. Original moisture, 82%. Final moisture, 0.12%. Material dried in 16 hr., dry weight, 642 lb. Temperature in drying chamber, 216° F. Thickness of material in trays, 1.4 in. Steam pressure, 26 lb. Steam consumed per lb. of water evaporated, 3.19 lb.

RABBLE AND HEARTH DRYER.—Hearth of 360 sq. ft. with mechanically-operated steel rabbles. Heat generated in Dutch oven type of furnace, gases of combustion passing underneath hearths, heating the material indirectly. Material, lead flotation concentrates. Original moisture, 13%. Final moisture, 6.5%. Weight of dried material, 6200 lb. per hour. Water evaporated, 469 lb. Fuel used per hour, 160 lb. of 11,000 B.t.u. coal. Water evaporated per lb. of fuel, 2.93 lb. Temperature of dried material, 214° F. Thermal efficiency of dryer, 40.05%.

Table 2.—Water Evaporated and Heat Required for Drying*

M = percentage of moisture in material to be dried. Q = lb. water evaporated per ton (2000 lb.) of dry material. H = B.t u. required for drying, per ton of dry material

M	Q	H	M	Q	H	M	Q	H
1	20.2	85,624	14	325.6	424,884	35	1,077	1,269,240
2	40.8	108,696	15	352.9	458,248	40	1,333	1,555,960
3	61.9	130,424	16	381.0	489,720	45	1,636	1,895,320
4	83.3	156,296	17	409.6	521,752	50	2,000	2,303,000
5	105.3	180,936	18	439.0	554,680	55	2,444	2,800,280
6	127.7	206,024	19	469.1	588,392	60	3,000	3,423,000
7	150.5	231,560	20	500.0	623,000	65	3,714	4,222,680
8	173.9	257,768	21	531.6	658,392	70	4,667	5,290,040
9	197.8	284,536	22	564.1	694,792	75	6,000	6,783,000
10	222.2	311,864	23	597.4	732,088	80	8,000	9,023,000
11	247.2	339,864	24	631.6	770,392	85	11,333	12,755,960
12	272.7	368,424	25	666.7	809,704	90	18,000	20,223,000
13	298.9	397,768	30	857.0	1,022,840	95	38,000	42,623,000

*Formulas: $Q = 2000\ M/(100 - M)$; $H = 1120\ Q \div 63{,}000$.

The value of H is found on the assumption that the moisture is heated from 62° to 212° F. and evaporated at that temperature, and that the specific heat of the material is 0.21; $[2000 \times (212 - 62) \times 0.21] = 63{,}000$.

3. DRYER DESIGN

CALCULATIONS FOR DESIGN OF DRYING APPARATUS.—A most efficient system of drying of moist materials consists in a continuous circulation of a volume of warm dry air over or through the moist material, then passing the air charged with moisture over the cold surfaces of condenser coils to remove the moisture, then heating the same air by steam-heating coils or other means, and again passing it over the material. In the design of apparatus to work on this system it is necessary to know the amount of moisture to be removed in a given time, and to calculate the volume of air that will carry that moisture at the temperature at which it leaves the material, making allowance for the fact that the moist, warm air on leaving the material may not be fully saturated, and for the fact that the cooled air is nearly or fully saturated at the temperature at which it leaves the cooling coils. A paper by Wm. M. Grosvenor, read before the Am. Inst. of Chemical Engineers (*Heat. and Vent. Mag.*, May, 1909) contains a "humidity table" and a "humidity chart" which greatly facilitate the calculations required. Table 3 is a condensation of the original table. It is based on the following data:

$D_a + 0.04\%\ CO_2 = 0.001293052/(1 + 0.00367 \times \text{Temp. deg. C.})$ kg. per cu. meter;
$D_w = 0.62186 \times D_a$,

where D_a and D_w = respectively, density of air and water vapor. Density at partial pressure ÷ density at 760 m.m. = partial pressure ÷ 760 m.m. Specific heat of water vapor = 0.475; specific heat of air = 0.2373. Kg. per cu. meter × 0.062428 = lb. per cu. ft. Compare also the tables of H. M. Prevost Murphy, p. 1–06. These latter tables are useful in calculating the amount of air necessary to absorb moisture, for any and all drying problems.

The term "humid heat" in Table 3 is defined as the B.t.u. required to raise 1° F. one pound of air plus the vapor it may carry when saturated at the given temperature and pressure; "humid volume" is the volume of one pound of air when saturated at the given temperature and pressure.

Table 3.—Humidity Tables for Drying Calculations

Temp., deg. F.	Vapor Tension, Milli-meters of Mercury	Water Vapor per lb. of Air, lb.	Humid Heat, B.t.u.	Humid Volume, cu. ft.	Density, lb. per cu. ft. at 760 Millimeters		Volume, cu. ft. per lb. of	
					Dry Air	Saturated Mixture	Dry Air	Satu-rated Mixture
32	4.569	0.003761	0.2391	12.462	0.080726	0.080556	12.388	12.414
35	5.152	.0042435	.2393	12.549	.080231	.080085	12.464	12.496
40	6.264	.0050463	.2398	12.695	.079420	.079181	12.590	12.629
45	7.582	.0062670	.2403	12.843	.078641	.078348	12.718	12.763
50	9.140	.0075697	.2409	12.999	.077867	.077511	12.842	12.901
55	10.980	.0091163	.2416	13.159	.077109	.076685	12.968	13.041
60	13.138	.010939	.2425	13.326	.076363	.075865	13.095	13.180
65	15.660	.013081	.2435	13.501	.075635	.075039	13.222	13.325
70	18.595	.015597	.2447	13.683	.074921	.074219	13.348	13.471
75	22.008	.018545	.2461	13.876	.074218	.073471	13.474	13.624
80	25.965	.021998	.2478	14.081	.073531	.072644	13.600	13.777
85	30.573	.026026	.2497	14.301	.072852	.071744	13.726	13.938
90	35.774	.030718	.2519	14.539	.072189	.070894	13.852	14.106
95	41.784	.036174	.2545	14.793	.071535	.070051	13.979	14.275
100	48.679	.042116	.2575	15.071	.070894	.069179	14.106	14.455
105	56.534	.049973	.2610	15.376	.070264	.068288	14.232	14.643
110	65.459	.058613	.2651	15.711	.069647	.067383	14.358	18.840
115	75.591	.068662	.2699	16.084	.069040	.066447	14.484	15.050
120	87.010	.080402	.2755	16.499	.068443	.065477	14.611	15.272
125	99.024	.094147	.2820	16.968	.067857	.064480	14.736	15.509
130	114.437	.11022	.2896	17.499	.067380	.063449	14.863	15.761
135	130.702	.12927	.2987	18.103	.066713	.062374	14.989	16.032
140	148.885	.15150	.3093	18.800	.066156	.061255	15.116	16.325
145	169.227	.17816	.3219	19.609	.065601	.060104	15.242	16.643
150	191.860	.21005	.3371	20.559	.065154	.058865	14.368	16.993
155	216.983	.24534	.3553	21.687	.064539	.057570	15.494	17.370
160	244.803	.29553	.3776	23.045	.064016	.056218	15.621	17.788
165	275.592	.35286	.4054	24.708	.063502	.054795	15.748	18.250
170	309.593	.42756	.4405	26.790	.062997	.053305	15.874	18.761
175	347.015	.52285	.4856	29.454	.062500	.051708	16.000	19.339
180	388.121	.64942	.5458	32.967	.062015	.050035	16.126	19.987
185	433.194	.82430	.6288	37.796	.061529	.048265	16.253	20.719
190	482.668	1.00805	.7519	44.918	.061053	.046391	16.379	21.557
195	536.744	1.4994	.9494	56.302	.060588	.044405	16.505	22.521
200	595.771	2.2680	1.3147	77.304	.060127	.042308	16.631	23.638
205	660.116	4.2272	2.1562	131.028	.059674	.040075	16.758	24.954
210	730.267	15.8174	15.9148	562.054	.059228	.037323	16.884	26.796

4. KILN DRYING OF LUMBER*

The basic object in kiln drying of wood is to remove part of the moisture naturally present in it, which if allowed to remain ordinarily would interfere with its use. As soon as evaporation from the surface of the wood begins, a moisture gradient is established. The wood has been made drier on the surface than in the interior, and thereby a movement of moisture from the interior toward the surface has been started. As wood dries it shrinks. If the moisture gradient is too great, surface shrinkage will cause checks and other defects. Hence, drying conditions must be so controlled that the moisture gradient will not exceed that which is safe for the species and thickness of stock being dried. Such control is obtained by controlling temperature and humidity, and maintaining a positive ınd ample circulation of air. In comparison with drying problems where moisture renoval is the only factor, kiln drying of wood is a complex process, for if drying conditions temperature, humidity and circulation) are not properly controlled the wood may be njured and its value depreciated.

DRY KILNS may be divided into two general types: 1. The stationary or compartnent type, in which the entire charge of wood is subjected to approximately the same emperature and humidity conditions, which are varied as the drying progresses; 2. The rogressive type in which the charge is moved progressively from the entering end to he discharge end. The lowest temperatures and highest humidities prevail at the enterıg end and highest temperatures and lowest humidities at the discharge end.

Kilns also may be either of the *natural* or *forced circulation* type. In natural circulaon kilns, air movement is caused by differences in temperature; in forced circulation

* Contributed by L. V. Teesdale, Senior Engineer, U. S. Forest Products Laboratory, Madison, Wis.

kilns, by means of blowers, fans, steam jets, or water sprays. The method of disposing of evaporated moisture separates kiln types into two classes: 1. The *condenser kiln*, where the evaporated moisture may be removed by condensers in the kiln; 2. The *ventilated kiln*, where it is removed by passing out of the kiln some of the moist, hot air and admitting an equivalent amount of fresh air.

No general comparison of efficiency of kiln types can be made, as one high in evaporating efficiency may produce a product that is poorly dried and has a high degrade. Also some species may be dried rapidly under severe conditions with little degrade, whereas others must be dried very slowly under mild conditions to avoid damage.

WORK OF EVAPORATING MOISTURE.—The useful work accomplished in evaporating moisture from lumber may be considered as the latent heat of the evaporated moisture. The actual heat absorption is greater than this. About 30 B.t.u. per pound of dry wood are absorbed in separating the moisture from the wood when drying to zero moisture content. In practice, wood seldom is dried below 5% moisture content, and the amount of heat required is only about $^1/_3$ as much as when drying to zero moisture content. It is necessary, of course, to heat the wood from its initial temperature to the final temperature of the kiln. The total heat expenditure required for evaporation only may be considered as the sum of: 1. Latent heat of vaporization; 2. Heat required to overcome hygroscopic attraction of the wood; 3. Heat required to raise temperature of the material and its contained water to temperature of evaporation. To this must be added the amount of heat lost by radiation through walls, ceiling, and floor; heat required to raise temperature of air in the kiln; heat required for air entering in ventilated kilns or for reheating air in condensing kilns; and the amount of steam used in steam sprays for humidifying and for steaming or conditioning treatments. The amount of steam required to evaporate a known amount of moisture at given conditions readily can be calculated within reasonable limits.

For estimating purposes, in ventilated kilns an allowance for heating of about 5% of the gross volume of kiln space should be made for the fresh air that is admitted per minute to replace exhausted moist air. In condensing kilns, cooling effects of the condensers can be calculated. The steam used for humidification varies enormously and depends to a considerable degree on the construction of kilns and skill of the operator. In well-insulated kilns with tight doors and vapor resisting construction, vapor losses are low and a skillful operator can save steam by careful operation of the exhaust dampers. Steam used in steam sprays may vary from 25 to 100% of that used in heating coils.

CONDENSATION.—The condensation per square foot of radiation in dry kilns, assuming proper size of supply lines, drain lines and traps, depends on: 1. Air circulation across the coils; 2. Steam pressure; 3. Kiln temperature. In natural circulation kilns, about 1 sq. ft. of radiation is allowed for each 6 cu. ft. of kiln space above the rail level. At 5 lb. per sq. in. steam pressure, maximum kiln temperature is between 165 and 180° F. This temperature occurs when the lumber is practically dry. For higher temperatures, high-pressure steam is used. In one test kiln, the maximum condensation per sq. ft., at a temperature of 185° F. and 15 lb. per sq. in. steam pressure, was 0.17 lb. per sq. ft. per hr. Only about $^1/_3$ of this amount was used to maintain a temperature of 120° F. At 75 lb. per sq. in. steam pressure and 180° F., maximum condensation was 0.39 lb. per sq. ft. In kilns having forced circulation the amount of radiation required is reduced because the efficiency of the heating surface is increased. In external blower kilns the ratio of radiation to kiln space may be as low as 1 to 20.

Steam Required to Dry Lumber.—Table 4 shows the amount of steam required per 1000 board feet to dry various hardwoods from an air-dried condition, averaging 20% moisture content, to a final average moisture content of 5%, and for various softwoods from the green condition to a moisture content of 5%. The table is based on well-

Table 4.—Estimated Steam Consumption for 1-in. Lumber in Commercial Kilns

Species of Lumber	Condition	Time of Drying, days	Lb. of Water Evaporated per 1000 Board-ft.	Lb. of Steam per lb. of Water Evaporated	Lb. of Steam per 1000 Board-ft.
Oak.	Air dried	8–12	600	3 –4.5	1800–2700
Birch, Maple	Air dried	6– 8	575	2.5–4	1450–2350
Gum, red.	Air dried	7–10	450	2.5–4	1100–1800
Douglas fir	Green	3– 4	750	2 –2.75	1500–2050
Longleaf yellow pine.	Green	4	2000	2 –2.75	4000–5500
Shortleaf yellow pine.	Green	4	2550	2 –2.75	5100–7000
Ponderosa Pine.	Green	4	1800	2 –2.75	3600–5000

constructed compartment kilns, utilizing steam to control humidity, and thermostatic devices to regulate temperature and relative humidity. Where no means of controlling humidity is used, somewhat less steam is required and the quality of the product is lower. In poorly insulated kilns and kilns with leaky doors and walls, the steam consumption increases.

HEAT INSULATION

By P. Nicholls

General Reference.—Traite de la Chaleur, Peclet, translated in Steam Covered and Bare Pipe, by Paulding.

The practice of heat insulation may be divided into fields based on temperature ranges, which fix the types of materials best suited to the conditions. See Table 1.

Table 1.—General Division of Heat Insulation Fields

Field	Temperature Range	Applications	Materials Used
1	Below 32° F.	Refrigeration. Cold storage.	(a) Organic materials, wood, cork, vegetable and animal fibers. (b) Rock wool. (c) Air spaces. (d) Vacuum.
2	32° to 100° F.	Cold-water pipes. Building and room insulation.	
3	100° to 230° F.	Hot-water heating. Low-pressure steam heating. Hot-air heating.	(a) Air spaces. (b) Lower grade asbestos goods. (c) Molded materials. (d) Rock wool.
4	230° to 800° F.	High-pressure steam plants. Industrial processes. Heating ovens, etc.	(a) 85% magnesia (600° F.). (b) High grade asbestos goods. (c) Diatomaceous earth.
5	800° to 1800° F.	Furnace settings. Kilns. High-temperature stills, etc.	(a) Diatomaceous earth. (b) Clays.
6	Above 1800° F.	Firebrick and ceramic products field.	Firebrick and clays of various types.
7	3000° F. up. Fire protection only	Safes and vaults. Walls and buildings. Structural steel.	Varies with the temperature likely to occur.

1. COMMERCIAL INSULATORS

COMMERCIAL INSULATORS are classed as follows

1. Loose Vegetable or Animal Fibers, felted or molded, used only at low temperatures on refrigerating, cold-water or hot-water pipes or surfaces. Such insulators are wool and hair felt, and cork, by themselves or combined with asbestos or roofing papers or containers.

2. Asbestos Insulations made from loose fibers, molded to shape, and the surface hardened with a binder; or formed into a mattress between woven asbestos cloth; also formed into paper and built-up with intervening air spaces. Available for all moderate temperatures up to the limit of steam temperatures, and, when properly supported, for fire protection.

3. Mineral Wools, made by steam blasting blast furnace slag or fusible rocks, is used for stuffing spaces, and for forming into blocks with other materials. It chiefly is used for low-temperature conditions. It does not rot, but is brittle, easily shaken down, and should be supported by wires or by tufting.

4. Molded Powders with or without binders. The efficiency of such insulators depends largely on the proportion of entrained air in the molded product. The larger the percentage of voids, the greater will be its efficiency as an insulator. Those powders whose crystals have the smallest absolute conductivity and the best reflecting surface have the best efficiency.

Plaster of Paris, lime, gypsum and other materials have been used. Plaster of Paris, being an acid salt, corrodes metals; lime and gypsum naturally have low efficiency due to their high density when molded.

Molded cork scrap is used for low temperatures and refrigerating conditions.

Infusorial earth molds to a low apparent density, but unless used in block form, as cut from its bed, it has poor binding qualities and requires an artificial binder to give it

cohesion. Being an oxide of silica, it can, in its natural form, be used on the highest temperatures.

Basic hydrated carbonate of magnesia molds into shapes with 90% voids, and, due to interlocking of the minute crystals, possesses considerable inherent strength. Commercially, mixed with 10 to 15% of asbestos fiber to give added strength, it is known as *85% Magnesia*. It can be machined accurately to shape. A limiting maximum temperature of 600° F. generally is recommended.

Alumina and other refractory powders also are molded with binders. Light-weight refractories are made by mixing clay with carbonaceous materials which are burnt out during firing.

SYMBOLS FOR HEAT TRANSMISSION.—The following symbols are selected from Symbols for Heat and Thermodynamics approved by Am. Stds. Assoc., Feb., 1931. A = area; t = temperature, deg. F. or deg. C.; T = absolute temperature (deg. F. + 460) or (deg. C. + 273); L = length of path of heat, or thickness; Q = total quantity of heat transferred; τ = time; q = thermal transmission (heat transferred per unit of time); k = thermal conductivity (heat transfer per unit time per unit area, and per deg. per unit length); R = thermal resistance (deg. per unit of heat transfer per unit time); C = thermal conductance (heat transfer per unit time, per deg.); C_A = thermal conductance per unit area; h = surface coefficient of heat transfer (heat transferred per unit time per unit area per deg.); U = overall coefficient of heat transfer (heat transferred per unit time per unit area, per deg. overall).

UNITS EMPLOYED IN HEAT INSULATION.—American engineering practice has greatly confused data and published papers by the variety of units used, and care is necessary in using formulas and coefficients. The units most commonly employed are the square foot for area, the inch for thickness, the hour for time, and the Fahrenheit scale for temperature. In refrigeration work it is quite common to use the 24-hr. day for the time unit. Problems are simplified by using the foot for all length measurements.

In the square foot-inch-hour system, the conductivity coefficient is in B.t.u. per hour per square foot per degree F. per inch thickness. With the day as the unit of time the coefficient is 24 times this value, and with the foot as the unit of thickness, one-twelfth of the value.

Values sometimes are given in French or electrical units. The following factors will transform them to standard units.

1 calorie (small or gram-calorie) = 0.003968 B.t.u.

1 watt = 3.4128 B.t.u. per hour.

1 calorie per sq. centimeter per sec. = 13,280 B.t.u. per sq. ft. per hour.

Conductivity in French units = one calorie per second per sq. centimeter per centimeter thickness per degree C. = 2903 B.t.u. per hour per sq. ft. per inch thickness per degree F.

2. LAWS AND MATHEMATICS OF HEAT INSULATION

The fundamental mathematics and laws were developed by the French physicist Peclet and published in his Traité de la Chaleur in 1853. He also developed empirical laws from experiments with laboratory scale apparatus, embodying in his results values of fundamentals which made his formulas perfectly general.

The majority of practical applications involve the transfer of heat under constant conditions and do not include changes in the amount of heat absorbed in the insulation itself. Thus they are not affected by the specific heat of the insulation. Even though the conditions or temperature may be variable, the actual weight of insulation usually is not large and its total heat capacity will not greatly affect the results. Exceptions to this, under some conditions, may include buildings, underground and fireproofing problems.

Insulation calculations can be divided into two general types: 1. Where the surface temperatures on each side of the insulation are fixed, and the heat transfer is dependent on the heat conductivity of the material and the conductivity laws. 2. Where the heat must pass through materials and also be dissipated from or absorbed at one or more surfaces by radiation, absorption or contact with a fluid, usually air, only the temperatures at the extreme of the transfer being known. Assuming that a constant condition of heat flow has been reached, the mathematics depend on the fact that the total flow calculated at any section at right angles to the direction of the flow must be equal to the total flow at any other section.

HEAT FLOW THROUGH A MATERIAL WITH KNOWN SURFACE TEMPERATURES.—Let t_1 and t_2 = temperatures of the two surfaces; k = heat conductivity co-

efficient, assumed same for all temperatures; S = a shape factor. Then the heat units q passing through the body per unit time are expressed in general by:

$$q = S \times k \times (t_1 - t_2) \quad . \quad . \quad . \quad . \quad . \quad . \quad . \quad . \quad . \quad [1]$$

If q = B.t.u. per hr., and k = B.t.u. per hr. per sq. ft. per deg. F. per inch thickness, the value of S, the shape factor, is:

For a Flat Block.—$S = A/L$; A = area, sq. ft., L = thickness, in.

For a Cylinder.—$S = 2\pi l/12 \log_e(r_2/r_1) = 0.227\, l/\log_{10}(r_2/r_1)$; r_1 and r_2 = inner and outer radii, in.; l = length of cylinder, ft.

For a Hollow Sphere.—$S = 4\pi\, r_1 r_2/144\,(r_2 - r_1) = 0.0875\, r_1 r_2/(r_2 - r_1)$; r_1 and r_2 = inner and outer radii.

Langmuir, Adams and Meekle (*Trans.* Am. Elect. Chem. Soc., vol. xxiv, 1913) give additional shape factors, one of which is:

For a Hollow Rectangular Construction with square edges and corners, and thickness of sides L, in.; S for each flat block side $= A/L$; A = inside area of side, sq. ft. S for each 0.54 $l/12$; l = length of inside edge, ft. S for each corner $= 0.15\,L/144$.

The complete S will be the sum of all such portions as are transmitting the heat; for a complete parallelopiped the whole S will be

$$S = \{2\,(l_1 l_2 + l_2 l_3 + l_3 l_1)/L\} + 0.45\,(l_1 + l_2 + l_3) + 0.008\,L \quad . \quad . \quad . \quad [2]$$

CONDUCTIVITY VARIABLE.—The formulas for heat transfer, given above, apply when conductivity is constant; this is sufficiently accurate for many purposes, but for greater accuracy the formula

$$\text{Heat transfer per unit time, } q = S\int_{t_2}^{t_1} f(t)\,dt \quad . \quad . \quad . \quad . \quad . \quad . \quad [3]$$

must be used, where $f(t)$ is true conductivity expressed as a function of temperature of the material. The true function would be expressed as a curve of conductivity against temperature. All insulating materials show an increase of conductivity with temperature; as a first approximation the curve can be taken as a straight line, so that $f(t) = a + bt$ expresses the conductivity, a and b being constants. The above general formula then reduces to $q = S\{a + b \times {}^1/_2\,(t_1 + t_2)\}\,(t_1 - t_2)$, which can be stated as

$$q = Sk_m(t_1 - t_2) \quad . \quad . \quad . \quad . \quad . \quad . \quad . \quad . \quad . \quad . \quad [4]$$

where k_m = conductivity for the mean temperature ${}^1/_2(t_1 + t_2)$. S has the same values as given above.

The derivation of the true conductivity curve is of interest mainly in research work. For a method of obtaining it and a graphical method of solving heat transmission problems with variable conductivity and surface transmission coefficients see P. Nicholls, *Trans.* A.S.H.V.E., vol. xxviii, 1922, p. 323–342; also G. B. Wilkes, *Jour.* Am. Ceramic Soc., vol. xvi, 1933, p. 129.

HEAT FLOW WITH COMBINED CONDUCTION AND SURFACE EFFECTS.—The rate of heat flow to or from a surface can be stated as: Heat per unit time passing through unit area of surface, $q = h(t_s - t_0)$, where t_s = surface temperature; t_0 = air temperature; h = coefficient of surface transmission in heat units per unit time per unit area per deg. F. temperature difference. The numerical values for h vary with temperature and conditions, and are treated more fully on p. 3-57.

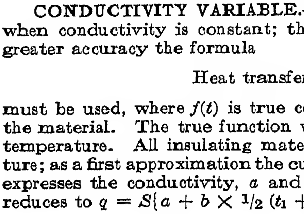


Fig. 1

With flow of heat under constant conditions the amount of heat passing through all sections at right angles to direction of flow must be the same. Problems are solved by equating these values. Practical problems usually are limited to the following:

Flat Parallel Plates.—Take the general example of a series of parallel plates of homogeneous materials with air spaces, coefficients and thicknesses as shown in **Fig. 1**; temperatures in deg. F., thickness L in inches, surface transmission coefficients h and air space conductance coefficients C_s in B.t.u. per hr. per sq. ft. per deg. F. temperature difference, and conductivity k in the same units per 1 in. thickness.

Let q = B.t.u. per hr. flowing through 1 sq. ft. of wall area. Then $q = h_1(t_0 - t_1) = (k_1/L_1)(t_1 - t'_1) = C_s(t'_1 - t_2) = (k_2/L_2)(t_2 - t'_2) = C'_s(t'_2 - t_3) = (k_3/L_3)(t_2 - t'_3) = h_2(t'_3 - t'_0)$.

If U = overall transmission, B.t.u. per hr. per sq. ft. per deg. F. difference between hot and cold air, these equations give

$$\frac{1}{(1/h_1) + (1/h_2) + (1/C_s) + (1/C'_s) + \qquad (L_3/k)_3}$$

It follows that $q = U(t_0 - t'_0)$. Values for the various surface temperatures are then successively found from equations: $t_1 = t_0 - (q_1/h_1)$; $t'_1 = t_1 - (qL_1/k_1)$; $t_2 = t'_1 - (h/s_1)$, etc. For a hot surface of temperature t_1, insulated with one thickness of L in. and with air temperature at t_0,

$$q = \{kh/(k + hL)\}(t_1 - t_0) \qquad [6]$$

Cylindrical Surfaces.—Let r_1 and r_2 = respectively inner and the outer radii of insulation, in.; t_1 and t_2 = respectively, temperature of inner and outer surfaces; t_0 = air temperature; q_1 = B.t.u. per hr. per ft. length. By equating heat transfer to surface loss,

$$\frac{1}{12} \times \frac{12 \cdot 2\pi r_2}{(1/h) + (r_2/k)\log_e(r_2/r_1)}(t_1 - t_0) = \frac{12 \cdot 0.523\, r_2 (t_1 - }{(1/h) + 2.3\,(r_2/k)} \qquad [8]$$

Having found the value of q_1, t_2 can be computed from one of the first equations.

If q = B.t.u. per sq. ft. of inner surface,

$$q \quad \frac{r_2}{r_1} \times \frac{1}{(1/h) + 2.3}$$

If insulation consists of two layers, the inner layer being of conductivity k_1 and radii r_1 and r_2, and the outer layer of conductivity k_2 and radii r_2 and r_3,

$$q_1 \text{ (per foot length)} = \frac{0.523\, r_3 (t_1 - t_0)}{(1/h) + 2.3\,(r_3/k_1)\log_{10}(r_2/r_1) + 2.3\,(r_3/k_2)\log_{10}(r_3/r_2)} \qquad [10]$$

These equations, both for flat surfaces and cylinders, assume that h and k are constants; for problems in the refrigerating and heating fields, values can be chosen and the equations solved directly. For the steam and refractories fields, in which variations of both h and k with temperature are large, assumptions must be made for the unknown temperatures, the corresponding values for the mean values of h and k selected, and the values for heat flow and temperatures computed. If temperatures do not correspond with those assumed, a new trial can be made. This process is rather tedious and for frequent computations graphic methods can be used. See L. B. McMillan, *Trans.* A.S.M.E., vol. xlviii, 1926, p. 1285; P. Nicholls, *Trans.* A.S.H.-V.E., vol. xxviii, 1922, p. 334.

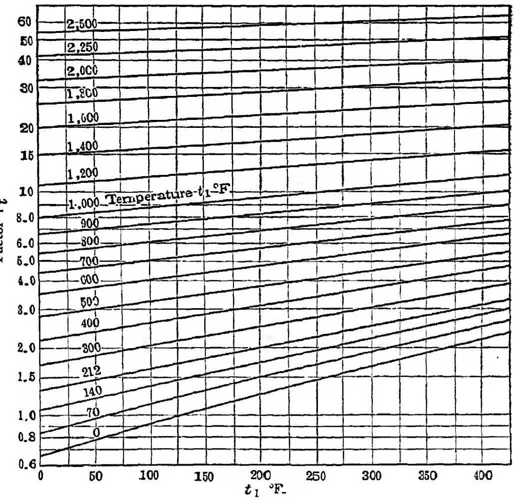


FIG. 2. Values of F_t

HEAT FLOW TO OR FROM A SURFACE.—The heat passing through 1 sq. ft. of surface can be expressed as $q = h\,(t_s - t_0)$ where t_s = temperature of surface, and t_0 = air temperature. Working values for surface coefficient h are available in each field of application of insulation for small temperature differences and for where surrounding objects are at the same temperature as the air. These values of h usually are for natural convection, but often include the effect of such average windage over the surface as is associated with the conditions. For more exact values, and for large temperature differences and for special conditions, the transfers by radiation and by convection must be computed.

The radiation q_r in B.t.u. per hr. per sq. ft. of surface is

$$q_r = 0.174\, e\,[\{(t_1 + 460)/100\}^4 - \{(t_2 + 460)/100\}^4] \qquad [11]$$

where t_1 = temperature deg. F. of surface considered, t_2 = temperature of the surrounding objects, e = total emissivity relative to a black body. The value to be used for e for

metallic surfaces may be uncertain, as it depends materially on polish, oxidization and cleanness. The following average values are sufficient for most insulation problems:

Surface	e	Surface	e
Aluminum or tin, polished..........	0.08	Steel and iron, commercial...	0.80 to 0.90
Aluminum and tin, varnished.......	.20	Non-metallic materials......	.92 to .96
Aluminum paint..................	.40	Lampblack................	.96
Oxide paints, all colors............	.94		

For parallel surfaces, large as compared with distance between them,

where e_1 and e_2 = emissivities of the two surfaces.

For most purposes labor in computation can be saved by expressing

$$q_r = eF_t\,(t_1 - t_2) \qquad [12]$$

where $$F_t = 0.174\,[\{(t_1 + 460)/100\}^4 - \{(t_2 + 460)/100\}^4] \div (t_1 - t_2) \qquad [13]$$

and obtaining F_t from Fig. 2, given by F. S. King.

The convection factor of the surface heat transfer depends on size and location of the surface; further, it may be natural or free convection, or forced convection due to windage. If q_c = transfer due to convection, B.t.u. per hr. per sq. ft.; $(t_1 - t_2)$ = difference between surface and air temperatures, deg. F.; an approximate formula for *free convection* for large vertical surfaces is

$$q_c = 0.22\,(t_1 - t_2)^{4/3} \qquad [14]$$

or per sq. ft. per deg. F., $$h_c = 0.22\,(t_1 - t_2)^{1/3} \qquad [15]$$

For horizontal surfaces the convection transfer is increased if a hot surface faces upwards or a cold surface downwards; it is decreased if a cold surface faces upwards or a hot surface downwards. The approximate increase or decrease of q_c can be taken as 30%. R. H. Heilman, *Trans.* A.S.M.E., FSP-51-41, pp. 294-5, gives complete diagrams and factors for obtaining both the radiation and convection surface of various shaped surfaces.

Windage greatly increases loss by convection. Experimental data for unconfined surfaces are meager, but it usually is not possible in practice to define the velocity of the wind, especially for large surfaces. From tests by F. B. Rowley on a 1 ft. square area, approximate values are given by the following:

Convection loss with wind = (convection loss in still air) $\times F \times V$, where V = velocity, and F = factor depending on the nature of the surface, F is greater as the surface is rough or as the wind can penetrate into the body. Values for F are:

Kind of Surface	F, with V in ft. per min.	F, with V in miles per hr.
Smooth (glass or paint)...........	0.33	0.0037
Moderately rough (brick).........	.54	.0061
Very rough (stucco).............	.80	.0090

TRANSFER OF HEAT ACROSS AIR SPACES.—An air space, rightly used, is an efficient insulator at low temperatures. The transfer of heat consists of direct radiation and air-transfer factors, *i.e.*, resultant of conduction and convection. The radiation factor depends on type of surface. It usually is assumed to be independent of width of

Table 2.—Average Value of Conductance for Vertical Air Spaces 24 in. High

(B.t.u. per sq. ft. per hr. per deg. F. temperature difference between surfaces)

Width of Space, in.	Air Transfer Coefficient. Conduction and Convection (without Radiation) for Given Width	Total Transmission Coefficient, C_g, for Given Width, B.t.u.					
		Between Building Materials		Between Bright Tin Surfaces		Between Building Materials and Bright Tin	
		At 32° F.	At 75° F.	At 32° F.	At 75° F.	At 32° F.	At 75° F.
1/8	1.36 B.t.u.	2.14	2.30	1.40	1.41	1.43	1.45
1/4	0.68	1.46	1.62	0.72	0.73	0.75	0.77
3/8	.45	1.23	1.39	.49	.50	.52	.54
1/2	.34	1.12	1.28	.38	.39	.41	.43
5/8	.28	1.06	1.22	.32	.33	.35	.37
3/4	.25	1.03	1.19	.29	.30	.32	.34
7/8	.23	1.01	1.17	.27	.29	.30	.32
1	.21	0.99	1.15	.25	.26	.28	.30
2	.18	.96	1.12	.22	.23	.25	.27
3	.19	.97	1.13	.23	.22	.26	.28

space; this is not strictly true, due to effect of the side of the spacers. Conduction is inversely proportional to width of space, until it becomes wide enough, at about $^3/_8$ in., to permit movement of air, when convection assists the transfer. Up to about $^3/_8$ in. width the total transfer is the same for vertical and horizontal spaces; above $^3/_8$ in., transfer with a horizontal space is smaller than for a vertical space if the hot face be on top, and greater if it be below. Efficiency of air spaces depends on their being sealed, so that no air can circulate through them. If circulation occurs, the insulating effect of the spaces is lost and it may actually increase the transfer.

The heat transfer, within close limits, varies with height of space and type of spacers used; for these reasons, the values obtained by investigators have differed. The method of test is to measure total transfer, compute radiation factor, and call the difference the air-transfer factor. In computing radiation (see p. 3–57), emissivity is $e_1e_2/(e_1 + e_2 - e_1e_2)$, where e_1 and e_2 are emissivity constants for the two inner surfaces of the space.

Table 2 gives values for C_S, conductance per hr. per sq. ft. per deg. F., using air-factor values determined by U. S. Bureau of Standards (Dickinson and Van Dusen, A.S.R.E., vol. iii, Sept., 1916). Emissivities of building materials are taken as 0.94, and of bright metal surfaces (tin or aluminum) as 0.08. Table 2 gives values for mean temperatures of 32° and 75° F., and an assumed temperature difference of 18° F. between faces; changes from 18° make little difference. The resulting radiation factors per deg. F. are:

	At 32° F.	At 75° F.
Building material to building material	0.78	0.94
Bright metal to bright metal	.04	.05
Building material to bright metal	.07	.09

For vertical air spaces 8 in. high, the following air transfer coefficients were found:

Width of space, in.	$^1/_8$	$^1/_4$			$^5/_8$	$^3/_4$	$^7/_8$	1.0
Coefficient	1.50	0.75	0.50	0.38	0.31	0.32	0.34	0.35

Adding the radiation coefficient gives total transmission coefficient for this height of space.

METAL ENCLOSED INSULATION.—If the metal is only an outside protecting sheet and does not pass into or through insulation it will not affect heat transfer computations. Insulating value will be improved in so far as the metal may seal the surface and prevent air circulation through the insulation, or penetration by wind or moisture.

In some applications, insulation is wholly or partly enclosed by metal, as in some panel constructions, on refrigerators and in safes. Such construction may materially decrease the average effectiveness of insulation. Exact computations for heat flow may be difficult but the analysis by M. S. Van Dusen (U. S. Bureau of Standards Research Paper 207) can be adapted to give sufficiently accurate results. The order of increase of heat flow which results from transfer by the metal sides is shown in Fig. 3, which is for a 40 × 40 in. panel completely enclosed by sheet iron. The panel is assumed to be exposed to air on both sides, and the insulation to have a value of $k = 0.3$ B.t.u. per hr. per sq. ft. per in. thickness per deg. F.

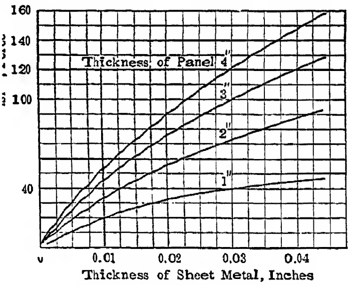


Fig. 3. Effect of Thickness of Metal on Heat Flow

The same bulletin treats the effects of bolts passing through insulation. Flow of heat through bolts increases greatly with increased area of head exposed to the air, or as it has direct thermal contact with a metal covering sheet. Hence the head should be insulated from the metal sheet. If the head is well insulated, or if there is no metal covering, heat flow is materially reduced, and by first principles is

$$\frac{0.0069\ (t_1 - t_2)}{(1/h_1 A_1) + (1/h_2 A_2) + (L/Ka)} \text{ B.t.u. per hr.} \qquad [16]$$

where t_1 and t_2 = air temperature on hot and cold sides, deg. F.; h_1 and h_2 = surface coefficients, B.t.u. per hr. per sq. ft.; A_1 and A_2 = exposed areas, sq. in., of bolt heads; a = area. sq. in., of bolt; K = thermal conductivity of the metal, B.t.u. per hr. per sq. ft., per in. thickness per deg. F. (steel = 300; copper = 2600). For a stud tapped into metal on one side $(1/h_2 A_2)$ will be zero. h_1 and h_2 may be taken as 1.5, or selected from Table 3.

3. INSULATION OF COLD SURFACES

The insulation of cold surfaces covers applications where transference of heat from hotter air must be prevented, and includes refrigerating, cold storage, ice houses and cold-water pipes. Its distinguishing feature is that the insulation is at a lower temperature than the air, so that moisture can condense in it, and thus lower its insulating efficiency. Air currents also tend to carry moisture from the hot side and condense it on the cold side.

CALCULATION OF HEAT TRANSMISSION.—The formulas given under Mathematics of Heat Insulation apply. Due to the small temperature range involved, variations with temperature of the coefficients of conductivity and the surface emissivity usually are neglected. These values usually have been determined at room temperatures, and will be smaller for lower temperatures so long as the materials remain dry. It is probable, however, that the greater amount of moisture present at the lower temperatures under service conditions more than offsets the decreased temperature gradient. Practical problems usually involve compound walls of such thickness that the surface coefficient has little influence on the result. The calculation of transmission through the walls does not include transfer of heat due to air leakage, infiltration, etc.

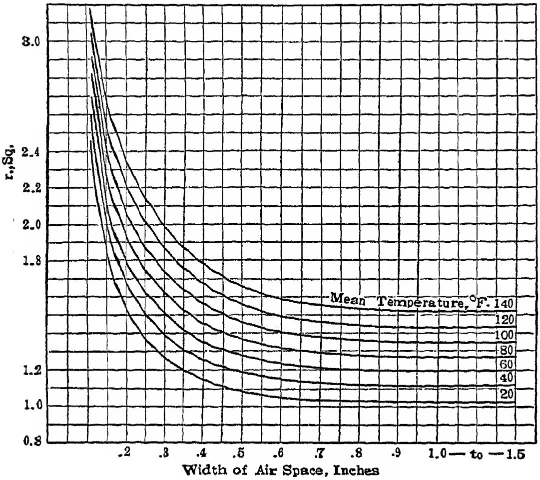


FIG. 4. Conductance of Air Spaces

The surface transfer coefficient will be lower on the cold than on the hot side. Table 3 gives average values for large areas for typical hot and cold temperatures.

AIR SPACES.—For air spaces in ordinary building construction, see Fig. 4 (F. B. Rowley, Univ. of Minn. *Bull.* No. 8). It shows the conductance C_s, B.t.u. per hr. per sq. ft. per deg. F. difference in temperature between surfaces. Heat flow values are larger than in Table 2, but will allow for leakage and rougher construction.

Increased application of air spaces as insulation has resulted from the availability of aluminum foil to provide surfaces of low emissivity, and the possibility of using a large number of sheets in a given thickness to reduce to a small value the transfer by radiation. Sheets as thin as 0.00028 in. are used and weight of the aluminum is low. Insulating value varies with the method used to separate sheets and maintain air spaces. Spacing strips produce the highest thermal resistance for a given space width; corrugated paper or asbestos and fibrous materials give good separation and some mechanical strength. Due to the thinness of the sheets there is little metallic conductance when spacing is

Table 3.—Average Values of Surface Coefficients *h*

(B.t.u. per hr. per sq. ft. per deg. F. difference)

Condition	Vertical Surface	Horizontal Surface: Hot Side Up or Cold Side Down	Horizontal Surface: Hot Side Down or Cold Side Up
HOT SIDE, 70° F.			
Building materials, still air	1.2	1.7	1.1
Building materials, with wind	3.0	...	...
Building materials, with wind and wet	4.0	...	...
Bright metal surface	0.8	1.1	0.5
COLD SIDE, 20° F.			
Building materials, still air	1.1	1.4	0.8

obtained by crumpling or embossing them, but the insulating value of crumpled sheets is inferior to corrugated structures. Insulating value depends on maintenance of the brightness of the foil; the surface does not tarnish easily under normal conditions. R. B. Mason (*Ind. and Eng. Chem.*, vol. xxv, 1933, p. 245) gives tests of various types of air spaces using aluminum foil in many forms; the above general conclusions are based on results of these tests.

THERMAL CONDUCTIVITIES OF BUILDING AND INSULATING MATERIALS.—Values found by investigators vary considerably, but as the only definition of the materials is by name and density, such differences can be expected. Even with the same density a material may have differences in physical qualities that will affect its conductivity, *e.g.*, temperature at which measurements are made, amount of contained moisture, hygroscopic qualities, and difficulties of making such tests. This applies in particular to such materials as brickwork and concrete. Standard insulating materials, especially if waterproofed, can be depended on to have a more constant value for the same density than can building materials. The true insulating value of a wall, particularly one ex-

Table 4.—Heat Conductivities of Various Materials for Low Temperature Conditions

Material	Conditions	Weight, lb. per cu. ft.	Mean Temp. Diff., deg. F.	B.t.u. per hr. per sq. ft. per in. per deg. F.	Authority
	Woods				
Balsa wood.	Light, treated, across grain.	8.0	90	0.35	Bur. of Stds
	Medium, across grain......	8.8	90	.38	" "
	Heavy, " "	20.0	90	.58	" "
Cypress......	Across grain...............	29.0	86	.67	" "
White Pine...		32.0	86	.79	" "
Virginia Pine.		34.0	86	.96	" "
Oak.........		38.0	86	1.00	" "
Sawdust......	Various............	12.0	86	0.40	" "
Shavings.....	Various, from planer.	8.8	86	.42	" "
	Building Materials				
Brickwork.	Dry conditions..................	132.0	100	4.0	Univ. of Ill.
"	Natural conditions...............	137.0	46	5.5	Nicholls
"	Average value used...............		70	5.0	....
Concrete..	1 : 2 : 4, Stone, thin...............	143	68	9.5	Rowley
	1 : 2 : 4, Stone, 24-in. thick, installed		44	11.0	Nicholls
	Cinder type....................	110	70	5.2	Peebles
Plaster....	Gypsum........................	46	86	2.3	Bur. of Stds.
"	Cement mortar..................			8.0	Rowley
Sandstone.	Dry.	140	70	9.0	Poensgen
"	Wet...............	143	70	11.7	
Soil.......	Very dry and light..			2.5	
	Clay or sand damp.			11.0	
	Wet sub-soil.......			16.0	Hencky
	Special Products				
Celotex.........	Rigid, from sugar cane fiber........	13.5	70	0.33	Peebles
Cork board......	No artificial binder................	11.3	90	.31	Bur. of Stds.
	With bituminous binder..........	16.0	90	.35	
Ground cork......	Less than 1/16 in..................	9.4	90	.27	
Regranulated cork.	About 3/16 in....................	8.1	90	.31	
Dry zero........	Pliable, ceiba fibers between treated burlap........................	1.5	75	.24	Peebles
Flaxinum........	Felted vegetable fiber............	11.0	90	.32	Bur. of Stds.
Hair felt........	Felted cattle hair................	15.0	90	.25	
Insulex or Pyrocell	Cellular gypsum dry..............	18.0	90	.59	
Insulite.........	Pressed wood pulp, rigid...........	12.0	90	.34	
Lithboard.......	Rock wool, vegetable fibers and binder, rigid..................	12.5	90	.38	
Masonite........	Exploded wood fiber, rigid.........	19.8	75	.33	Peebles
Rock cork.......	Rock wool with binder; block.....	16.0	86	.33	Bur. of Stds
Rock wool.......	Loosely packed.................	10.0	90	.27	
	Firmly "	21.0	90	.30	
Salamander.	Cattle hair, paper and muslin cover, flexible..........................	7.6	90	.25	

posed to wind pressure, also will depend on its permeability to air. Apparent differences also may be due to confusion of terms. The heat resistance of the surface effect in some tables is included with that of the material; that is, the overall conductivity from air to air is given instead of the conductivity from surface to surface. The former decreases with the thickness of the test piece, whereas the true conductivity coefficient should be the same for any thickness of test sample.

The values given in Table 4 include most commercial insulations and are mainly taken from tests at the Bureau of Standards (Van Dusen, *Jour.* A.S.H.V.E., Oct., 1920). The tests were made by the hot plate method at a mean temperature of about 68° F.

Cork board has been largely used as a standard of reference but its thermal conductivity varies with its density and method of manufacture. According to tests by Wood and Grundhofer (Penna. State College, *Bull.* No. 30) the temperature gradient is 1% increase for every 4° F. Van Dusen gives curves of tests of cork board varying in density from 7 to 14 lb. per cu. ft., from which it can be deduced that $k = 0.161 + 0.011\,W + 0.00035\,T$, where k = conductivity coefficient, B.t.u. per hr. per sq. ft. per 1 in. thickness per deg. F.; W = specific weight, lb. per cu. ft.; t = mean temperature of the material, deg. F.

ECONOMIC THICKNESS OF INSULATION FOR REFRIGERATION.—The economic thickness for heat insulation is that thickness which will reduce to a minimum the sum of the expenses due to heat passed through it plus the expenses of preventing heat

For insulation applied to a flat surface the economic thickness is

$$X = 1.\sqrt{\frac{A(t_a - t)Fk}{B\{I + R + (100/Y) + 8.3S\}}} \qquad U \qquad [17]$$

where X = economic thickness of insulation, in.; k = thermal conductivity coefficient of insulation, B.t.u. per hr. per sq. ft. per in. thickness per deg. F. temp. difference; B = cost of insulation installed, dollars per sq. ft. per in. thickness; A = cost of one ton of refrigeration, dollars = (cost per B.t.u. × 288,000); I = percent interest allowed on investment; Y = life of insulation, years; R = yearly repair cost, as percent of investment in insulation; F = fraction of year room is in operation; t_a = average temperature, deg. F., of air on the wall side, during year or period of operation; t = cold air temperature, deg. F.; S = yearly value of 1 cu. ft. of space as taken up by insulation; U = overall thermal coefficient of heat transmission from air to air of the given thickness of wall, other than insulation, and including surface transmission coefficients of the wall and of the insulation exposed surface found in the usual way from $(1/U) = (1/h) + (1/h_w) + \Sigma(L/k_w)$, where h and h_w are surface coefficients, L is thickness of wall without insulation, in., and k_w is its coefficient.

Common values for the above factors are $I = 6\%$; $R = 2$ to 4%; $Y = 10$ to 20 years; $B = 5$ to 10 cents; $A = \$1.75$ for a small plant, to 90 cents for a large one. S usually can be neglected. Values for the other factors can be found in the tables. h and h_w each can be taken as 1.2.

For brine tank insulation t_a becomes room temperature, t is temperature of the brine, and S would be zero.

The same approximate thickness of insulation is recommended by the various manufacturers. The Armstrong Cork Co. gives the following:

Inside temp. of room, deg. F.	−20 to −5	−5 to +5	5 to 20	20 to 35	35 to 45	Over 45
Thickness of insulation	8	6	5	4	3	2

It is advantageous to make insulation thick enough to keep the temperature of the hotter side higher than the dew point of the air. This avoids condensation of moisture in the insulation, with consequent lowering of its insulation value and tendency to rot.

PREVENTION OF CONDENSATION ON COLD SURFACES.—Moisture will condense on a surface whose temperature is below that of the dew point of air in contact with it. In using insulation to increase the surface temperature, the first step is to fix

Table 5.—Temperature Difference between Dry Bulb and Wet Bulb for Various Relative Humidities, t_d, deg. F.

Relative Humidity, Percent	Air temperature, dry bulb, deg. F. 50	70	90	Relative Humidity, Percent	Air temperature, dry bulb, deg. F. 50	70	90
	Temperature Difference, deg. F.				Temperature Difference, deg. F.		
20	....	43	47	70	9.5	11	11.5
30	29	33.5	36.5	80	6.5	7	7
40	23	25.5	28	90	3.5	3.5	3.5
50	18.5	20	21.5	100	0	0	0
60	13.5	15	16				

the expected dry-bulb temperature of the air and its maximum relative humidity. Table 5 gives corresponding temperature drop t_d to dew point. If t_0 is temperature of the air, temperature of surface must be not less than $t_0 - t_d$.

If temperature of surface to be insulated will not be changed by applying insulation, as pipes carrying a cold liquid, the following formulas can be used for thickness of insulation: Let t_0 = air temperature, deg. F.; t_2 = temperature of surface to be insulated, deg. F.; t_d = drop to dew point, deg. F.; h_1 = surface coefficient, B.t.u. per hr. per sq. ft. per deg. F.; k = conductivity of insulation to be used, B.t.u. per hr. per sq. ft. per in. thickness, per deg. F.; L = thickness, in. Then

For flat surfaces. $$L = (k/h_1) \times \{(t_0 - t_2 - t_d)\}/t_d \qquad [18]$$

An average value for h_1 is 1.4; see Table 3 for more exact values.

For pipes the thickness is obtained from

$$r_2 \log_{10} r_2/r_1 = (k/2.3\, h_1) \times \{(t_0 - t_2 - t_d)/t_d\}. \qquad [19]$$

where r_1 = known radius of pipe, in.; r_2 = radius of outside insulation. A few trial values will give r_2.

For a wall exposed to air on both sides, let t_0 and t'_0 be temperatures of the air on hot and cold sides, respectively; let the wall without insulation have an overall coefficient from air to air of U, or a conductance from surface to surface of C_A, both in B.t.u. per hr. per sq. ft. per deg. F.; let h_1 and h_2 = surface coefficients on hot and cold sides respectively; k = conductivity of insulation; L = its thickness; all in same units as before. Then

$$k[\{(t_0 - \qquad \cdot 1/U], \text{ or } =. k[\{(t_0 - t'_0 - t_d)/h_1 t_d\} - \{(1/C_A) - (1/h_2)\}] \qquad [20]$$

4. INSULATION OF HOT SURFACES UP TO 800° F.

REQUIREMENTS FOR A GOOD COVERING. (Am. Rwy. Assoc. Circular No. S 111–147.)—1. Good non-conducting properties. 2. Fireproof. 3. Easily molded and of light weight. 4. Impervious to moisture. 5. Insoluble. 6. Unaffected by steam. 7. Non-corrosive. 8. Structurally strong to resist handling and vibration. 9. Sanitary and vermin proof.

STANDARD COMMERCIAL SIZES.—Block insulation is made in various sizes according to the material, but molded materials are standardized for 6 × 36 in., or 3 × 18 in., and of any thickness from 1/2 in. to 4 in.

Pipe covering is made in 36-in. lengths in sectional and segmental forms. Sectional forms are split longitudinally for ease of application, and used for all pipes up to 10 in. diam. and for larger sizes for laminated insulations. Molded covering for pipes over 10 in. diam. is supplied as segmental blocks about 6 in. wide.

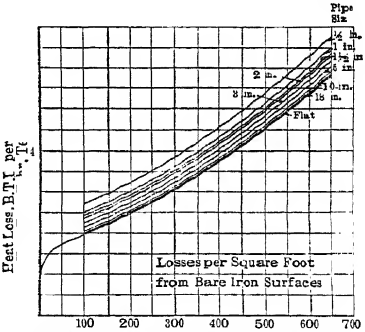


Fig. 5. Heat Loss from Bare Iron Pipe

HEAT LOSS FROM BARE HOT SURFACES.—For the principles and laws which fix the losses from any surface, see pages 3-55 to 3-59. The value of heat losses is best expressed as a given quantity of fuel wasted. It then can be translated into money values by multiplying by the unit value of fuel as fired. This applies to surfaces heated directly by the fire. If surfaces are heated by steam, hot water, or hot air, values so found must be divided by the efficiency of the boiler or other heat generator. If fuel other than coal is used, the following average conversion figures may be taken:

Fuel	Fuel oil	Natu	Producer gas	City gas
Heating value, B.t.u.	19,000 per lb.	1000 per cu. ft.	650 per cu. ft.	635 per cu. ft.
Quantity equivalent to 1 lb. coal of 13,000 B.t.u.	0.69 lb.	13 cu. ft.	20 cu. ft.	21 cu. ft.

The general laws, which were based on tests with hot surface temperatures of 150° F, and lower, developed by Peclet, were commonly used. The Mellon Institute (Heilman,

Mech. Engg., May, 1922) determined the heat loss of horizontal iron pipes of various sizes, and the surface transmission coefficients found are shown in Fig. 5. These agree closely with Peclet's laws up to about 400° F. temperature difference, after which the values found are lower. The flat surface extension below 100° F. has been added, based on Peclet's laws. The falling off of the curve is due to the convection losses being lower. This was confirmed by the experiments of Langmuir. The values are based on an air temperature of 70° F.

HEAT LOSSES FROM FLAT IRON SURFACES.—Average losses from flat vertical surfaces derived in the same way as for pipes are given in Table 6. They also are safe for use for building materials, such as brick. The actual loss will vary with height, convection loss increasing with decrease in height below 24 in. in the same manner as with

Table 6.—Heat Losses from Flat Bare Iron Surfaces, Steam and Direct Heated, per Square Foot per Hour

(Air at 70° F. Coal taken as 13,000 B.t.u. per lb. Boiler efficiency 70%)

Temp. Diff., deg. F.	Steam Gage Pressure, lb. per sq. in.	Superheat, deg. F.	Actual Surface Temp., deg. F.	B.t.u. Lost per sq. ft. per hour	Lb. of Coal Wasted per Hour: From Steam-heated Surfaces	Lb. of Coal Wasted per Hour: From Fire-heated Surfaces
50	Hot water	....	120	80	0.009	0.006
100	Hot water	....	170	190	.021	.015
110	Hot water	....	180	215	.023	.017
169	10.	....	239	375	.041	.029
200	27.3	....	270	479	.052	.037
254	80.	....	324	685	.075	.053
280	120.	....	350	798	.088	.061
300	160.	....	370	897	.099	.070
317	200.	....	387	982	.108	.076
350	160.	60	420*	1162	.128	.090
400	160.	125	470*	1480	.162	.113
418	200.	125	488*	1590	.175	.122
450	200.	165	520*	1830	.202	.142
500	200.	230	570*	2260	.250	.174
550	....	....	620	2700	.285	.200
600	....	....	670	3230	.355	.250

* Approximate; allows for temperature drop from pipe to steam.

Table 7.—Losses from Horizontal Bare Iron Steam Pipes per Lineal Foot per Hour

(Air Temp. 70° F. Coal taken as 13,000 B.t.u. per lb. Boiler efficiency 70%)

Gage Pressure	Hot Water		10 lb.		80 lb.		120 lb.		160 lb.		200 lb.		200 lb. 125° F. Superheat	
Pipe Temp.	180° F.		239° F.		324° F.		350° F.		370° F.		388° F.		488° F.	
Pipe Size, in.	B.t.u.	Lb. Coal	B.t.u.	Lb. Coal	B.t.u.	Lb. Coal	B.t.u.	Lb. Coal	B.t.u.	Lb. Coal	B.t.u.	Lb. Coal	B.t.u.	Lb. Coal
1/2	67	.008	113	.012	198	.022	228	.025	254	.028	278	.031	425	.047
3/4	80	.009	137	.015	239	.026	277	.030	310	.034	336	.037	512	.056
1	97	.011	164	.018	289	.032	330	.037	372	.041	407	.045	620	.067
1 1/4	118	.013	203	.022	357	.039	411	.045	460	.051	502	.055	779	.085
1 1/2	134	.015	229	.025	403	.044	470	.052	525	.058	574	.063	885	.097
2	164	.018	271	.030	495	.055	575	.063	643	.071	675	.071	1110	.121
2 1/2	197	.022	336	.037	590	.065	690	.075	770	.084	810	.089	1340	.145
3	231	.025	416	.045	701	.077	815	.089	900	.099	1000	.110	1590	.174
3 1/2	262	.029	449	.049	800	.088	925	.101	1030	.113	1140	.125	1810	.198
4	292	.032	500	.055	895	.098	1040	.114	1150	.126	1260	.138	2020	.222
4 1/2	322	.035	551	.060	985	.108	1140	.125	1270	.139	1400	.153	2220	.243
5	352	.039	605	.067	1070	.118	1240	.136	1390	.152	1520	.167	2490	.273
6	414	.045	716	.079	1270	.139	1480	.162	1650	.181	1820	.200	2900	.319
7	476	.052	815	.089	1470	.161	1700	.186	1890	.207	2080	.228	3340	.365
8	555	.061	920	.101	1640	.180	1910	.208	2140	.235	2320	.255	3760	.412
9	590	.065	1030	.113	1830	.201	2130	.234	2380	.261	2610	.286	4130	.452
10	652	.072	1110	.121	2030	.223	2360	.260	2640	.290	2910	.320	4520	.495
12	765	.084	1330	.146	2390	.261	2800	.306	3120	.342	3530	.387	5400	.590
14	840	.092	1460	.160	2630	.288	3060	.336	3400	.372	3740	.410	5900	.648
16	952	.104	1650	.181	2970	.325	3460	.380	3840	.420	4300	.471	6650	.730
18	1060	.116	1820	.200	3300	.361	3860	.425	4310	.474	4730	.518	7450	.816

pipes. In so far as free convection is prevented the loss will be decreased. The loss from exposed horizontal surfaces facing upwards will be greater, due to an increase of about 10% in the convection factor (Langmuir) or approximately the following percentage increase over the figures given in Table 6.

Temp. difference, deg. F.....	50	100	200	300	400	500	600
Percent increase.............	4.0	4.4	4.7	4	3.5	3	2

When the hot surface faced downward the loss was found to be less and to depend on the ability of the hot air to escape. The decrease from the values given in Table 6 may run to five to eight times the above percentages when convection currents are prevented.

HEAT LOSSES FROM BARE IRON PIPES.—Table 7 gives values of heat losses for common steam temperatures. The values are for horizontal pipes. Those for vertical pipes would be somewhat smaller. The pipes are presumed to have their natural oxidized finish.

STEAM PIPE TEMPERATURES.—Losses from hot surfaces are dependent only on surface temperature, and not on the method of heating or the fluid inside. The surface temperature may, however, be less than that of the fluid, and in general it will be. For saturated steam the difference is very small if the steam is free from air and if the pipe is well drained. With superheated steam, there will be an appreciable drop, the amount depending on size of pipe, velocity of steam and rate of loss. The drop will increase with the surface loss, and thus will be more for bare than for insulated pipe. The actual relationship is complicated, as the average temperature of superheated steam in a pipe decreases along its length, due to the loss of heat which occurs from the pipe. For ordinary insulation purposes it is sufficient to take the surface temperature as that of the nominal temperature of the superheated steam, less 20% of the superheat for bare, and less 5% for insulated surfaces (see R. Poensgen, *Zeit.* V.D.I., lx, p. 27). For long-distance transmission with superheat, the total heat loss would have to be approximated to find the drop in steam temperature.

HEAT TRANSFER COMPUTATIONS.—The fundamental formulas given under Mathematics of Heat Insulation are used, but due to the large temperature ranges, k and h (conductivity and surface transmission coefficients) cannot be considered constant except for approximate values. It is customary for manufacturers to publish conductivities at mean temperatures, the meaning of which is described under Variable Conductivities, p. 3–56. Unless a graphical method is used, computation for heat flow for a given insulation must be by trial, in which a surface temperature of the air side of the insulation is assumed. If conductivity at the mean temperature of the insulation is used, rate of heat flow should equal computed air loss; if it does not, another surface temperature is tried. Values in Table 7 can be used as a guide.

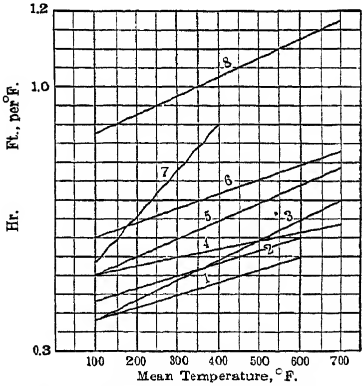


FIG. 6. Mean Conductivity of Insulation

1. Asbestos pape and sponge (sponge felt, multi-ply); 2. 85% Magnesia; 3. Rock wool; 4. High-temp. compound (High-temp., super X); 5. Molded diatomaceous earth (Non-Pareil); 6. Asbestos with hard coal (Thermalite); 7. Asbestos air cell. 8. Asbestos molded with silicate of soda (Firefelt)

For Canvas Coverings with natural or painted finish (other than aluminum paint) value of surface coefficient h can be taken as $(1.2 + 0.01\ T_d)$ for 4 in. outside diam.; $(1.3 + 0.01\ T_d)$ for 4 to 8 in.; $(1.4 + 0.01\ T_d)$ for larger sizes, where $T_d =$ difference of temperature between surface of covering and the air. Aluminum paint will reduce this value about 30%.

Many types of insulation are on the market. Values for mean conductivity usually are given by manufacturers. Fig. 6 gives average values for a few standard types.

HEAT TRANSFERENCE TABLES.—Average calculated values for the heat passing through high-grade insulating coverings for all pipe sizes and for flat surfaces are given in Tables 8 and 9. Intermediate temperatures can be interpolated. The values are actually for 85% Magnesia but they are sufficiently close to those of other high-class insulations such as felted asbestos, or Nonpareil infusorial coverings to be used for them. Naturally, all commercial insulations show some variations for different samples, depending mainly on the specific density of the material.

The tables are calculated for commercial thicknesses of the 1907 Manufacturers' List. Conductivity coefficients used are as follows, temperatures being actual and not temperature differences:

Temperature, deg. F......	100	200	300	400	500	600	700
Conductivity.............	0.5	0.52	0.55	0.583	0.62	0.66	0.71

The value of k used is approximately $k = 1.35 + 0.01(t_2 - t_0)$.

ECONOMIC THICKNESS OF INSULATION.—Computations for economic thickness of insulation for pipes involve trial and error or graphical methods. Since they include uncertainties of predictions for value of the heat and life of insulation, great accuracy usually is not required. Approximate values can be obtained from Fig. 7, which was originated by L. B. McMillan. The chief uncertainty is the choice of value for mean conductivity, but temperature drop from the surface of insulation to air as given in Table 7 can be used as a guide to surface temperature of insulation.

UNDERGROUND INSULATION OF STEAM MAINS.—Exact calculations of loss of heat from pipes buried in the ground includes the heat absorption power of the soil, and its change of temperature. It also is affected by intermittent operation, the seasons, and the rainfall. Prof. J. Allen (*Jour.* A.S.H.V.E., May, 1920) covers the mathematics of buried pipes on the assumption that constant conditions have been reached, and shows that: 1. Heat lost per sq. ft. of pipe decreases with increasing diameter. 2. As the soil acts as an insulator, the advantages of an increase in thickness of insulation is not as great as for a pipe in air. 3. Little increase in insulating effect is gained by burying pipe more than 2 ft. deep. 4. Nature and dampness of soil largely will affect results independent of the special covering.

The general systems of underground steam pipes and the insulation associated with them are: 1. The insulated pipe is buried in the ground without additional construction other than loose stone drainage under the pipe if soil is very damp. The pipe is insulated as for overhead construction, covered with overlapping layers of tar paper sealed with hot asphalt. Insulation must be able to sustain weight of the soil. 2. Conduit systems, where pressure of the soil is taken by the conduit; such systems always have efficient drainage. The conduit is laid in rubble, which, if necessary, contains drainage pipes. Some types of conduits are: *a.* Impregnated wood ducts; pipes are insulated in the regular manner, supported in center of the duct, and thus have the additional insulating value of the air space; *b.*

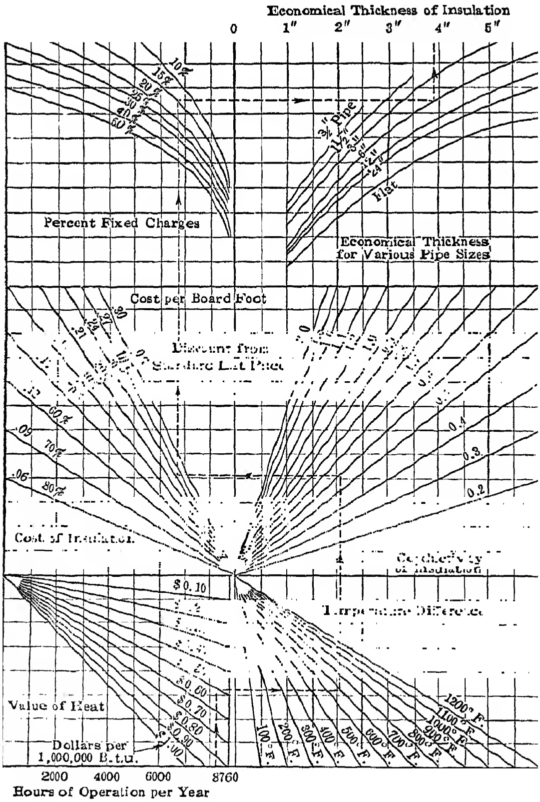


FIG. 7. Economical Thickness of Insulation

Table 8.—Losses and Surface Temperatures of High-class Insulations on Flat Surfaces

Columns *A*—Temperature of the surface of insulation above that of the air, deg. F.
Columns *B*—Losses through the insulation in B.t.u. per hour per square foot

Temp. Diff. Surface to Air, deg. F.	Actual Surface Temp. Air = 70° F.	Steam Gage Pressure, lb.	Thickness of Insulation 1 1/2 in.	2 in.	2 1/2 in. *B*	3 in.	4 in.	5 in.
50	120		18	15	12		7	5
100	170		38	27	21		15	10
120	200		50	36	30		19	13
169	239	10	67	48	38		25	17
200	270	27	79	56	44		30	20
254	324	80	105	75	58		39	25
280	350	120	116	82	64		44	28
300	370	160	126	88	68		48	30
317	387	200	134	94	72		51	32
350	420	300	147	105	79		56	35
400	470		173	121	93		65	40
418	488		182	128	99		67	42
450	520		200	142	109		75	47
500	570		228	158	122		85	53
550	620		255	177	135		95	59
600	670		286	200	152		105	66

Rectangular ducts built in a trench; ducts may be all concrete or concrete top and bottom with tile or brick sides. See Fig. 8. *c.* Split molded tile conduit with sealed joints. Fig. 9 shows various types. 3. Man size tunnels; method of insulation is similar to that of overhead construction, but there may be some gain if air temperature is higher than normal.

How much heat loss is reduced by burial of pipe or conduit depends on dampness of the soil, and on whether dampness is maintained and heat carried away by water perco-

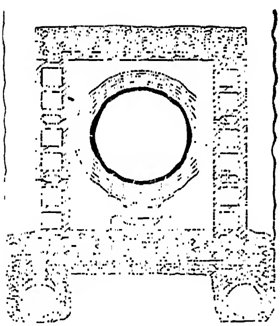


FIG. 8. Rectangular Pipe Duct

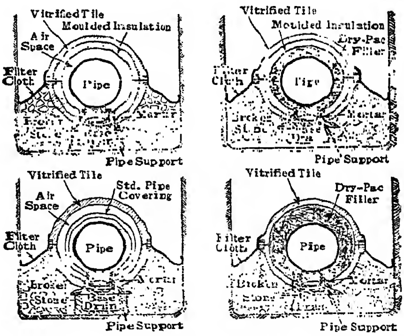


FIG. 9. Methods of Insulating Buried Pipe

lating through the soil. The most important factor is to keep the insulation dry, whether moisture comes from the soil or steam leaks. Insulation should not be damaged by wetting and subsequent drying; water vapor produced by drying should be able to escape. If loose packing is used, it should not sag if wetted, and should be able to stand vibration if buried below traffic lanes. Insulation with low water-absorbing properties is desirable. The effect on insulation of movement of the pipe, due to expansion, should be considered.

Exact computations for heat loss must depend on assumptions for dryness and gradual heating up of surrounding conduits and soil. The thickness of insulation for suspension in air sometimes is computed, and its thickness then decreased 1/2 in.

Insulation Efficiency as used for underground pipes is determined as follows: Compute the heat loss of the same length of bare pipe at the same steam temperature, suspended in air; subtract the actual loss as measured by test; the difference, divided by bare pipe loss, multiplied by 100 is the efficiency.

5. HIGH-TEMPERATURE AND FURNACE WALL INSULATION

The high-temperature field includes conditions where the temperature is greater than, say, 1000° F., as in furnaces, boiler walls, kilns, stills, ovens, etc. It involves insulation of surfaces at high temperature, or deals with heat transfer through refractories with or without additional better heat-insulation material. A good refractory material is essentially strong and dense, and consequently is a good conductor of heat; to prevent excessive heat loss it must be made very thick, or a material of lower conductivity must be built into the wall, or placed on the cool side of the refractory. The large area of the outer

Table 9.—Heat Losses through High-class Insulations

Nominal Pipe Size, in.	Thickness of Covering, in.	Temp. Difference, deg. F., between Air and Hot Surface													
		100	130	169	200	254	280	300	317	350	400	450	500	550	600
		Actual Hot Surface Temp., deg. F.; Air = 70° F.													
		170	200	239	270	324	350	370	387	420	470	520	570	620	670
		Equivalent Steam Gage Pressure, lb. per sq. in.													
				10	27	80	120	160	200	300					
		B.t.u. per hour per lineal foot													
3/4	S— 7/8	19	28	37	44	57	64	69	74	83	96	110	125	139	156
	1 1/2	16	23	29	35	46	52	55	58	65	75	86	98	109	122
	D—1 3/4	15	21	27	33	43	48	52	55	61	72	81	91	102	114
	2	14	20	26	31	40	45	49	53	57	67	76	86	96	107
	3	12	17	22	26	34	38	41	44	48	57	65	73	82	91
1	S— 7/8	23	32	42	50	65	72	78	83	93	107	124	141	157	175
	1 1/2	18	25	33	39	47	57	62	65	73	85	97	110	123	137
	D—1 3/4	17	24	31	37	48	53	57	61	70	79	90	102	114	128
	2	16	22	29	34	45	50	54	57	64	74	84	94	106	117
	3	14	19	24	29	38	42	45	48	54	62	71	80	90	100
1 1/4	S— 7/8	29	38	50	59	78	87	94	100	111	129	149	169	188	211
	1 1/2	21	29	38	46	59	67	71	76	84	98	112	127	142	158
	D—1 3/4	19	27	35	42	55	61	66	70	78	91	104	118	131	146
	2	18	25	32	39	51	56	61	64	71	83	95	108	120	134
	3	17	21	27	32	42	46	50	53	59	69	79	89	99	111
1 1/2	S— 7/8	31	42	55	65	85	95	103	109	122	141	163	185	207	237
	1 1/2	24	32	42	50	64	71	77	82	91	106	122	137	154	172
	D—1 3/4	22	29	38	47	60	66	71	76	84	98	112	127	141	158
	2	20	27	36	43	56	62	67	71	79	92	105	121	134	148
	3	17	23	30	35	46	51	55	58	65	75	86	97	109	121
2	S—1 1/32	33	45	58	70	90	101	109	115	129	150	172	196	218	244
	1 1/2	27	36	47	56	74	82	88	94	105	121	139	158	177	197
	D—2 1/16	23	30	40	47	62	69	74	79	88	101	116	132	147	164
	3	19	25	33	40	51	57	62	65	73	84	97	109	122	136
2 1/2	S—1 1/32	38	51	67	80	104	116	125	133	148	173	199	224	252	282
	1 1/2	31	41	54	64	84	94	101	107	119	139	159	180	202	224
	D—2 1/16	26	34	45	54	71	78	84	89	99	115	132	150	167	186
	3	21	28	37	44	58	64	69	73	81	94	108	122	136	151
3	S—1 1/32	44	60	77	92	120	134	144	153	172	199	230	260	291	325
	1 1/2	36	48	62	73	97	107	116	123	137	160	182	207	231	258
	D—2 1/16	30	39	51	61	80	88	96	102	113	132	151	170	190	212
	3	24	32	42	52	65	72	77	82	91	106	121	137	153	170
3 1/2	S—1 1/32	49	67	86	102	134	150	161	171	192	222	257	290	325	363
	1 1/2	39	52	69	81	106	118	127	135	151	176	202	233	255	285
	D—2 1/16	33	43	56	67	88	98	105	111	124	144	165	187	208	232
	3	26	35	46	57	71	78	84	89	100	106	132	150	167	186
4	S—1 1/8	32	69	89	114	140	155	167	178	198	231	266	288	336	376
	1 1/2	43	57	75	88	117	129	140	148	164	192	220	250	278	310
	2	37	48	62	74	97	108	116	123	137	159	183	207	230	257
	D—2 1/4	33	44	58	69	90	100	108	114	128	147	170	192	214	239
	3	28	37	49	58	76	84	90	96	107	124	142	161	179	200
4 1/2	S—1 1/8	35	74	97	116	152	169	183	194	216	252	289	328	366	410
	1 1/2	47	62	81	96	126	140	151	160	178	208	238	260	300	338
	2	39	52	67	81	105	117	126	133	149	173	198	224	250	279
	D—2 1/4	37	48	63	75	98	108	117	124	138	160	183	208	232	258
	3	31	41	53	63	82	91	98	104	115	134	153	174	194	215
5	[illegible]	[illegible]	[illegible]	106	128	165	[illegible]	[illegible]	211	235	274	315	358	400	438
	1 1/2	51	67	88	118	137	153	165	175	194	226	509	297	328	367
	2	42	57	73	87	114	126	136	144	161	187	214	242	270	302
	D—2 1/4	39	52	67	81	105	117	126	134	149	172	198	223	250	278
	3	33	43	57	67	88	97	105	112	124	144	164	186	208	232

surfaces of furnaces involves a great total heat loss, with consequent waste of fuel, uneven temperatures, undue heating of surroundings and conduction of heat to floors.

The high-temperature field of heat insulation has received much attention in recent years (1935), both in design of structures and in developing insulations to withstand higher temperatures without deterioration, and refractories of lower heat conductivity. In addition, industrial process developments, particularly in the oil industry, have made it profitable to prevent losses, because of the higher value of heat or because of more uniform operation resulting from decreased heat loss.

CALCULATION OF HEAT TRANSFER.—The formulas given under Mathematics of Heat Insulation apply. Due to the large ranges in temperature, the change of conductivity with temperature cannot be neglected in exact computations. Great accuracy in computation, however, usually is unwarranted for furnaces unless there is full knowledge of temperatures on the hot side of the walls. These temperatures often are not closely predictable, vary over the area of the wall, and are not constant. Other factors affect

Table 9.—Heat Losses through High-class Insulations—*Continued*

Nominal Pipe Size, in.	Thickness of Covering, in.	Temp. Difference, deg. F., between Air and Hot Surface													
		100	130	169	200	254	280	300	317	350	400	450	500	550	600
		Actual Hot Surface Temp., deg. F.; Air = 70° F.													
		170	200	239	270	324	35	370	387	420	470	520	570	620	670
		Equivalent Steam Gage Pressure, lb. per sq. in.													
				10	27	80	120	160	200	300					
		B.t.u. per hour per lineal foot													
6	S—1 1/8	69	91	120	145	187	209	224	239	267	310	357	405	453	505
	1 1/2	57	76	100	119	156	173	187	199	220	256	294	341	373	415
	2	48	64	83	99	129	144	155	164	183	212	243	276	307	343
	D—2 1/4	45	59	76	91	119	132	142	151	168	195	224	253	282	315
	3	37	49	64	76	99	109	118	126	140	162	186	210	234	261
7	S—1 1/4	74	97	127	152	180	219	237	252	279	327	374	425	475	531
	1 1/2	65	86	113	133	174	194	211	222	248	286	331	374	418	466
	2	53	70	92	100	144	160	173	183	203	236	271	308	343	383
	D—2 1/2	45	61	79	94	124	137	147	157	175	202	232	262	294	327
	3	40	53	70	83	108	120	130	137	153	178	203	230	258	286
8	S—1 1/4	80	108	141	168	219	243	263	278	310	363	415	477	529	590
	1 1/2	73	96	125	149	195	217	236	248	276	322	369	418	465	520
	2	58	77	102	121	158	176	190	201	224	260	298	338	378	421
	D—2 1/2	50	67	80	104	136	151	163	173	193	224	256	290	324	361
	3	46	59	77	92	120	133	144	152	170	196	225	254	284	316
9	S—1 1/4	89	120	156	187	243	271	292	310	345	404	462	525	589	655
	1 1/2	78	105	137	159	212	236	256	270	302	349	401	456	509	567
	2	64	85	112	133	173	193	207	220	244	284	326	370	412	460
	D—2 1/2	54	73	88	113	148	165	178	189	211	244	279	316	353	394
	3	48	64	84	100	131	145	157	166	185	214	246	278	310	346
10	S—1 1/4	98	132	171	205	267	298	321	342	380	444	510	579	647	723
	1 1/2	86	115	151	179	234	261	283	298	332	385	442	504	561	626
	2	71	94	123	146	191	212	228	242	269	313	359	407	455	507
	D—2 1/2	60	80	96	124	161	179	194	206	230	266	301	345	384	429
	3	52	70	92	109	141	158	170	181	201	234	267	303	337	376
12	S—1 1/2	100	133	175	208	272	303	325	346	386	447	515	585	653	728
	2	82	108	142	169	219	244	263	279	310	361	413	470	525	585
	3	60	80	106	125	162	181	196	208	232	269	306	347	387	432
	4	49	65	85	101	132	146	158	168	186	216	247	280	312	348
14 in. O.D.	1 1/2	108	143	189	225	293	326	351	373	417	484	555	631	705	785
	2	89	116	153	182	238	264	284	302	335	390	447	508	567	632
	3	65	87	114	135	175	196	212	224	250	290	332	375	418	467
	4	53	70	92	110	143	158	170	181	201	234	268	303	338	376
16 in. O.D.	1 1/2	123	163	214	255	333	370	398	424	473	550	630	715	800	890
	2	99	131	170	203	265	295	318	338	375	436	500	566	634	706
	3	73	97	127	151	198	220	237	252	281	328	372	421	470	525
	4	59	78	102	122	158	175	188	200	223	258	298	326	374	416
20 in. O.D.	1 1/2	155	205	270	358	419	465	500	534	595	690	794	898	1010	1140
	2	125	164	214	254	333	370	398	423	470	547	626	710	795	895
	3	89	118	155	184	240	267	288	308	341	396	452	512	551	638
	4	72	95	124	149	177	213	230	244	271	316	361	408	455	507
24 in. O.D.	1 1/2	182	240	316	420	490	546	588	625	697	810	934	1060	1180	1320
	2	146	191	250	297	390	433	465	495	550	641	734	831	930	1035
	3	103	137	179	214	279	310	334	355	394	458	525	594	640	739
	4	83	109	143	170	221	245	264	281	311	363	415	470	525	584

heat flow and may affect materially the accuracy of computed, as compared with actual, heat flow. These are: *a*. Uncertainty of conductivity of refractories. *b*. Effect of slag erosion, slag accumulations, or contamination of the refractory by slag. *c*. Deterioration of wall structure. In many instances, however, exact computation is warranted.

VALUES OF THE SURFACE COEFFICIENT.—Relatively few data exist from which to derive a value of h for surfaces exposed to flame. It usually is sufficient for most purposes to base computations on an assumed temperature of the hot surface. At furnace temperatures, the main part of the transfer is by radiation; for a full treatment of this subject see W. H. McAdams, Heat Transmission, chap. iii.

Table 10 gives values of h for the surface of the wall exposed to air. Values are for vertical walls of iron, brick or other building materials. Table 10 also gives the B.t.u. dissipated per hr., *i.e.*, $h \times$ temperature difference. The total loss in fuel per sq. ft. per year of 300 days (7200 hr.) with coal of 13,000 B.t.u. per lb. is shown. This is the fuel loss with direct-fire heat. If it is indirect, as in steam heat, the equivalent pounds of coal loss is increased by dividing the values by furnace efficiency.

THERMAL CONDUCTIVITIES OF REFRACTORIES.—Values for thermal conductivities of refractories are not well defined, because of variations in properties of products bearing the same name, and because experimental work is difficult and incomplete. Values obtained by some investigators show that the change of conductivity with temperature does not follow the straight line law $k = a + bt$, and therefore mean conductivity (see Variable Conductivity, p. 3–56) will depend on the range of temperature covered. However, the mean method is more convenient and values for the true conductivities are so uncertain that values in Table 11 are on the mean conductivity basis. They are from experiments and compilations by F. H. Norton. No claim is made that these exact values are true for all bricks of the same name.

For approximate calculations in heat insulation problems for average furnace conditions, the following mean conductivity values for k can be used in B.t.u. per hr. per sq. ft. per deg. F. per in. thickness.

Kind of Brick..............	Common	Fire	Silica	Magnesia
k.....................	6	10	12	30

THERMAL CONDUCTIVITIES OF INSULATION.—If insulation is placed outside of a refractory structure the maximum temperature to which it will be subjected may fall within the range of materials used in the steam temperature field; for conductivities of such materials, see p. 3–65. Table 12 gives conductivities and other data on materials

Table 10.—Heat Losses from Metal-encased Furnaces
(Air Temperature 70° F.)

Actual Surface Temperature, deg. F.	Temperature Difference, deg. F.	Heat Loss per sq. ft.: Surface Coefficient, B.t.u. per deg. F. per hr.	Heat Loss per sq. ft.: B.t.u. per hour	Heat Loss per sq. ft.: Lb. of Coal per Year of 300 Days	Actual Surface Temperature, deg. F.	Temperature Difference, deg. F.	Heat Loss per sq. ft.: Surface Coefficient, B.t.u. per deg. F. per hr.	Heat Loss per sq. ft.: B.t.u. per hour	Heat Loss per sq. ft.: Lb. of Coal per Year of 300 Days
70	0	....	0	0	240	170	2.24	380	210
80	10	1.35	13	7	260	190	2.35	446	247
100	30	1.48	44	25	280	210	2.46	516	286
120	50	1.58	79	44	300	230	2.57	590	326
140	70	1.70	119	66	320	250	2.68	670	370
160	90	1.83	165	91	340	270	2.80	755	418
180	110	1.96	215	119	360	290	2.92	845	468
200	130	2.05	266	147	380	310	3.04	940	520
220	150	2.13	320	177	400	330	3.17	1040	575

Table 11.—Conductivity of Refractories
(B.t.u. per hr. per sq. ft. per in. thickness per deg. F.)

Material	Mean Temperature, deg.: 400	600	800	1000	1200	1400	1600	1800	2000	2200	2400	2600
Firebrick (Missouri)	7.0	7.5	8.0	8.5	8.9	9.3	9.5	9.8	10.1	10.3	10.6	10.8
Firebrick (Penna.)..	7.0	7.9	8.7	9.5	10.1	10.6	11.0	11.3	11.6	11.9	12.2	12.5
Chrome...........	9.8	10.3	10.7	11.1	11.4	11.6	11.7	11.8	11.9	12.0	12.1	12.2
Silica.............	8.2	9.3	10.3	11.2	12.0	12.5	13.2	13.8	14.4	14.9	15.5	16.0
Zirconia............	10.2	10.8	11.4	11.9	12.3	12.7	15.0	15.3	15.6	15.9	16.2	16.5
Magnesite..........	40.0	38.0	36.0	34.0	32.0	30.0	28.5	27.0	26.0	25.5	25.0	25.0
Fused alumina......	18.0	20.0	21.8	23.0	24.0	25.0	26.0	27.0	28.0	29.0	30.0	31.0
Silicon carbide.....	....	....	26.0	25.0	24.0	23.1	22.2	21.6	21.0	20.4	19.8	19.2

able to stand higher temperatures, which values were obtained at the Mellon Institute by R. H. Heilman. Results obtained by different investigators lack agreement, even for such homogeneous materials. This statement applies particularly to conductivities at higher temperatures, but the values of Table 12 have been determined by the same method and are true relatively. Maximum temperatures are those recommended by the manufacturers. The materials listed are supplied in standard 9 in. size, and some also in blocks or special shapes.

TYPICAL WALLS.—Table 13 gives heat flow and heat content of typical walls. Construction is assumed to be firebrick and Sil-O-Cel insulation. Relative changes in values with other materials can be determined from Tables 11 and 12, but these will not be great.

APPLICATION OF INSULATION.—The decision of whether it will pay to apply insulation to a furnace cannot be made on the economic basis alone. It is an engineering problem for each class of furnace and the conditions of its use, and in particular whether the use is continuous or intermittent. Following are some main considerations and precautions necessary: *a.* Inclusion of insulation must not risk weakening the structure. *b.* Thickness of insulation, and its location in the walls, must be such that maximum temperature in insulation will not exceed the limiting temperature specified for it; it is important that this maximum temperature be that which will obtain under the most severe conditions of operation, and not the average value. *c.* Addition of insulation usually will not materially increase the temperature of the hot face of the refractory, but it will decrease drop in temperature through it, *i.e.*, increase its average temperature. If furnace temperature is so high that refractory temperature is near the melting point, or where rapid erosion by slagging will occur, it is necessary to consider whether or not the life of the refractory may be shortened. *d.* Insulation exposed to weather should be well waterproofed, particularly for furnaces which may be shut down in winter, when insulation may be damaged by freezing of moisture in it. *e.* If a furnace operates intermittently, the effect of heat stored in the walls is important. If it must be cooled between runs, it is desirable that insulation be as near the hot side as allowable; if temperature is to be maintained between runs, insulation can be placed on the outside.

Table 12.—Thermal Conductivities of High-temperature Insulation
(B.t.u. per hr. per sq. ft. per in. thickness per deg. F.)

Material	Wt. per cu. ft., lb.	Max. Temp., deg. F.	Mean Temperature, deg. F. 400	600	800	1000	1200	1400	1600	1800	2000	2200	2400
Sil-O-Cel, natural, good grade........	29	1600	0.66	0.76	0.84	0.93	1.01	1.09	1.17				
Sil-O-Cel, natural, laminated........	29	1600	0.92	1.00	1.09		1.25	1.33	1.42				
J. M. Co., C–22 *...	36	2000	1.28	1.43	1.59		1.90	2.06	2.22	2.38	2.54		
J. M. Co., Super Sil-O-Cel †........	44	2500	1.67	1.81	1.96	2.10	2.25	2.40	2.54	2.70	2.84	2.98	3.12
P. C. Co., Alumino ‡.	25	1850	0.59	0.65	0.72	0.78	0.85	0.92	0.98	1.05			
B. and W. No. 80 insulating §.........	34	2800	1.25	1.36	.46	1	1.68	1.78	1.89	2.00	2.11	2.22	2.32
A. C. Co., Non-Pariel	30	1600	0.90	0.97	.07	1	1.25	1.34	1.42				
A. C. Co., Armstrong	36	2500	1.18	1.43	.68	1	2.16	2.40	2.65	2.90	3.13	3.38	3.62
Corundite, L.W.–10 ¶	48	2600	1.64	1.79	.94	2	2.25	2.40	2.75	2.70	2.85	3.00	3.16

* Calcined Sil-O-Cel. † Calcined semi-refractory brick. ‡ High-porosity alumina made from bauxite, chemically processed. § High-porosity kaolin. ¶ Bloated clay mixture.

Table 13.—Heat Loss through, and Heat Content of Walls
(Flow = B.t.u. per hr. per sq. ft.; H.C. = Heat content, 1000 B.t.u. per sq. ft.)

Thickness, in.			Hot Face Temperature, deg. F. 1600		2000		2400		2800	
Total	Brick	Insulation	Flow	H. C.	Flow	H. C.	Flow	H. C.	Flow	H. C.
13 1/2	13 1/2	0	770	35	1060	42	1400	50	1800	58
13 1/2	9	4 1/2	400	37	570	47	750	57	940	66
18	18	0	600	43	830	53	1080	64	1340	75
18	13 1/2	4 1/2	340	52	500	65	660	78	830	91
18	9	9	260	45	380	57	510	68	650	80
22 1/2	22 1/2	0	440	52	610	65	780	78	970	90
22 1/2	18	4 1/2	340	64	490	80	660	97	830	113
22 1/2	13 1/2	9	230	57	330	72	440	86	550	100
27	18	9	200	73	290	91	390	110	490	130

THERMODYNAMICS

Revised by A. G. Christie

1. THERMODYNAMIC LAWS

THERMODYNAMICS, the science which deals with the transformations of energy, is useful in studies of steam, gas and air engines, steam and gas turbines, air and gas compressors, refrigerating machines, etc. The works of Goodenough, Lucke, Barnard, Ellenwood and Hirshfeld, Kiefer and Stuart, Ennis, Ewing, Dalby, Ripper, and Emswiler are among the standard textbooks on Thermodynamics.

ENERGY may be defined as the capacity for producing an effect. Energy may exist in stored forms as the kinetic energy of a moving body, the potential energy of an elevated or elastically strained body, the internal molecular energy in a hot substance, the chemical energy in fuel or a charged storage battery, etc.; it may be in transition as work due to the action of a superior force, as heat transfer due to a superior temperature, or as electrical energy due to a superior electrical potential.

Kinetic Energy is possessed by a body when work can be done as a result of a change in velocity. Thus if a body of weight w has a velocity of V_a, its kinetic energy $= wV_a^2/2g$. If the velocity is changed to V_b in the process of doing work, then the work done $W = w\,(V_a^2 - V_b^2)/2g$.

Internal or Intrinsic Energy is the energy stored in a body above some datum state, due to its molecular movements and arrangements or other internal effects. This quantity will be denoted by U.

WORK is the action of a force in displacing a resistance through a distance. It is measured by the product of the force multiplied by the distance moved in the direction of the force. (Barnard, Ellenwood, and Hirshfeld).

UNITS OF ENERGY.—The conventional units of energy are derived from simple phenomena associated with the transition of energy. They are:

a. **The Foot-pound** (ft.-lb.), which is the amount of energy (exhibited as work) associated with the action of a force of 1 lb. applied through a distance of 1 ft.

b. **The British Thermal Unit** (B.t.u.), which is 1/180 of the energy supply required to raise the temperature of a pound mass of pure water from 32° F. to 212° F. at a constant atmospheric pressure of 14.696 lb. per sq. in. absolute.

c. **The Horsepower-hour** (Hp.-hr.), which is the transition of energy at the rate of 33,000 ft.-lb. per min., or 550 ft.-lb. per sec.

d. **The Kilowatt-hour** (kw.-hr.), which is the transition of energy at the rate of 1000 watts per hour or 1.341 Hp.-hr., or 44,253 ft.-lb. per minute.

The relative magnitudes of the several units are: 1 B.t.u. = 778 ft.-lb.; 1 Hp.-hr. = 2543 B.t.u.; 1 kw.-hr. = 3411.5 B.t.u. Values for the Hp.-hr. and kw.-hr. are based on the recommendations of the International Steam Tables Conference, 1929.

THE ENTHALPY (formerly called *Total Heat* or *Heat Content*) of a unit weight (1 lb.) of a substance is the sum of its internal energy U, in B.t.u. per lb., above some datum and the product, Pv_s, of its pressure and volume at the given state. Enthalpy in B.t.u. per lb. $= h = U + A\,P\,v_s$, where $A = 1/778.6$; P = absolute pressure, lb. per sq. ft.; and v_s = specific volume of 1 lb. in cu. ft.

THE FIRST LAW OF THERMODYNAMICS is simply a statement of the principle of the conservation of energy, namely, that energy may exist in many varied and interchangeable forms but may not be quantitatively destroyed or created.

ENERGY EQUATIONS.—If a unit weight of 1 lb. of a substance flowing steadily through a heat engine passes from initial state 1 to final state 2, the energy equation covering the general case is $E_m = h_1 - h_2 + Q_h + \dfrac{(V_1^2 - V_2^2)}{778.6 \times 2g} + \dfrac{(Z_1 - Z_2)}{}$

where E_m = amount of mechanical energy furnished by the machine, B.t.u. per lb.; h_1 and h_2 = enthalpy (or total heat) of 1 lb. of the substance at state 1 and state 2 respectively; V_1 and V_2 = velocities of substance, ft. per sec., passing the sections at states 1 and 2, respectively; Z_1 and Z_2 = elevations of sections at states 1 and 2, respectively, above datum, measured in ft., of the substance; g = gravity = 32.16; Q_c = loss of heat by conduction and radiation between states 1 and 2, B.t.u. per lb. of the working substance. Q_h = heat added to (+) or taken from (−) working substance in the heat engine, as by jackets or by cooling water, B.t.u. per lb. of working substance.

In an engine or turbine without jackets $Q_h = 0$, if $V_1 = V_2$, and $Z_1 = Z_2$; then $E_m = h_1 - h_2 - Q_c$.

In a perfectly heat insulated engine $Q_c = 0$ and the work done, $E_m = h_1 - h_2$.

ABSOLUTE TEMPERATURE.—Temperatures are measured in degrees Fahrenheit. Absolute zero is at $-459.6°$ F. Absolute temperature $= t°$ F. $+ 459.6$, where $t =$ observed temperature on a thermometer which has 180 divisions between the freezing temperature and the boiling point of pure water.

THE SECOND LAW OF THERMODYNAMICS has been stated in a variety of ways by different writers. This law embodies the idea of availability of energy and of degradation of energy, or the tendency of high-grade energy to degenerate into low-grade energy. The following are selected statements of this law:

It is impossible for a self-acting machine unaided by any external agency to transfer heat from one body to another at a higher temperature (Clausius).

It is impossible by means of inanimate material agency to derive mechanical effect from any portion of matter by cooling it below the temperature of surrounding objects (Kelvin).

No machine, actual or ideal, can both completely and continuously transform heat into mechanical energy (Barnard, Ellenwood and Hirshfeld).

No change in a system of bodies that can take place of itself, can increase the available energy of a system (Goodenough).

If all the heat absorbed be at one temperature, and that rejected be at one lower temperature, then will the heat which is transmuted into work be to the entire heat absorbed in the same ratio as the difference between the absolute temperature of the source and refrigerator is to the absolute temperature of the source. In other words, the second law is an expression for the efficiency of the perfect elementary engine (Wood).

The expression $\frac{Q_1 - Q_2}{Q_1} = \frac{T_1 - T_2}{T_1}$ thus may be called a symbolical or algebraic enunciation of the second law—which law limits efficiency of heat engines, and which does not depend on the nature of the working medium employed (Trowbridge). Q_1 and $T_1 =$ quantity and absolute temperature of heat received; Q_2 and $T_2 =$ quantity and absolute temperature of heat rejected.

The expression $(T_1 - T_2)/T_1$ represents the efficiency of a perfect heat engine cycle which receives all its heat at the absolute temperature T_1, and rejects heat at the temperature T_2, converting into work the difference between the quantity received and rejected.

A conclusion that is of outstanding engineering significance derived from the Second Law is that the *maximum* work output obtainable from an engine equals the change of enthalpy for an adiabatic isentropic expansion of the fluid flowing through the engine.

Example.—What is the efficiency of a perfect heat engine cycle which receives heat at 428.3° F. (temperature of steam at 400 lb. gage pressure) and rejects heat at 79° F. (temperature of a condenser at 29 in. vacuum or 1 in. absolute pressure)?

$$\{(428.3 + 459.6) - (79 + 459.6)\}/(428.3 + 459.6) = 39.34\%$$

This efficiency never can be attained in an actual engine or turbine, since the energy available between the condition of the steam entering and the pressure at which it leaves is not all converted into work. Some of this available energy is dissipated in conduction and radiation, leakage, reheating effects, and in the case of reciprocating engines, in cylinder condensation and re-evaporation, all of which tend to reduce efficiency. The quantity $Q_1(T_1 - T_2)/T_1$ is available to do work in a perfect engine. The amount Q_1T_2/T_1 is unavailable under the given conditions.

REVERSIBILITY.—A process is said to be *reversible* when the following conditions are fulfilled: *a.* When the direction of the process is reversed, the system taking part in the process can assume in inverse order the states traversed in the direct process. *b.* The external actions are the same for the direct and reversed processes. *c.* Not only the system undergoing the change but all connected systems can be restored to initial conditions (Goodenough).

Any process that fails to meet these requirements is an *irreversible process*. Three irreversible processes are of frequent occurrence: *a.* The direct conversion of work into heat through the agency of friction. *b.* The conduction of heat from a body at temperature t_1 to another at temperature t_2. *c.* The throttling or wire-drawing of a fluid in flowing through an orifice from a region of higher to a region of lower pressure.

A PRESSURE-VOLUME DIAGRAM is one whose ordinates are pressures and abscissas are volumes. If a line is drawn on this diagram which represents the successive changes of state of the substance in passing from state a to state b, the area under this line represents the work done on or by the substance during this change if the state change be mechanically reversible, in that no mechanical or fluid friction or turbulence occurs.

ENTROPY may be considered as the length of a diagram, the area of which represents a quantity of energy added or removed, and the height at any point represents absolute temperature. The gain in entropy Δs for any reversible path between any two temperatures T_1 and T_2 is $\Delta s = \int_{T_1}^{T_2} \frac{dQ}{T}$. For constant temperature changes, entropy change $\Delta s = Q/T$, where $Q =$ energy added, B.t.u., per lb., and $T =$ absolute temperature, deg. F. For constant volume changes of gases remote from condensation conditions $\Delta s = c_v \log_e T_2/T_1$, and for constant pressure changes $\Delta s = c_p \log_e T_2/T_1$, where c_v and

c_p are the specific heats, B.t.u. per lb., at constant volume and pressure. T_1 and T_2 are the initial and final absolute temperatures, deg. F. Where energy is added to water at constant pressure to raise it to the boiling point, or energy is added to steam at constant pressure to superheat it, the mean specific heat at constant pressure over the given temperature range must be used.

The Temperature-entropy Diagram is one in which the ordinate or vertical line represents absolute temperature and the abscissa or horizontal line measures the change in entropy. Entropy usually is measured above 32° F. in steam tables and on these diagrams. Horizontal lines on temperature-entropy diagrams are isothermals, while vertical lines are reversible adiabatics. The employment of entropy often will simplify thermodynamic calculations by using algebraic or graphical methods instead of calculus.

An Isothermal Change is one during which the temperature of the working substance remains constant.

An Adiabatic is any process in which no heat is transferred to or from the working substance from any outside source.

A Reversible Adiabatic takes place at constant entropy, *i.e.*, an *Isentropic*. In this case no heat from an outside source is added to the working substance, nor is any heat lost to an outside body and no friction exists, either mechanical or fluid. The reversible adiabatics make available the greatest amount of energy for doing external work, and are the standards or reference in the Rankine cycle. In steam turbine calculations the ideal heat drop $(h_1 - h_2)$ per lb. of steam, and consequent maximum work, is found from an isentropic adiabatic expansion.

An Irreversible Adiabatic is a process in which there is no transfer of heat from the outside to the working substances, but in which more or less of the ideally available energy of the working substance fails to be obtained as work output, due to the presence of fluid or mechanical friction.

MOLLIER DIAGRAM.—Besides the temperature-entropy diagram, another, known as the Mollier Diagram, is widely used for vapors and gases. This diagram is plotted with enthalpy h as ordinate and entropy s as abscissa. (See p. 5–19 for a diagram for steam.) Vertical lines of constant entropy represent reversible adiabatic changes. Pressure, temperature, and volume lines can be placed on such a chart.

2. PROPERTIES OF GASES

Notation.—P = absolute pressure, lb. per sq. ft.; v = volume, cu. ft.; P_0, v_0 = pressure and volume at 32° F.; t = observed temperature, deg. F.; T = absolute temperature = $459.6 + t°$ F.; c_p = specific heat per lb. at constant pressure, B.t.u.; c_v = specific heat per lb. at constant volume, B.t.u.; K_p = ft.-lb. of energy equivalent to specific heat at constant pressure = $778.6\, c_p$; K_v = ft.-lb. of energy equivalent to specific heat at constant volume = $778.6\, c_v$; R = a constant for a given gas = $(K_p - K_v)$; $\gamma = c_p/c_v$; r = ratio of isothermal expansion or compression = P_1/P_2 or v_2/v_1 for expansion when $P_1 v_1 = P_2 v_2$ and P_1 is the higher pressure; w = lb. of working substance.

Boyle's or Mariotte's Law.—The volume of a perfect gas at constant temperature varies inversely as the absolute pressure. Thus, if P_1 = absolute pressure at volume v_1, and P_2 = absolute pressure at volume v_2, then with T = a constant, $v_2/v_1 = P_1/P_2$, or $P_1 v_1 = P_2 v_2 = C$, where C = a constant.

This law has been found by experiment to be very nearly true for all gases. For instance, air compressed to 75 atmospheres has a volume 2% less than that computed by Boyle's law, but this is the greatest divergence that is found below 160 atmospheres.

Charles's Law.—The volume of a perfect gas at constant pressure is proportional to its absolute temperature. If v_0 be the volume of gas at 32° F., and v_1 the volume at any other temperature t_1, then $v_1 = v_0 \left(\frac{t_1 + 459.6}{491.6}\right)$; v_1 $\quad \frac{32}{1.6}\Big)$;

$$v_1 \quad \{1 + 0.002034\,(t_1 - 32)\}v_0.$$

If the absolute pressure also changes from P_0 to P_1, $v_1 = \frac{P_0}{P_1}\left(\frac{t_1 + 459.6}{491.6}\right)$

The Characteristic Equation of Gases is a combination of the two preceding laws, $Pv = wRT$, where R is a constant for a given gas or mixture of gases. (See Table 3.)

Avagadro's Law.—Equal volumes of all gases under the same conditions of pressure and temperature contain the same number of molecules. Hence the densities of gases at the same pressure and temperature are proportional to their molecular weights.

THE MOL is that quantity of a gas at a given pressure and temperature whose weight in pounds equals its molecular weight. It is evident from Avagadro's Law that the vol-

ume of a mol is constant for all gases at any given pressure and temperature. At 32° F. and atmospheric pressure of 14.7 lb. per sq. in. the volume of a mol of gas = 358.7 cu. ft. This volume of any gas weighs m lb. where m = the molecular weight. (See Table 3 for molecular weights.)

The volume of 1 lb. of any gas at atmospheric pressure and 32° F. = $358.7/m$ cu. ft. The weight of 1 cu. ft. of any gas at 32° F. and atmospheric pressure = $m/358.7$ lb.

Universal Gas Constant.—Let v in the characteristic equation of a gas = 1 mol at atmospheric pressure and 32° F. On solving

$$mR = Pv/T = (2116.4 \times 358.7)/(459.6 + 32) = 1544,$$

which is called the *Universal Gas Constant*. For any gas, $R = 1554/m$. Also

$$R/778.6 = 1544/778.6\,m = 1.985/m = c_p - c_v;\ \text{or}\ R = 778.6\,(c_p - c_v) = (K_p - K_c),$$

the difference between the specific heats of 1 lb. of gas at constant pressure and at constant volume, expressed in ft.-lb.

THE SPECIFIC HEAT OF GASES.—The instantaneous molal specific heats of gases at constant pressure, *i.e.*, the specific heat of m lb. of gas at a given temperature when raised 1° F., where m = molecular weight, as given in Combustion, 3d edition (Amer. Gas Assoc., 1932, p. 59), and based on the Eastman data given in Tech. Paper 445 and Circular 6337 of the U. S. Bureau of Mines, have been selected for presentation in Table 1, although the differences from values given by other authorities are small. These molal specific heats are expressed in B.t.u. per lb.-mol per deg. F., and the temperatures in the equations are deg. F.

Instantaneous molal specific heat at constant volume $mc_v = mc_p - 1.985$. The instantaneous specific heats per pound can be found by dividing each of the molal specific heats by the molecular weight of the gas.

All the instantaneous molal specific heat equations take the general form of

$$mc_p = a + bt + dt^2 + et^3.$$

The Quantity of Heat Added to a mol of gas in raising its temperature from t_1 to t_2 at constant pressure

$$= (t_2 - t_1)[a + {}^1/_2\,b\,(t_2 + t_1) + {}^1/_3\,d\,(t_2^2 + t_2 t_1 + t_1^2) + {}^1/_4\,e\,(t_2^3 + t_2^2 t_1 + t_2 t_1^2 + t_1^3)]$$

Mean molal specific heat between t_2 and t_1 is this expression divided by $(t_2 - t_1)$, that is $a + {}^1/_2\,b\,(t_2 + t_1) + {}^1/_3\,d\,(t_2^2 + t_2 t_1 + t_1^2) + {}^1/_4\,e\,(t_2^3 + t_2^2 t_1 + t_2 t_1^2 + t_1^3)$.

The Mean Specific Heat per Pound is found by dividing the mean molal specific heat by the molecular weight m of the gas.

The instantaneous specific heats per pound derived from the molal specific heats in Table 1 and the molecular weights in Table 3, are given in Table 2.

When many computations must be made involving the enthalpy or heat contents of gases at various temperatures, the work may be lessened by means of curves. Enthalpy may be computed at certain temperatures and curves drawn on a sufficiently large scale

Table 1.—Instantaneous Molal Specific Heats at Constant Pressure, mc_p

CO_2	$= 8.999 + 0.002709\,t - (10^{-7} \times 2.56\,t^2)$
H_2O	$= 8.319 + 0.0003364\,t + (10^{-7} \times 6.879\,t^2) - (10^{-10} \times 1.054\,t^3)$
N_2, O_2, CO and air	$= 6.923 + 0.0003734\,t + (10^{-8} \times 4.01\,t^2)$
H_2	$= 6.935 + 0.000217\,t + (10^{-8} \times 6.79\,t^2)$
CH_4	$= 8.35 + 0.00533\,t$ (up to 261° F.) *
$C_nH_{(2n+2)}$	$= 4.19 + 4.29\,n + (0.00666 + 0.00333\,n)\,t$
n	= number of carbon atoms in the molecule.

* For CH_4 at temperatures above 261° F. the formula of Goodenough and Felbeck, *Bull.* 139, Engg. Exp. Sta., Univ. of Illinois, may be used, *viz.*, $CH_4 = 8.312 + 0.01056\,t$ (above 261° F.).

Table 2.—Instantaneous Specific Heats per Pound at Constant Pressure c_p *

CO_2	$= 0.2045 + 0.0000616\,t - (10^{-9} \times 5.82\,t^2)$
H_2O	$= 0.4618 + 0.00001867\,t + (10^{-8} \times 3.818\,t^2) - (10^{-12} \times 5.85\,t^3)$
N_2	$= 0.2471 + 0.00001333\,t + (10^{-9} \times 1.4311\,t^2)$
O_2	$= 0.2163 + 0.00001167\,t + (10^{-9} \times 1.253\,t^2)$
CO	$= 0.24725 + 0.00001333\,t + (10^{-9} \times 1.432\,t^2)$
Air	$= 0.2391 + 0.0000129\,t + (10^{-9} \times 1.385\,t^2)$
H_2	$= 3.441 + 0.0001077\,t + (10^{-8} \times 3.37\,t^2)$
CH_4	$= 0.5209 + 0.0003325\,t$ (up to 261° F.)
† CH_4	$= 0.5185 + 0.0006587\,t$ (above 261° F.)
C_2H_4	$= 0.4556 + 0.0004752\,t$

* Temperatures t are in deg. F. † Formula by Goodenough and Felbeck.

per mol (or B.t.u. per lb. or per cu. ft. under given conditions if needed) as ordinates and temperatures in deg. F. as abscissæ. The heat added to or subtracted from a unit quantity of a gas between any two temperatures can be found from the curve as the difference in enthalpy at the two temperatures. The mean specific heat of the unit quantity between these two temperatures also can be computed by dividing this difference in enthalpy by the temperature change.

THE INTERNAL ENERGY of a perfect gas has been shown by Joule to depend on temperature only, and to be independent of volume.

The Change in Internal or Intrinsic Energy of 1 lb. of gas

$$778\ (U_1 - U_0) = K_v\ (T_1 - T_0) = R\ (T_1 - T_0)/(\gamma - 1).$$

The total internal energy of w lb. of gas, wU_1, at pressure P_1 in lb. per sq. ft. absolute and volume v_1 in cu. ft., with $\gamma = c_p/c_v$, would be $P_1 v_1/(\gamma - 1)$ measured above absolute zero F., if the substance should follow perfect gas laws down to absolute zero and its specific heats should be constant. When air is expanded or compressed isothermally, Pv = constant, and the internal energy remains constant, the work done in expansion = the heat added, and the work done in compression = the heat rejected.

Work Done in Reversible Adiabatic Expansion, no transmission of heat, from

$$P_1 v_1 \text{ to } \quad : P_1 v_1 \{1 - (v_1/v_2)^{\gamma - 1}\} \div (\gamma - 1) = (P_1 v_1 - P_2 v_2) \div (\gamma - 1)$$
$$= P_1 v_1 \{1 - (P_2/P_1)^{\gamma - 1/\gamma}\} \div (\gamma - 1).$$

Work of Reversible Adiabatic Compression from $P_1 v_1$ to $P_2 v_2$ (P_2 here being the higher pressure) $= P_1 v_1 \{(v_1/v_2)^{\gamma - 1} - 1\} \div (\gamma - 1) = (P_2 v_2 - P_1 v_1) \div (\gamma - 1) = P_1 v_1 \{(P_2/P_1)^{\gamma - 1/\gamma} - 1\} \div (\gamma - 1)$.

Loss of Internal Energy in ft.-lb. per lb. of substance in any expansion or gain in compression $= K_v\ (T_1 - T_2)$, T_1 being the higher temperature.

Work of Isothermal Expansion, in ft.-lb. per lb. from P_1 to P_2, temperature remaining constant = heat expended $= P_1 v_1 \log_e (v_2/v_1) = P_1 v_1 \log_e r = RT \log_e r$, where $r = v_2/v_1$.

Work of Isothermal Compression in ft.-lb. per lb. of substance from P_1 to P_2, $= P_1 v_1 \log_e r = RT \log_e r$ = heat discharged, where $r = P_2/P_1$.

Relation between Pressure, Volume and Temperature for reversible adiabatic changes.

$$P_2 = P_1$$

$$P_1 v_1^{\gamma} = P_2 v_2^{\gamma};\ c_p/c_v = \gamma.$$

For air, $\gamma = 1.40$; $1/\gamma = 0.7143$; $1/(\gamma - 1) = 2.5$; $\gamma/(\gamma - 1) = 3.5$; $(\gamma - 1)/\gamma = 0.2857$

CHARACTERISTIC EQUATION FOR REAL GASES.—Although gases have varying specific heats, in general as can be shown by Table 3, the difference $m(c_p - c_v)$ remains a constant. mR also is substantially constant for real gases and is called the "Universal Gas Constant." Hence one may assume that $Pv = RT$ may be used for 1 lb. of actual gas.

DALTON'S LAW OF GASEOUS PRESSURES.—Every portion of a mass of gas enclosed in a vessel contributes to the pressure against the sides of the vessel the same amount that it would have exerted by itself had no other gas been present.

That is, the total pressure in the vessel is the sum of the partial pressures of each of the constituent gases.

Table 3.—Gas Constants

For a temperature of 32° F. and an absolute pressure of 14.696 lb. per sq. in.

Name of Gas and Formula	Exact Molecular Weight	c_p B.t.u. per lb.	c_v B.t.u. per lb.	$\gamma = \frac{c_p}{c_v}$	$R = 778.6 \times (c_p - c_v)$	v_s = Specific Volume, cu. ft. per lb.	Density, lb. per cu. ft. $= \frac{14.696 \times 144}{491.6} = 4.3048/R$
Air	(28.95)	0.2396	0.1710	1.40	53.39	12.40	0.08063
Carbon monoxide, CO	28.000	0.2477	0.1768	1.401	55.19	12.82	0.07799
Hydrogen, H_2	2.015	3.445	2.460	1.40	767.0	178.18	0.00561
Nitrogen, N_2	28.016	0.2475	0.1767	1.401	55.16	12.81	0.0780
Oxygen, O_2	32.00	0.2167	0.1547	1.401	48.30	11.22	0.08913
Carbon dioxide, CO_2	44.000	0.2065	0.1614	1.28	35.12	8.159	0.1226
Methane, CH_4	16.031	0.5315	0.4077	1.304	96.41	22.40	0.04465
Ethylene, C_2H_4	28.031	0.4707	0.400	1.177	55.17	12.81	0.07807

MIXTURES OF VAPORS AND GASES.—The pressure exerted against the interior of a vessel by a given quantity of a perfect gas enclosed in it is the sum of the pressures which any number of parts into which such quantity might be divided would exert separately, if each were enclosed in a vessel of the same bulk alone, at the same temperature. Although this law is not exactly true for any actual gas, it is very nearly true for many.

EXAMPLE.—0.12344 lb. of CO_2 at 32° F., enclosed in a vessel of 1 cu. ft. capacity, exerts a pressure of 1 atmosphere; consequently, if 0.080728 lb. of air, the weight of 1 cu. ft. at 32° F., and 14.7 lb. per sq. in. absolute, and 0.12344 lb. of CO_2 be mixed, and enclosed at the temperature of 32° F., in a vessel of 1 cu. ft. capacity, the mixture will exert a pressure of 2 atmospheres.

As a second example, let 0.080728 lb. of air, at 212° F., be enclosed in a vessel of 1 cu. ft. capacity; it then will exert a pressure of $(212 + 459.2)/(32 + 459.2) = 1.366$ atmospheres. Let 0.03728 lb. of steam, at 212° F., be enclosed in a vessel of 1 cu. ft. capacity; it will exert a pressure of one atmosphere. Consequently, if 0.080728 lb. of air and 0.03728 lb. of steam be mixed and enclosed together, at 212° F., in a vessel of 1 cu. ft. capacity, the mixture will exert a pressure of 2.366 atmospheres.

If 0.0591 lb. of air (= 1 cu. ft. at 212° and atmospheric pressure) is contained in a vessel of 1 cu. ft. capacity, and water at 212° is introduced, heat at 212° being furnished by a steam jacket, the pressure will rise to two atmospheres.

If air is present in a condenser along with water vapor, the pressure is that due to the temperature of the vapor plus that due to the quantity of air present at its partial pressure.

GENERAL EQUATIONS.—Let w_m be the weight of a mixture of gases of volume v_m at absolute temperature T_m, whose individual weights are $w_1, w_2, w_3 \ldots . w_n$, and whose partial pressure are $P_1, P_2, P_3, \ldots . P_n$ respectively,

then $$w_1 \cdot \quad \cdot w_3 \cdot \quad + w_n = \Sigma w$$

$$v_m = v_1 = v_2 = v_3 = \ldots . v_n. \quad T_m = \text{constant}$$

$$P_m = P_1 + P_2 + P_3 + \ldots . P_n = \Sigma P$$

But $$P_1 = w_1 R_1 T_m/v_m, \text{ and } P_2 = w_2 R_2 T_m/v_m. \therefore P_m = \{\Sigma(wR) \cdot T_m\}/v_m$$

But for a mixture, $P_m = (w_m R_m T_m)/v_m. \quad \therefore R_m = (\Sigma wR)/w_m.$

Also the specific heat at constant pressure of a mixture

$$c_{pm} = (\Sigma wc_p)/w_m, \text{ and } c_{vm} = (\Sigma wc_v)/w_m.$$

If a specific heat equation of the form $mc_p = a + bT + dT^2 + eT^3$ is to be written for a mixture of gases, then $a_m = (\Sigma wa)/w_m$; $b_m = (\Sigma wb)/w_m$; $d_m = (\Sigma wd)/w_m$; $e_m = (\Sigma we)/w_m$, where m = molecular weight, c_p = specific heat per lb. at constant pressure, and a_m, b_m, d_m, and e_m are the coefficients for the mixture. The use of such values may simplify heat computations in a mixture of constant composition, as air or natural gas.

POLYTROPIC CHANGES are those which may be represented on a pressure-volume diagram by a curve following the law Pv^n = constant where n is also a constant. These occur in air compressors, air engines, gas and oil engines. When $P_1 v_1{}^n = P_2$ then

$$\text{or} \quad \left(\frac{P_1}{P_2}\right) = \left(\frac{T_1}{T_2}\right)^{n/(n-1)}$$

The work done, $W \quad \dfrac{P_1 v_1 - P_2 v_2}{n - 1}$ ft.-lb., or $Q = w\left(\dfrac{\gamma - n}{n - 1}\right) c_v (\quad T_2)$

where $\gamma = c_p/c_v$ for the gas or mixture of gases.

To Find n from a Given Curve, which follows the law Pv^n = constant, plot successive values of P and v on logarithmic cross-section paper. All points should be on a straight line if n is a constant. By choosing two points on this line $P_1 v_1$ and $P_2 v_2$, then $n = (\log P_1 - \log P_2)/(\log v_2 - \log v_1)$.

FLOW OF FLUIDS.—According to the equation of continuity, the sum of the enthalpy and kinetic energy of a fluid remains constant at all stages of its flow, provided there is no loss or gain of heat to the outside and no work is done. Hence

$$778.6\,h_1 + V_1{}^2/2g = 778.6\,h_2 + V_2{}^2/2g = 778.6\,h_n + V_n{}^2,$$

where $h_1, h_2, \ldots . h_n$ = enthalpy per lb., B.t.u., and V_1, V_2, V_n = velocities, ft. per sec., at the various state points $1, 2, \ldots . n$.

Also at any point the equation of the continuity of mass will represent conditions thus,

$$w = A_1 V_1/v_{1s} = A_2 V_2/v_{2s} = A_n V_n/v_{ns},$$

where w = weight of steam, lb. per sec., $A_1, A_2, \ldots . A_n$ = area of section, sq. ft., and v_{1s}, $v_{2s} \ldots . v_{ns}$ = specific volume of fluid, cu. ft. per lb.

THE DISCHARGE OF A FLUID THROUGH AN ORIFICE from a vessel at pressure P_1 into a region where pressure P_2 is lower than P_1 but not below the critical pressure is given by the equation $w = A\sqrt{2g\dfrac{P_1}{v_1} \cdot \dfrac{\gamma}{\gamma - 1}\left[\left(\dfrac{P_2}{P_1}\right)^{2/\gamma} - \left(\dfrac{P_2}{P_1}\right)^{(\gamma+1)/\gamma}\right]}$.

where w = weight of fluid, lb. per sec.; A = area of orifice, sq. ft.; P_1 and P_2 = absolute

pressures, lb. per sq. ft.; $g = 32.16$; v_1 = specific volume at pressure P_1, cu. ft. per lb.; γ = exponent in equation for the expansion of the gas, *i.e.*, $P_1 v_1^{\gamma} = P_2 v_2^{\gamma}$.

Critical Pressure.—The flow is a maximum when the absolute pressure P_t at the throat of the orifice is $P_t = P_1 \left(\frac{2}{\gamma + 1}\right)^{\gamma/(\gamma - 1)}$. This pressure is called *Critical Pressure.* The discharge remains constant at the critical pressure value for all pressures P_2 below the critical pressure. For air, $\gamma = 1.4$ and $P_t = 0.528\, P_1$. For superheated steam $\gamma = 1.3$ and $P_t = 0.5457\, P_1$.

Also, since $P_1 v_1^{\gamma} = P_t v_t^{\gamma}$, where v_t = specific volume at the critical pressure P_t, and $P_t = P_1 \left(\frac{2}{\gamma + 1}\right)^{\gamma/(\gamma - 1)}$, $P_1 v_1 = \frac{\gamma + 1}{2} P_t v_t$; the velocity of discharge, V_t at the throat, in ft. per sec., at the critical pressure is

$$V_t :$$

which latter equation is the velocity of sound in the fluid, and often is called the *acoustic velocity.*

FLOW OF GASES.—By the principle of the conservation of energy, it may be shown that the velocity with which a gas under pressure will escape into a vacuum is inversely proportional to the square root of its density; that is, oxygen, which is sixteen times as heavy as hydrogen, would, under exactly the same circumstances, escape through an opening only one-fourth as fast as the latter gas.

3. IDEAL CYCLES

THE CARNOT CYCLE.—Let one pound of gas of an absolute pressure P_1, volume v_1 and absolute temperature T_1 be enclosed in an ideal cylinder, having non-conducting walls but with the bottom a perfect conductor, and having a moving non-conducting frictionless piston. Let the pressure and volume of the gas be represented by the point 1 on the *Pv* or pressure-volume diagram, Fig. 1, and let it pass through four operations, as follows:

1. Apply heat at temperature T_1 to the bottom of the cylinder and let the gas expand, doing work against the piston, at the constant temperature T_1, or isothermally, to $P_2 v_2$ or 2. For isothermal $P_1 v_1 = P_2 v_2$. All pressures are absolute.

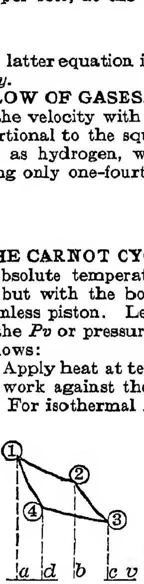


Fig. 1

2. Remove the source of heat, put a non-conducting cover on the bottom, and let the gas expand adiabatically isentropically without eddies or losses or without transmission of heat, to $P_3 v_3$, or 3, while its temperature is being reduced to T_2.

3. Apply to the bottom of the cylinder a cold body, or refrigerator, of temperature T_2, and let the gas be compressed by the piston isothermally to the point 4, or $P_4 v_4$, rejecting heat into the cold body. $P_3 v_3 = P_4 v_4$.

4. Remove the cold body, restore the non-conducting bottom, and compress the gas, adiabatically and isentropically without losses or transmission of heat, to 1, or the original $P_1 v_1$, while its temperature is being raised to the original T_1. The point 4 on the isothermal line 3-4 is so chosen that an isentropic line passing through it also will pass through 1, and so that $v_1/v_4 = v_2/v_3$, and let $v_2/v_1 = r = v_3/v_4$.

The area *a*123*c* represents the work done by the gas on the piston, the area *c*341*a* the negative work, or the work done by the piston on the gas, the difference 1234, is the net work.

1*a*. The area *a*12*b* represents the work done during isothermal expansion. It is equal in ft.-lb. to $W_1 = P_1 v_1 \log_e (v_2/v_1)$, $= P_1 v_1 \log_e r = RT_1 \log_e r$, where P_1 = initial absolute pressure, lb. per sq. ft., and v_1 = the initial volume in cu. ft. It also is equal to the quantity of heat supplied to the gas, $Q_1 = ART_1 \log_e (v_2/v_1)$ in B.t.u. $A = 1/778.6$. R is a constant for a given gas or mixture of gases. See Table 3.

2*a*. The area *b*23*c* is the work done during adiabatic expansion

$\gamma \cdot \quad \{1 - \quad {}^{\gamma - 1}\}$ $\gamma = c_p/c$ = ratio of the specific heat at constant pressure to the specific heat at constant volume. The loss in internal energy during this expansion $= 778.6\, c_v (T_1 - T_2)$ in ft.-lb.

3*a*. *c*34*d* is the work of isothermal compression

$$= W_3 = P_4 v_4 \log_e (v_3/v_4) = P_4 v_4 \log_e r = RT_2 \log_e r.$$

This is equivalent to the heat rejected $= Q_2 = ART_2 \log_e$ in B.t.u.

4*a*. *d*41*a* is the work of adiabatic compression

$$= W = \frac{P_1 v_1}{} \left(\quad \right)^{\gamma - 1}$$

which is the same as W_2, and therefore, being negative, cancels it The net work 1234 $W_1 - W_3$. The gain in internal energy during the compression $d41a = 778.6\, c_v\, (T_1 - T$ in ft.-lb.

$$\text{Efficiency} = \frac{W_1}{Q_1} = \frac{R\,(T_1 - \quad) \log_e r}{} = \frac{T_1 - T_2}{} = 1 - \frac{T_2}{}$$

The Carnot Cycle in the Temperature-entropy Diagram.—Let the pound of gas of temperature T_1 and entropy s be subjected to the four operations described above. 1. T_1 being constant, heat (area *a*12*c*, Fig. 2) is added; entropy increases from 1 to 2, isothermal expansion. 2. No heat is transferred, but the temperature is reduced from T_1 to T_2, entropy constant, isentropic adiabatic expansion from 2 to 3. 3. Heat is rejected at the constant temperature T_2, the area *c*34*a* being subtracted; entropy decreases from 3 to 4, isothermal compression. 4. Entropy constant, temperature increases from 4 to 1, or from T_2 to T_1, no heat transferred, isentropic adiabatic compression. The area *a*12*c* represents the total amount of heat added during the cycle, the area *c*34*a* the amount of heat rejected, the difference, or the area 1234, is the heat converted into work. The ratio of this area to the whole area *a*12*c* is the efficiency. It is the same as the ratio $(T_1 - T_2)/T_1$. It appears from this diagram that the efficiency may be increased by increasing T_1 or by decreasing T_2; also since T_2 cannot be lowered by any self-acting engine below the temperature of the surrounding atmosphere, say $460° + 62°$ F. $= 522°$ F., it is not possible, even in a perfect engine, to obtain an efficiency of 50% unless the temperature of the source of heat is above 1000° F. It is shown also by this diagram that the Carnot cycle gives the highest possible efficiency of a heat engine working between any given temperatures T_1 and T_2, and that the admission and rejection of heat, each at a constant temperature, gives a higher efficiency than the admission or rejection at any variable temperatures within the range $T_1 - T_2$.

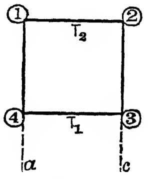


Fig. 2

THE REVERSED CARNOT CYCLE—REFRIGERATION.—Let a pound of cool gas whose temperature and entropy are represented by the *state-point* 4 on the diagram: 1. Receive heat at a constant temperature T_2 (the temperature of a refrigerating room) until its entropy is 3. 2. Let it be compressed isentropically (no losses as 3–2) to a high temperature T_1. 3. Let it reject heat into the atmosphere at this temperature, T_1 (isothermal compression). 4. Let it expand isentropically again with neither friction, eddy, nor other losses, 1–4, doing work by pushing a piston. It then will cool to a temperature which may be far below that of the atmosphere and be used to absorb heat from the refrigerating room. (See Refrigeration, p. 10–11.)

4. VAPORS

A VAPOR is a substance in a condition intermediate between the liquid and gaseous states, and such that it fills the entire container, thereby exerting a pressure as does a gas, but does not follow the laws of ideal gases even approximately.

A liquid when heated at constant pressure reaches a temperature, depending on the pressure, at which it commences to change its physical state to a vapor at that constant temperature. At temperatures below the boiling point the water is *subcooled*. When the liquid reaches the boiling point under these conditions, its energy is called the *saturated heat of the liquid.*

The heat added to complete evaporation at constant pressure and temperature at the boiling point is the *latent heat of vaporization.* When the last particle has been evaporated but the temperature has not risen above the boiling point, the vapor is *dry and saturated.* Any conditions in which droplets of liquid in the form of fog are suspended in the vapor and are not completely evaporated produces a *wet* mixture. The mixture has the same temperature as dry saturated vapor at the same pressure. The ratio of completely evaporated liquid to total substance present in such a wet mixture is known as the *percent quality.* Thus, if 2% of moisture is present as droplets, the quality is 98%. If the dry

and saturated vapor is still further heated, its temperature rises and it becomes *superheated.* In this condition it approximately follows the laws of perfect gases. The *degree of superheat* is the difference between the actual temperature of the superheated vapor and the boiling temperature at the same pressure.

In every liquid there is a certain pressure at which the saturated liquid changes into vapor, with no change in volume and with the addition of no latent heat. This is the *critical state.*

THE ENTHALPY (OR TOTAL HEAT) PER POUND OF LIQUIDS AND VAPORS is expressed by the general equation $h = U + A\,P\,v$, as in perfect gases, where h = enthalpy per lb., B.t.u.; U = internal energy, B.t.u. per lb.; $A = 1/778.6$; P = absolute pressure, lb. per sq. ft.; v = specific volume, cu. ft. per lb.

If v_f = volume of 1 lb. of saturated liquid at a given pressure, and v_g = volume of 1 lb. of dry and saturated vapor at the same pressure, the increase in volume during vaporization is $(v_g - v_f) = v_{fg}$. The volume of the mixture at any quality x is therefore $v_x = v_f + x(v_g - v_f) = v_f + xv_{fg}$.

If h_f = enthalpy or total heat above 32° F. of 1 lb. of saturated liquid at a given pressure in B.t.u., h_{fg} = heat of vaporization of 1 lb. at the same pressure, in B.t.u., and x = quality of the mixture, the *enthalpy or total heat of* 1 *lb. of the mixture* in B.t.u. = $h_x = h_f + xh_{fg}$

SATURATION POINT OF VAPORS.—A vapor that is not near the saturation point behaves like a gas under changes of temperature and pressure. But if it is sufficiently compressed or cooled, it reaches a point where it begins to condense. It then no longer obeys the same laws as a gas, since its pressure cannot be increased by diminishing the size of the vessel containing it, but remains constant, except when the temperature is changed. The only gas that can prevent a liquid evaporating seems to be its own vapor.

The properties of the various vapors such as steam, ammonia, carbon dioxide, etc., and the discussion of humidity are given in other sections of this book.

ABSORPTION OF GASES BY LIQUIDS.—Many gases are absorbed readily by water. Other liquids also possess this power in a greater or less degree. Water will, for example, absorb its own volume of CO_2, 800 times its volume of NH_3, 2 1/3 times its volume of chlorine, and only about 1/20 of its volume of oxygen.

The weight of gas that is absorbed by a given volume of liquid is proportional to the pressure. But as the volume of a mass of gas is less as the pressure is greater, the volume which a given amount of liquid can absorb at a certain temperature will be constant, whatever the pressure. Water, for example, can absorb its own volume of CO_2 at atmospheric pressure; it also will dissolve its own volume if the pressure is twice as great, but in that case the gas will be twice as dense, and consequently twice the weight of gas is dissolved. This principle is the basis of deaeration of condensate in surface condensers.

Table 4 gives the critical temperature and critical pressure of some gases that have been liquefied. The critical temperature is that at which the properties of a liquid and its vapor are indistinguishable, and above which the vapor cannot be liquefied by compression. The critical pressure is the pressure of the vapor at the critical temperature.

Table 4.—Critical Temperatures and Pressures of Gases

(Smithsonian Tables)

Substance	Critical Temperature, deg. F.	Critical Pressure, Atmospheres	Substance	Critical Temperature, deg. F.	Critical Pressure, Atmospheres
Air	−220	39.0	Hydrogen	−401.44	14.
Alcohol (Methyl)	463.91	78.5	" chloride	124.25	86.0
" (Ethyl)	470.48	62.76	" sulphide	212	88.7
Ammonia	266.	115.0	Krypton	−80.5	54.3
Argon	−179.32	52.9	Methane	−115.24	54.9
Carbon dioxide	88.16	73.	Neon	<−337.	29.
" monoxide	−221.98	35.9	Nitric oxide (NO)	−136.3	71.2
" disulphide	523.4	72.9	Nitrogen	−230.8	35.0
Chlorine	285.8	83.9	" monoxide (N_2O)	95.72	75.0
Ethane	91.04	49.0	Oxygen	−180.4	50.0
Ethylene	49.82	51.1	Sulphur dioxide	311.72	78.9
Helium	<−450.4	2.3	Water	706.1	217.5

Section 4

COMBUSTION AND FUELS

COMBUSTION

By Robert Thurston Kent

SOLID FUELS

By Robert Thurston Kent

LIQUID FUELS

By Harry L. Tapp

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ILLUMINATING GAS

By Alfred E. Forstall

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COMBUSTION AND FUELS

COMBUSTION

By Robert Thurston Kent

1. HEAT OF COMBUSTION

THE COMBUSTIBLE CONSTITUENTS of fuel are carbon, hydrogen, sulphur and hydrocarbons. Other constituents are oxygen, nitrogen, moisture and ash. In burning, the combustibles combine with oxygen in the air and in the fuel to form carbon dioxide, carbon monoxide, sulphur dioxide and water. The ash includes all non-combustible matter.

With sufficient air supplied to effect complete combustion carbon burns to CO_2. With insufficient air, it burns to CO. If additional air is supplied, the CO will burn to CO_2.

The hydrocarbons burn to CO_2 and steam. If the temperature is too low for ignition they will pass off unburned. If the air supply is deficient, they will break down to hydrogen and carbon, the latter burning to CO_2 if later brought into contact with sufficient hot air; otherwise it will appear as smoke or soot. The total heat of combustion of any hydrocarbon compound, except CH_4, is approximately the sum of the heats of combustion of the constituents burned separately.

The hydrogen burns to water. Hydrogen and oxygen combined in the fuel as water, or existing in other constituents in the proportions of water are neglected in determining the heating value of the fuel. If the water, actually or virtually present in the fuel, is in sufficient quantity to make its latent heat of evaporation an appreciable quantity, that latent heat is deducted from the heating value of the fuel. Only the excess of hydrogen above that required to form water when combined with the oxygen in the fuel is considered in determining heating value.

Nitrogen in fuel is inert, and is not burned. Nitrogen in the air enters the furnace at atmospheric temperature and leaves it at the temperature of the chimney gases. It is the chief source of heat loss in furnace operation.

Sulphur in fuel is objectionable, as it causes clinker to form. It may exist as iron sulphide or sulphate of lime. The latter form has no heating value.

HEAT OF COMBUSTION. DULONG'S FORMULA.—The heat of combustion of any substance may be determined by burning it in oxygen in a calorimeter (see below), or approximately by Dulong's formula. Let C, H and O equal, respectively, the percentages by weight of carbon, hydrogen and oxygen in 1 lb. of fuel. Then

$$\text{B.t.u. per lb.} = 14{,}600\ C + 62{,}000\ (H - {}^1/_8\,O) + 4000\,S.$$

If the fuel contains CO, the formula is modified to

B.t.u. per lb. $= 14{,}600\,C + 62{,}000\ (H - {}^1/_8\ O) + 4000\,S + 10{,}150\,C_1$, where C_1 = proportionate part of carbon burned to CO. The formula does not hold for methane, CH_4, but appears to hold for ethylene, C_2H_4, and benzole, C_6H_6. Dulong's formula has a probable error not exceeding 2%. Its accuracy has been proved both by Mahler and by Lord and Hass. A deviation in calorimeter tests of more than 2% in the heating value as calculated by the formula (lignites and cannel coals excepted) probably is due to an error in calculation or in the calorimeter test, rather than to the formula.

In calculations of heating value, heat of combustion of hydrogen usually is taken as 62,000 B.t.u. per lb.; of carbon burned to CO_2 as 14,600 B.t.u. per lb. If carbon is burned to CO its heating value is taken as 10,150 B.t.u. per lb. Table 1 gives the heat of combustion of the various constituents of fuels when burned in oxygen.

The total heat of combustion of hydrogen is not available, unless the products of combustion are cooled to the initial temperature of the gases. Since hydrogen burns to steam, a portion of the heat evolved is latent in the steam. The difference between the total heat of combustion, termed the *high heating value*, and the latent heat of the steam is the *low heating value* of hydrogen. The low heating value should not be used in computations concerning fuels containing hydrogen, unless a complete statement is given of all the conditions surrounding combustion. Commercial fuels are burned in air instead of in oxygen, usually with an excess of air. The gases of combustion are discharged at

high temperatures and the steam formed by combustion is superheated to the temperature of the escaping gases. Both cause loss of available heat.

EXAMPLE.—Determine the available heating value of 1 lb. of hydrogen, burned in twice the quantity of air at 62° F. required for combustion, gases of combustion escaping at 562° F.

Solution.—

High heating value of H, determined by calorimeter		62,000 B.t.u.
9 lb. of water heated from 62° to 212° F.	1349.6 B.t.u.	
Latent heat of 9 lb. of water at 212°, 9 × 970.2	8731.8	
Heat required to superheat 9 lb. of steam to 562° F. (specific heat = 0.48), 9 × (562 − 212) × 0.48	1512.0	
Heat required to heat 26.56 lb. of nitrogen to 562° F. (specific heat = 0.2438), 26.56 × (562 − 62) × 0.2438	3237.6	
Heat required to heat 34.56 lb. excess air to 562° F. (specific heat = 0.2375), 34.56 × (562 − 62) × 0.2375	4104.0	
Total heat losses		
Net available heating value		43,065 B.t.u.

CALORIMETERS used to determine heats of combustion are of the bomb type, first devised by M. Pierre Mahler. Some type of adiabatic calorimeter generally is used for precise work. In the bomb calorimeter a weighed sample of the fuel is burned in the bomb, in the presence of oxygen, at a pressure of about 350 lb. per sq. in. The bomb is immersed in a weighed bath of water, and the rise in temperature of the water is noted. From the known weights of water and fuel, the known constants and the temperature rise of the calorimeter, the heat of combustion of the fuel is determined.

The Parr Adiabatic Calorimeter is of the oxygen bomb type, water jacketed. A 1-gram sample is burned in a closed bomb containing oxygen under a pressure of 500 lb. per sq. in. In Fig. 1, the bomb B is submerged in 2000 millileters of distilled water in an oval bucket A, completely surrounded by a water jacket C_1, C_2. Impellers F maintain circulation through the jacket. Sensitive thermometers, D_1, D_2, enter both bucket and jacket. The sample in the bomb is ignited electrically, and as the temperature in the bucket rises, the temperature of the jacket is kept in equilibrium with it by injecting hot water through jet E. This prevents radiation and conduction losses from the bucket A, and the rise in the thermometer D_1 indicates the true heat input due to combustion in the bomb B plus the heat due to fuse wire and formation of acids. Excess water overflows at G. The temperature rise multiplied by the water equivalent of the instrument, gives the uncorrected thermal value of the sample. By deducting the heat due to fuse wire and acids, the true thermal value of the sample is obtained. If Fahrenheit thermometers are used, the result will be in B.t.u. per lb., and if Centigrade thermometers are used the result will be in calories per gram. It is to be noted that in this calorimeter, two major provisions are made for the elimination of errors: 1. The thermally-controlled water jacket bars radiation at its source. 2. The bomb is made of a metal or alloy highly resistive to attack by hot nitric and sulphuric acids; thus in the absence of corrosion, true corrections for the formation of acids are obtained. The results, therefore, are devoid of errors due either to radiation or to corrosion. The water equivalent, or total specific heat, for the instrument is determined by the combustion of a uniform material of known heat value, such as benzoic acid, under conditions identical to those present during the test of an unknown material. The factor thus may be applied directly.

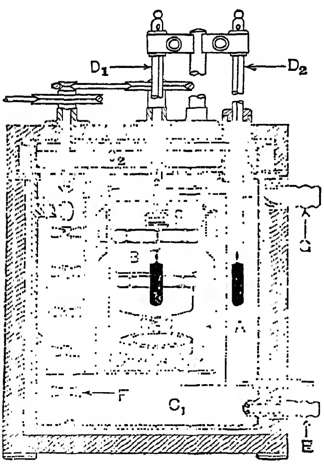


FIG. 1. Parr Adiabatic Calorimeter

HEATING VALUE OF COMPOUND OR MIXED FUELS is the sum of the heating value of their constituents. It is calculated by Dulong's formula, above.

HEAT ABSORBED BY DECOMPOSITION.—The same amount of heat is required to break down a compound as is evolved in its formation. Thus, if 1 lb. of carbon is burned to CO_2 and the CO_2 thus formed is reduced to CO by exposure to glowing carbon, the re-

Table 1.—Heating Value and Products of Combustion of Various Fuels

(Heating values based on tables in U. S. Bureau of Standards Journal of Research, vol ii, 1929)

Fuel	Reaction	Molecular Weight	Lb. O per lb. of Fuel	Lb. N per lb. of Fuel = 3.32 O	Lb. Air per lb. of Fuel = 4.32 O	Gaseous Products of Combustion, lb.	Heating Value, B.t.u. Per lb.	Heating Value, B.t.u. Per cu. ft.*
Carbon to CO_2...	$C+2O=CO_2$		2 2/3	8.85	11.52	12.52	14,542	
Carbon to CO....	$C+O=CO$		1 1/3	4.43	5.76	6.76	4,451	
Carbon monoxide.	$CO+O=CO_2$	28	4/7	1.90	2.47	3.47	10,150	322
Sulphur.........	$S+2O=SO_2$		1	3.32	4.32	5.32	3,940	....
Hydrogen.....	$2H+O=H_2O$	2.014	8	26.56	34.56	35.56	62,032	329
Methane......	$CH_4+4O=CO_2+2H_2O$	16.03	4	13.28	17.28	18.28	23,713	998
Ethane........	$C_2H_6+7O=2CO_2+3H_2O$	30.04	3.74	12.42	16.16	17.16	22,102	1746
Hexane.......	$C_6H_{14}+19O=6CO_2+7H_2O$	86.11	3.07	10.19	13.26	14.26	20,688	4804
Octane........	[illegible]$_8H_{18}$ [illegible] $8CO_2+9H_2O$	114.14	3.51	11.65	15.16	16.16	20,542	6239
Ethylene......	$C_2H_4+6O=2CO_2+2H_2O$	28.03	3.43	11.39	14.82	15.82	21,315	1571
Propylene.....	$C_3H_6+9O=3CO_2+3H_2O$	42.04	3.43	11.39	14.82	15.82	21,007	2323
Propane.......	$C_3H_8+10O=3CO_2+4H_2O$	44.06	3.64	12.08	15.72	16.72	21,449	2487
Butylene......	$C_4H_8+12O=4CO_2+4H_2O$	56.06	3.43	11.39	14.82	15.82	20,801	3064
Butane........	$C_4H_{10}+13O=4CO_2+5H_2O$	58.07	3.59	11.92	15.51	16.51	21,171	3231
Acetylene.....	$C_2H_2+5O=2CO_2+H_2$	26.02	3.07	10.19	13.26	14.26	21,582	1474
Benzene.......	$C_6H_6+15O=6CO_2+3H_2O$	78.05	3.08	10.23	13.31	14.31	18,040	3702
Toluene.......	$C_7H_8+18O=7CO_2+4H_2O$	92.06	3.13	10.39	13.52	14.52	18,264	4427
Naphthalene.....	$C_{10}H_8+14O=5CO_2+4H_2O$	128.06	3.00	9.96	12.96	13.96	17,286	5808
Alcohol (ethyl)...	$C_2H_6O+6O=2CO_2+3H_2O$	46.05	2.09	6.94	9.03	10.03	12,804	1548
Alcohol (methyl).	$CH_4O+3O=CO_2+2H_2O$	32.03	1.50	4.98	6.48	7.48	9,603	818

* Measured as a gas at 62° F. and 14.7 lb. per sq. in. pressure.

sult is the same as if the 2 lb. of carbon had been burned directly to CO. The reactions are

$$2C + 2O_2 = 2CO_2, \text{ generating } 2 \times 14{,}600 = 29{,}200 \text{ B.t.u.}$$

$$C + CO_2 = 2CO, \text{ generating } 2 \times 4450 = 8900 \text{ B.t.u.}$$

The difference, 20,300 B.t.u. is the heat required to liberate 2 lb. of carbon from the CO, or 10,150 B.t.u. per lb. of carbon.

Similarly, if 9 lb. of steam be injected into a bed of glowing coal, it will decompose into 1 lb. H and 8 lb. O, abstracting 62,000 B.t.u. from the bed of coal. If additional air is supplied the hydrogen again will burn to steam, evolving 62,000 B.t.u., and the 8 lb. of oxygen will combine with 6 lb. of carbon to form 14 lb. CO, generating 6 × 4450 = 26,700 B.t.u., or it will combine with 3 lb. of carbon to form 11 lb. CO_2, generating 3 × 14,600 = 43,800 B.t.u.

TEMPERATURE OF THE FIRE.—Let T and t = respectively, temperature of the fire and of the air supplied for combustion, deg. F.; h = total heating value of the fuel; W and S = respectively, weight and specific heats of gases of combustion. Then $T = \{h/(W \times S)\} + t$. Theoretically, rapidity of combustion does not affect temperature. Practically, radiation reduces the temperature of the fuel bed and the gases immediately above it, and considerable heat may be lost if combustion is slow. Usually, at ordinary rates of combustion, in firebrick boiler furnaces, radiation loss is not over 1%, and the temperature of the fire will be almost as high at a combustion rate of 10 lb. per sq. ft. of grate per hr. as at a rate of 20 or 40 lb.

Theoretical temperatures are attained only if the gases combine instantaneously and simultaneously throughout their mass. Evolution of heat is retarded by dissociation and the cooling effect of the sides of the furnace. Also, the specific heat of gases increases at very high temperatures.

To attain high temperature, air must be delivered to the incandescent fuel at a uniform rate, and combustion of carbon and hydrogen must be complete, carbon burning to CO_2 when 11.52 lb. of air or more is supplied per 1 lb. of carbon. If less air is supplied, all the oxygen is used to form CO or CO_2. Also, with insufficient air some oxygen passes through the fire without combining with the carbon. An excess of air also will decrease the temperature. Radiation from the incandescent fuel bed to the surrounding furnace or boiler walls must be prevented to obtain maximum temperatures. The nearest approach to maximum temperature conditions is with gaseous or dust fuels, with an intimate and regular admixture of air in a thick-walled firebrick combustion chamber.

TEMPERATURE OF THE FIRE, FUEL CONTAINING HYDROGEN AND WATER (William Kent, Steam Boiler Economy, 2d. edition, p. 29).—The following assumptions are made: Hydrogen and water exist in the combustion chamber as superheated steam at

the temperature of the fire; specific heat of the gases of combustion is constant at 0.237. The last assumption is probably in error as the specific heat of gases increases with temperature, but the effect is negligible when relative figures only are desired. Let C, H, O and W = respectively, the percentage, by weight, of carbon, hydrogen, oxygen, and water in the fuel; T = temperature of the fire, deg. F.; t = temperature of the atmosphere, deg. F.; f = pounds of dry gas per pound of fuel. Then, approximately,

$$T = \frac{616\,C + 2220\,H - 327\,O - 44\,W}{f + 0.02\,W + 0.18\,H} + t$$

Example.—Required the maximum temperature from combustion of wood of composition C, 38; H, 5; O, 32; ash, 1; moisture, 24. Dry gas per lb. of wood = 15 lb.; atmospheric temperature, 62° F.

$$T = \frac{(616 \times 38) + (2220 \times 5) - (327 \times 32) - (44 \times 24)}{15 + (0.02 \times 24) + (0.18 \times 5)} + 62 = 1503° \text{ F.}$$

Temperature Due to Burning Carbon in Dry Air.—One pound of carbon burned to CO_2 in dry air generates 14,600 B.t.u., evolving 8.85 lb. nitrogen, of specific heat 0.2438, and 3.667 lb. of CO_2, of specific heat 0.217. The mean specific heat is 0.2359. Temperature of fire, with atmospheric temperature at 62° F., then is

$$T = \{14{,}600/(8.85 + 3.67) \times 0.2359\} + 62 = 5005° \text{ F.}$$

This temperature cannot be obtained in practice, due to the excess air necessary for complete combustion. Table 2 gives the temperature of the fire with a deficient air supply, and Table 3 gives temperatures with excess air.

2. AIR REQUIRED FOR COMBUSTION

AIR REQUIRED FOR COMBUSTION.—The minimum amount of air required for combustion is determined from the combustion equation, *i.e.*, the chemical equation expressing the reaction of combustion as given in Table 1. Dividing the total molecular weight of the oxygen required for combustion by the total molecular weight of the combustible will give the weight of oxygen required per pound of combustible. Thus in the equation $2\,C + 2\,O_2 = 2\,CO_2$, the total molecular weights of the combustible and oxygen required for combustion are 24 and 64, respectively. The oxygen required for combustion

Table 2.—Temperature Due to Burning Carbon with Deficient Air Supply

Carbon burned partly to CO_2, partly to CO. Specific heat of gases of combustion assumed = 0.24.

	Deficiency below 11.52 lb. of air per lb. of carbon, percent					
	0	10	20	30	40	50
Lb. of air per lb. of carbon	11.52	10.37	9.22	8.06	6.91	5.76
Total gas per lb. of carbon, lb.	12.52	11.37	10.22	9.06	7.91	6.76
Carbon burned to CO_2, percent	100	80	60	40	20	0
Carbon burned to CO, percent	0	20	40	60	80	100
Heat generated in burning to CO_2, B.t.u.	14,600	11,680	8,760	5,840	2,920	0
Heat generated in burning to CO, B.t.u.	0	890	1,780	2,670	3,560	4,450
Total heat generated, B.t.u.	14,600	12,570	10,540	8,510	6,480	4,450
Loss due to CO, B.t.u.	0	2,030	4,060	6,090	8,120	10,150
Elevation of temp. of fire above atmosphere, deg. F.	4,859	4,606	4,297	3,914	3,413	2,743
Gas analysis by volume, percent:						
CO_2	20.86	18.12	14.87	10.94	6.10	0
CO	0	4.53	9.91	16.41	24.42	34.51
N	79.14	77.35	75.22	72.65	69.48	65.49

Table 3.—Temperature Due to Burning Carbon with Excess Air

All carbon burned to CO_2. Specific heat of gases of combustion assumed = 0.24.

	Excess air above 11.52 lb. per lb. of carbon, percent					
	25	50	75	100	150	200
Lb. of air per lb. of carbon	14.40	17.28	20.16	23.04	28.80	34.56
Total gas per lb. of carbon, lb.	15.40	18.28	21.16	24.04	29.80	35.56
Elevation of temp. of fire above atmosphere, deg. F.	3950	3328	2875	2531	2041	1711
Gas analysis by volume, percent:						
CO_2	16.69	13.91	11.92	10.43	8.34	6.95
CO	4.17	6.95	8.94	10.43	12.52	13.91
	79.14	79.14	79.14	79.14	79.14	79.14

then is 64/24 = 2 2/3 lb. per lb. of carbon. Since air is composed of 23.2% oxygen and 76.8% nitrogen, by weight, dividing the weight of the oxygen required by 0.232 will give the minimum weight of air required.

The minimum air required for a compound fuel is calculated by computing the air necessary per lb. of each constituent of the fuel, and then combining the quantities so found in the proportions in which the constituents exist in the fuel. A greater amount of air than the minimum is required in practice. See Temperature of the Fire, above.

EXAMPLE.—Required the minimum quantity of air to burn a coal whose proximate analysis is fixed carbon, 66%; volatile matter, 22%; ash, 12%. Assume all the volatile to be methane, CH_4. *Solution.*—The combustion equations (Table 1), with the respective molecular weights are:

$$\underset{24}{2\,C} + \underset{64}{2\,O_2} = \underset{88}{2\,CO_2}; \qquad \text{oxygen required} = 64/24 = 2\,{}^2/_3 \text{ lb. per lb. C.}$$

$$\underset{16}{CH_4} + \underset{64}{2\,O_2} = \underset{44}{CO_2} + \underset{36}{2\,H_2O}; \qquad \text{oxygen required} = 64/16 = 4 \text{ lb. per lb. } CH_4$$

Since the fuel contains 66% carbon and 22% CH_4, the oxygen required per lb. of fuel will be $(0.66 \times 2.67) + (0.22 \times 4) = 2.64$ lb. of oxygen per lb. of fuel. Air required is $2.64/0.232 = 11.38$ lb. per lb. of fuel.

AIR REQUIRED TO BURN VARIOUS FUELS.—William Kent (Steam Boiler Economy, 2d edition) gives a formula for the air required to burn fuels of known ultimate analysis, based on 50% excess air above that theoretically necessary for complete combustion. Let C, H and O = respectively, the carbon, hydrogen and oxygen in 1 lb. of fuel = percentage ÷ 100. Then

$$\text{Lb. of air per lb. fuel} = 1.5\{11.52\,C + 34.56(H - {}^1/_8\,O)\}$$

The result, divided by the combustible or the carbon per pound of fuel, gives the air required per pound of combustible or per pound of carbon. Table 4 gives the air required on this basis by typical fuels. Table 5 shows the effect of varying quantities of excess air on CO_2 content in the flue gases, flame temperature and heat loss.

Table 4.—Air Required to Burn Various Fuels

Based on 50 percent of excess air

Kind of fuel	Ultimate analysis of fuel, percent. Coal dried at 221° F.						Pounds of air required for combustion of		
	Carbon	Hydrogen	Oxygen	Nitrogen	Sulphur	Ash	Dry Coal	Combustible	Carbon
Anthracite	76.86	2.83	2.27	0.82	0.78	16.64	14.50	17.39	18.86
Semi-anthracite	78.32	3.63	2.25	1.41	2.03	12.36	15.27	17.42	19.50
Semi-bituminous	86.47	4.54	2.68	1.08	0.57	4.66	17.12	17.96	19.40
Bituminous, Penna.	77.10	4.57	6.67	1.58	0.90	9.18	15.26	16.81	19.65
Bituminous, Ohio	75.82	5.06	10.47	1.50	0.82	6.33	15.04	16.05	19.84
Lignite, Texas	64.84	4.47	16.52	1.30	1.44	11.43	12.45	14.06	19.21
Crude oil, Texas	84.8	11.6	1.1	0.8	1.7	......	......	20.60	24.29
Dried wood	50.0	6.0	41.0	1.0	0	2.0	....	9.09	18.18

Table 5.—CO_2, Heat Loss and Flame Temperature Variations with Excess Air

Based on high-carbon fuel, 14,500 B.t.u. per lb.; flue gas temperature, 575° F.; boiler room temperature, 75° F.

(Uehling Instrument Co., Passaic, N. J.)

CO_2 in Flue Gas Percent, by Volume	Heat Loss in Flue Gas per lb. of Combustible		Lb. Air per lb. of Combustible	Lb. Flue Gas per lb. of Combustible	Flame Temperature, deg. F.	Excess Air, Percent
	Percent	B.t.u.				
19	11.3	1640	13.3	14.3	4710	10
18	11.9	1726	14.0	15.0	4465	17
17	12.6	1828	14.8	15.8	4230	24
16	13.3	1930	15.7	16.7	4000	32
15	14.1	2044	16.8	17.8	3777	40
14	15.0	2175	18.0	19.0	3543	50
13	16.1	2335	19.4	20.4	3300	62
12	17.4	2522	21.0	22.0	3080	74
11	18.9	2740	22.9	23.9	2840	90
10	20.7	3000	25.2	26.2	2600	110
9	22.9	3321	28.0	29.0	2375	130
8	25.8	3740	31.5	32.5	2135	160
7	29.3	4250	36.0	37.0	1837	200
6	34.0	4930	42.0	43.0	1660	250
5	40.7	5900	50.5	51.5	1420	320
4	50.5	7330	63.0	64.0	1165	425

Effect of Preheated Air on Combustion.—P. Nicholls and M. G. Eilers (*Trans.* A.S.M.E. FSP–56–5, May, 1934) describe experiments to determine the effect of varying amounts of air with various coals and high- and low-temperature coke. They show that at each temperature of preheat there is a maximum amount of fuel that can be burned with a given supply of primary air. The maximum varies for each preheat temperature, and an increase in the air supply decreases the amount of fuel that can be burned. They also show that as the size of the coal increased, the rate of combustion with a given amount of air decreased. The authors conclude that preheat gives only limited assistance to combustion, without affecting its value as a means of increasing overall economy. Preheat increases the rate of ignition; based on a normal air temperature of 80° F., preheating to 200° F. increased maximum rate of ignition of Illinois coal 35%, and preheating to 300 increased it 85%. Corresponding figures for Pittsburgh coal were 19% and 43%.

Hosea Webster (*Trans.* A.S.M.E. FSP–51–53, 1929) gives data on the effect of preheated air on boiler efficiency. He cites results of a number of boiler tests from which the following are selected:

	Pulverized Coal	Underfeed Stoker	Chain Grate Stoker	Oil
Efficiency of boiler, percent	82.5	79.9	73.5	69.9
Efficiency of boiler, and air heater, percent	88.2	87.5	82.6	78.4
Gain in efficiency due to air heater, percent	5.7	7.6	9.1	8.5

Highly preheated air may increase stoker maintenance costs. An investigation of a large number of stoker maintenance costs by R. E. Dillon and M. D. Engle (*Trans.* A.S.M.E., FSP–56–17, Dec. 1934) showed that maintenance costs increased sharply with air preheated above 350° F.

FUELS

By Robert Thurston Kent

FUELS IN GENERAL

CLASSIFICATION OF FUELS.—Fuels generally are classified as: 1. Solid, including coal, coke, wood and waste products. 2. Liquid, including petroleum and its derivatives and alcohol. 3. Gaseous, including natural and manufactured gas.

Solid fuels, particularly coal, may be classified in a number of different ways. See paper Constitution and Classification of Coal, H. C. Fieldner, *Trans.* A. S. M. E., FSP–50–51, 1929. They may be classified by *rank*, that is, according to the ratio of fixed carbon to volatile matter in the fuel, peat representing the lowest rank and superanthracite or graphitic coal representing the highest. Table 1 from Fieldner's paper above cited gives the distinguishing chemical and physical characteristics of the various ranks of coal. They also may be classified according to their use, as steam coal, gas coal, etc. Other systems of classification depend on the relations of heating value, proximate analysis and ultimate analysis, as in the Parr system and the Ralston method described below.

Fieldner states that the system of classification based on proximate analysis and calorific values is supplemented by certain physical criteria, especially in differentiating

Table 1.—Chemical and Physical Characteristics of Various Ranks of Coal

Rank, Ash-free	Chemical Characteristics: Approximate Moisture Content*	Chemical Characteristics: Fuel Ratios, FC/VM†	Physical Characteristics
Peat	80–90	....	
Lignite	30–45	....	Brown; clay-like or woody in appearance; slacks on exposure.
Sub-bituminous	12–30	....	Black; disintegrates on exposure, but less rapidly than lignite.
Bituminous	3–15	3 or less	Little or no slacking on exposure to weather.
Semi-bituminous	3–6	3–6	Friable; burns with little smoke.
Semi-anthracite	3–6	4–10	Hard; burns with very little smoke.
Anthracite	2–3	Over 10	Hard; high specific gravity; smokeless.
Superanthracite	2–13	Over 10	Resembles graphite.

* Mine sample. † FC = fixed carbon, percent; VM = volatile matter, percent.

between lignite, sub-bituminous and low-rank bituminous coal. Lignite and sub-bituminous coals disintegrate or slack readily on exposure to weather, the former more rapidly and more completely than the latter. Low rank bituminous coal slacks slowly and incompletely. The slacking is directly proportional to bed moisture in the coal and can be predicted from the moisture content. An accelerated test for slacking characteristics, developed by the U. S. Bureau of Mines, comprises drying for 24 hours of 1 1/2- to 2-in. lumps of coal or lignite of definite weight, at a temperature of 30 to 40 deg. C. The coal then is screened on a 1/4-in. screen and the percentage of undersize determined.

Ralston's Method of Classification is based on taking the sum of the carbon, hydrogen and oxygen of the fuel as 100% and plotting the percentages of these three elements on trilinear coordinates, as in Fig. 1. See O. C. Ralston, Graphic Studies of the Ultimate Analyses of Coal, Tech. Paper No. 93, U. S. Bureau of Mines. A plot of many thousands of coal samples gave a curve that lies in the shaded portion of the lower left hand corner of Fig. 1. For practical use, only the shaded portion of the chart is necessary and it may be reproduced on a much larger scale, as in Fig. 2. Ralston found the fields of the different fuels on the chart to be quite sharply defined as shown in Fig. 2. The analysis of the coal being known, its classification quickly can be determined by locating it on the chart. Thus a fuel, dry and free of ash, whose composition is 74.8 C + 5.2 H + 20 O = 100%, would be located at point 1 in Fig. 2, and would be definitely classified as a lignite.

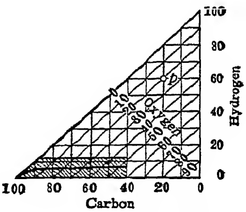
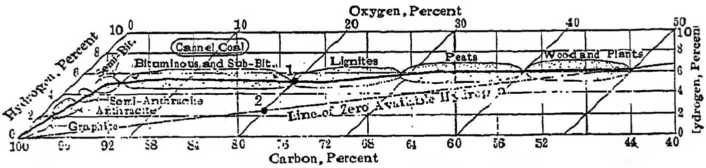


Fig. 1. **Ralston Chart**

It may be assumed that all the oxygen in the fuel, as determined by analysis, is combined with some of the hydrogen, to form H_2O. This appears combined with carbon in some form of carbohydrate, $C_x\ (H_2O)_y$. From this assumption, a line may be drawn on the chart which is the line of zero available hydrogen. The amount of hydrogen available for combustion then is the difference between the total hydrogen as shown by



. 2. Classification of Coal by Ralston Chart

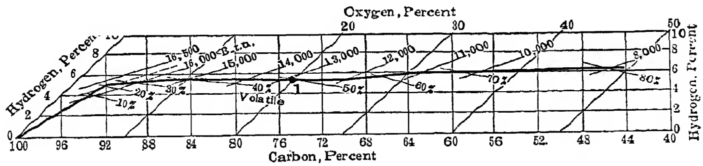


Fig. 3. Ralston Chart with Isocalorific and Isovolatile Lines

analysis and the hydrogen combined with all the oxygen in the form of carbohydrates; for example, the distance on the hydrogen scale between points 1 and 2 in Fig. 2.

If isocalorific and isovolatile lines, *i.e.*, lines showing equal heating values and equal percentages of volatile matter, be superposed on the chart, it may be used to classify a fuel when the percentage of volatile matter and the heating value are known. The percentage of volatile matter is determined by proximate analysis, which should include the sulphur percentage, and in conjunction with the high heating values, reduced to the basis on which the chart is constructed, will locate a point on the isovolatile and isocalorific co-ordinates.

The position of the point determines the classification of the coal and also its approximate composition with respect to carbon, hydrogen and oxygen. See point 1, Fig. 3. Ralston's diagram shows a definite relation to exist between ultimate analysis of a fuel and its calorific value. It permits the application of either ultimate analyses or of volatile matter plus calorific value to the classification of a fuel. It also permits the determination of the volatile matter and calorific value of the fuel from its content of carbon, hydrogen and oxygen.

Parr's System of Fuel Classification is based on the relation of the percentage of volatile matter and the heating value per pound of unit coal. Unit coal is dry coal, ash-free, corrected for contained sulphur, water of hydration and shale. Fig. 4 illustrates the Parr system of classification. See S. W. Parr, The Classification of Coal, *Ind. and Engg. Chem.*, vol. 14, p. 919, 1922.

THE RELATIVE ECONOMY OF VARIOUS FUELS does not depend on the heating value of the fuel itself as much as on the conditions attending its use. Martin Frisch (*Trans.* A.S.M.E., FSP–52–11, 1930) describes boiler tests with the nine fuels shown in Table 2, from which he derives Table 5 showing the relative price that can be paid for different fuels to generate steam at the same total cost with each fuel. The solid fuels were burned in pulverized form. The index of boiler performance used was the sensible heat imparted to the products of combustion per pound passed through the boiler. This index depends on: 1. Heating value of the fuel. 2 The weight of gas and vapor formed by its combustion. 3. The amount of heat made unavailable by incomplete combustion and the evaporation of moisture and formation of water vapor by burning hydrogen. The effect of (2) is small while that of (3) may be large. Excess air influences the magnitude of the index as it affects the total quantity of gas formed per available heat unit. The author presents curves based on this index, which show that the fuels imparting the greatest amount of sensible heat to the flue gases have, in all cases, lower exit temperatures of flue gases and lower draft losses than fuels imparting less sensible heat. See Table 3 compiled from curves in the paper.

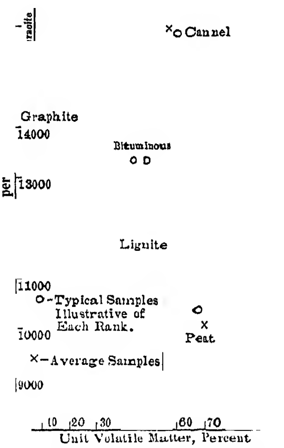


Fig. 4. Parr System of Coal Classification

The capacity developed with a given draft loss increased with the sensible heat per pound of gas passed through the boiler. Flame temperature has a greater effect on capacity than exit temperature. For example, the combustion temperature of natural gas was about 1000° F. higher than that of blast-furnace gas, but the exit temperatures were within 35° F. of each other Preheating the combustion air increased efficiency about 1% at 500% of rating and decreased draft loss over $^3/_4$ in. It also lowered exit temperatures. See also p. 4–08. The difference in efficiency attainable with the different fuels at a given percentage of boiler rating is due also to the completeness of combustion and latent heat losses, which vary with each fuel. Combustion losses with the gaseous fuels and oil are negligible, but the latent heat losses are high, and may exceed 10% with coke-oven and natural gas. With solid fuels burned in pulverized condition the losses due to incomplete combustion and unburned carbon increase with increasing fixed carbon content. Curves in the paper show that the losses due to unburned carbon range about as follows:

B.t.u. liberated per cu. ft. of combustion space	8000	12,000	16,000	20,000	24,000	30,000
Loss due to unburned carbon, percent:						
Anthracite	7.0	7.10	7.25	7.4	7.6	8.1
Pocahontas semi-bituminous	1.30	1.40	1.55	1.70	1.95	2.70
Pittsburgh bituminous	0.60	0.72	0.82	0.90	1.00	2.05
Illinois bituminous	0.35	0.40	0.45	0.50	0.60	0.75
Lignite	0.20	0.20	0.25	0.25	0.25	0.40

The efficiency attainable with Pittsburgh coal is about 0.5% higher than with Pocahontas, and about 0.75% higher than with Illinois coal. Maximum attainable efficiencies with anthracite and Texas lignite are 5 to 6% lower than with Pittsburgh coal, while

Table 2.—Comparative Analyses and Heating Values of Various Fuels

Analysis, percent	Texas Lignite	Illinois Bituminous	Pittsburgh Bituminous	Pocahontas Semi-bituminous	River Anthracite	Oil	Blast-furnace Gas	Natural Gas	Coke-oven Gas
C	39.6	68.0	75.4	80.7	77.6	84.3	...	..	...
H_2	3.0	4.3	4.8	4.7	2.3	12.7	0.2	7.0	9.7
O_2	10.0	8.3	6.1	4.5	3.6	1.0	0.1	0.8	...
N_2	0.9	1.5	1.4	1.3	0.8	0.2	59.6	7.3	30.8
S	0.5	1.4	1.4	1.2	0.7	0.8			
Ash	10.0	10.0	8.5	5.0	14.0	.....	.....	.....	.....
Moisture	36.0	6 5	2.4	2.6	1.0	1.0	.....	.	.....
CO	..	...	.....	.....	.....	......	27.6	25.6	15 2
CH_4	. .	.....	.....	...	.....	.. ..	0.4	30.3	41.3
CO_2	.....	....	.....	...	.. ..	....	12.1	16.0	3.0
C_2H_4	..	.	.	..	..	...	.	5.6	.. ...
C_2H_6	.		.		...	..	. .	7.4	...
Volatile matter	32.0	37.0	34.5	18.2	8.2	....	.....	...	...
Fixed carbon	22.0	46.5	54.6	74.2	76.8	....	..	. .	...
Heating value, B.t.u. per lb.	7000	11,700	13,400	14,500	12,450	18,600	1426	15,530	16,354
Fusion point of ash, deg. F.	2200	2,000	2,100	2,500	2,400	......	.....	...	......
Theoretical air required, lb. per lb. of fuel	5.21	9.04	10.14	10.77	9.63	14.12	0.82	10.3	10 86
FLUE GAS ANALYSIS, PERCENT									
CO_2	15.0	15.0	15.0	15.0	15.0	13.0	21.6	11.3	8.4
O_2	4.3	4.2	3.9	4.1	5.0	3.1	0.9	2.0	1.9
N_2	80.7	80.8	81.1	80.9	80.0	83.9	77.5	86.7	89.7

Table 3.—Sensible Heat, Exit Temperatures and Draft Losses with Various Fuels

Based on 15% CO_2 in flue gas with solid fuels, 13% CO_2 with oil fuel, and 10% excess air with gaseous fuels. Fuels burned in straight-tube boiler.

Fuel	Net Available Heat, B.t.u. per lb. of Flue Gas	Percent of Rated Capacity of Boiler: 100	200	300	400	500	600	100	200	300	400	500	600
		Exit Temperatures, deg. F.						Draft Loss, in. of Water					
Blast-furnace gas	740	503	581	683	792	915	1042	0.36	1.24	2.71	4.89	7.38	9.76
Texas lignite	890	489	544	616	700	789	889	.18	0.80		3.29	5.00	6.80
River anthracite	944	484	533	600	669	752	818	.13	.65		2.87	4.40	6.07
Illinois bituminous	953	484	533	600	669	752	818	.13	.65		2.84	4.36	6.00
Pittsburgh bituminous	986	482	531	587	658	731	818	.11	.58		2.67	4.09	5.62
Pocahontas semi-bituminous	1001	482	526	580	650	722	800	.11	.56		2.58	3.91	5.42
Oil	1001	482	526	580	650	722	800	.11	.56		2.58	3.91	5.42
Coke-oven gas	1117	475	513	562	617	681	745	.03	.44		2.13	3.22	4.36
Natural gas	1130	475	513	560	613	672	733	.00	.38		2.00	2.96	4.00

Table 4.—Efficiency and Heat Input of Various Fuels at Different Rates of Driving

Heat Input = Millions of B.t.u. required in furnace, per 1,000,000 B.t.u. in the steam

Fuel	Percent of Rated Capacity of Boiler: 100 Efficiency, percent	100 Heat Input	200 Efficiency, percent	200 Heat Input	300 Efficiency, percent	300 Heat Input	400 Efficiency, percent	400 Heat Input	500 Efficiency, percent	500 Heat Input	600 Efficiency, percent	600 Heat Input	Maximum Efficiency: Percent of Rated Capacity	Efficiency, percent	Heat Input
Texas lignite	77 48	291	78.00	1 282	76 73	1 303	74.72	1.338	72.39	.381	69.85	432	160	'8.26	1.278
Illinois bituminous	82.41	.213	83.40	1.199	82 51	1 212	80.86	1.237	'8.62	272	76.38	.309	180	83.35	.200
Pittsburgh bituminous	83.1	.202	84 00	1 190	83 00	1.205	81.41	1.228	79.81	.253	77.28	.294	190	84.10	1.189
Pocahontas semi-bituminous	82.70	209	83 71	.195	82.55	1.211	81.00	1.235	79 00	1 266	76.69	1.304	200	83 71	1.195
River anthracite	78.29	1 277	79.14	1 263	77.86	1 284	76.15	1.313	73 85	1.354	71 48	1 399	160		.261
Oil	80.41	.243	81.21	1.231	80 46	1 243	80.48	1 243	77.52	1.290	75.62	1.322	175	81.38	1.229
Blast-furnace gas	80 90	1 236	80.00	1.250	77.30	1.294	73 55	1.360	68.76	1 454	64.19	1.558	115	81.00	1.235
Natural gas	77.00	1 298	78.48		78.00	1.28:	77.07	1.298	75.93	1.317	74 66	1.339	200	78.48	1.274
Coke-oven gas	76 1·	1 313	77 46	1.291	77.05	1.298	76 25	1.311	75 00	1.333	73.76	1.356	230	77.42	1.291

Table 5.—Maximum Relative Price to be Paid for Fuels to Produce Steam at Equal Cost

Prices: Solid fuels, dollars per ton; oil, dollars per barrel; natural gas, cents per 1000 cu. ft.

Fuel for Which Furnace is Designed	Total Cost of Steam, 30 cts. per 1,000,000 B.t.u. — Maximum Price of Fuel Actually Used							Total Cost of Steam, 40 cts. per 1,000,000 B.t.u. — Maximum Price of Fuel Actually Used							Total Cost of Steam, 50 cts. per 1,000,000 B.t.u. — Maximum Price of Fuel Actually Used						
	Lignite	Illinois bituminous	Pittsburgh bituminous	Pocahontas semi-bituminous	River anthracite	Oil	Natural gas	Lignite	Illinois bituminous	Pittsburgh bituminous	Pocahontas semi-bituminous	River anthracite	Oil	Natural gas	Lignite	Illinois bituminous	Pittsburgh bituminous	Pocahontas semi-bituminous	River anthracite	Oil	Natural gas
Lignite only	1.34	2.66	3.15	3.42	....	....	....	2.38	4.52	5.30	5.74	....	....	....	3.41	6.38	7.45	8.06	....	....	....
Lignite and natural gas	1.31	2.59	3.07	3.33	....	....	12.1	2.35	4.45	5.22	5.64	....	....	19.8	3.38	6.31	7.37	7.96	....	....	27.2
Lignite and oil	1.18	2.35	2.79	3.04	....	0.62	....	2.22	4.21	4.94	5.36	....	1.07	....	3.25	6.07	7.09	7.68	....	1.53	....
Lignite, natural gas and oil	1.15	2.30	2.72	2.97	....	0.61	10.7	2.19	4.16	4.87	5.29	....	1.06	18.3	3.22	6.02	7.02	7.61	....	1.52	25.8
Illinois bituminous only	....	3.02	3.55	3.90	....	....	....	....	4.88	5.70	6.22	....	....	....	....	6.74	7.85	8.54	....	....	....
Illinois bituminous and natural gas	....	3.00	3.51	3.85	....	....	13.5	....	4.86	5.66	6.17	....	....	21.0	....	6.72	7.81	8.49	....	....	28.5
Illinois bituminous and oil	....	2.72	3.20	3.51	....	0.70	....	....	4.58	5.35	5.83	....	1.16	....	....	6.44	7.50	8.15	....	1.62	....
Illinois bituminous, gas, and oil	....	2.64	3.16	3.47	....	0.69	12.3	....	4.50	5.31	5.79	....	1.15	19.8	....	6.36	7.46	8.11	....	1.61	27.3
Pittsburgh bituminous only	....	....	3.61	3.96	....	....	....	....	....	5.76	6.28	....	....	....	....	....	7.91	8.60	....	....	....
Pittsburgh bituminous and natural gas	....	....	3.57	3.92	....	....	13.7	....	....	5.72	6.24	....	....	21.2	....	....	7.87	8.56	....	....	28.7
Pittsburgh bituminous and oil	....	....	3.26	3.58	....	0.72	....	....	....	5.41	5.90	....	1.17	....	....	....	7.56	8.22	....	1.63	....
Pittsburgh bituminous, gas and oil	....	....	3.22	3.54	....	0.71	12.5	....	....	5.37	5.86	....	1.16	20.0	....	....	7.52	8.18	....	1.62	27.5
Pocahontas semi-bituminous only	....	....	....	3.99	....	....	....	....	....	....	6.31	....	....	....	....	....	....	8.63	....	....	....
Pocahontas semi-bituminous and natural gas	....	....	....	3.94	....	....	13.8	....	....	....	6.26	....	....	21.3	....	....	....	8.58	....	....	28.9
Pocahontas semi-bituminous and oil	....	....	....	3.60	....	0.72	....	....	....	....	5.92	....	1.18	....	....	....	....	8.24	....	1.63	....
Pocahontas semi-bituminous, gas and oil	....	....	....	3.56	....	0.71	12.6	....	....	....	5.88	....	1.17	20.1	....	....	....	8.20	....	1.62	27.6
River anthracite only	....	....	3.34	3.69	2.40	....	....	....	....	5.49	6.01	4.27	....	....	....	....	7.64	8.33	6.14	....	....
River anthracite and natural gas	....	....	3.31	3.66	2.38	....	12.7	....	....	5.46	5.98	4.25	....	20.3	....	....	7.61	8.30	6.12	....	27.8
River anthracite and oil	....	....	2.99	3.31	2.10	0.72	....	....	....	5.14	5.63	3.97	1.18	....	....	....	7.29	7.95	5.84	1.63	....
River anthracite, gas and oil	....	....	2.96	3.27	2.07	0.71	11.5	....	....	5.11	5.59	3.94	1.17	19.1	....	....	7.26	7.91	5.81	1.62	26.6
Natural gas	....	....	....	....	....	....	15.2	....	....	....	....	....	....	22.7	....	....	....	....	....	....	30.2
Oil	....	....	....	....	....	0.79	....	....	....	....	....	....	1.25	....	....	....	....	....	....	1.72	....
Natural gas and oil	....	....	....	....	....	0.79	13.7	....	....	....	....	....	1.24	21.2	....	....	....	....	....	1.70	28.8

they are but 2 to 2.75% lower with mechanically atomized oil. The attainable efficiencies with blast-furnace gas, and with coke-oven or natural gas are respectively 2 to 20% and 3 to 5% lower than with Pittsburgh coal. See Table 4.

The final cost of steam which determines the most economical fuel depends on: 1. Fixed operating and maintenance costs of boiler plant, exclusive of fuel and fan equipment. 2. Cost of fuel. 3. Fixed operating and maintenance charges on fuel storage, handling, preparation and burning equipment. 4. Fixed and operating charges on fans and draft equipment. 5. Fixed charges on plant capacities reserved to provide for peak requirements of fuel and fan equipment. Item (1) is common to all fuels; items (2) to (5) vary with each fuel.

Table 5 was derived from a study of the cost of operating steam plants designed to burn the fuels shown in the first column. The several columns of the table show the maximum price that can be paid for different fuels other than that for which it was designed if steam is to be generated at the same cost. The figures are absolute only for the conditions stated in the paper, but they will serve as a guide to the relative value of the various fuels when a choice is to be made of fuel in a primary design or it becomes necessary to change to a different fuel. In general, a plant designed for any solid fuel will be most economical when burning that fuel or any higher grade fuel obtainable at the same price per B.t.u. without additional capital expenditure. Some of the higher grade fuels may be more economical, even at a higher cost per B.t.u. A plant designed for a high grade solid fuel will not be economical, with a lower grade fuel. Thus a plant designed for lignite probably would prove more economical with Pocahontas or Pittsburgh coal, but if designed for these could not develop its required capacity with anthracite or lignite without additional capital expenditure. Pulverized solid fuels can compete with oil or natural gas, except in certain localities where oil or gas prices are low or where gas is a waste product.

SOLID FUELS

1. COAL

CLASSIFICATION OF COAL.—Coal is classified according to its percentage of fixed carbon and volatile matter as shown in Table 2. The progressive change from wood to the various grades of coal is shown in Table 1.

William Kent (*Trans.* A.S.M.E., xxxvi, 1914) developed the classification given in Table 2, in which the bituminous coals are divided into three grades according to the percentage of moisture in air-dried coal. The coals of highest inherent moisture also are highest in oxygen.

Table 1.—Progressive Change from Wood to Graphite

(J. S. Newberry in Johnson's Cyclopedia)

	Wood	Loss	Lignite	Loss	Bituminous	Loss	Anthracite	Loss	Graphite
Carbon.....	49.1	18.65	30.45	12.35	18.10	3.57	14.53	1.42	13.11
Hydrogen...	6.3	3.25	3.05	1.85	1.20	0.93	0.27	0.14	0.13
Oxygen....	44.6	24.40	20.20	18.13	2.07	1.32	0.75	0.75	0.00
	100.0	46.30	53.70	32.33	21.37	5.82	15.55	2.31	13.24

Table 2.—Classification of Coal

Class	Type of Coal	Volatile, percent of Combustible	Oxygen in Combustible, percent	Moisture in Air-dry, Ash-free Coal, percent	B.t.u. per pound of Combustible	B.t.u. per pound of Air-dry, Ash-free Coal
I	Anthracite..........	Under 10	1 to 4	Under 1.8	14,800 to 15,400	14,600 to 15,400
II	Semi-anthracite.......	10 to 15	1 to 5	" 1.8	15,400 to 15,500	15,200 to 15,500
III	Semi-bituminous......	15 to 30	1 to 6	" 1.8	15,400 to 16,050	15,300 to 16,000
IV *	Cannel..............	45 to 60	5 to 8	" 1.8	15,700 to 16,200*	15,500 to 16,050*
V	Bituminous, high grade	30 to 45	5 to 14	1 to 4	14,800 to 15,600	14,350 to 14,400
VI	" medium grade	32 to 50	6 to 14	2.5 to 6.5	13,800 to 15,100	11,300 to 14,400
VII	" low grade	32 to 50	7 to 14	5 to 12	12,400 to 14,600	11,300 to 13,400
VIII	Sub-bituminous; lignite	27 to 60	10 to 33	7 to 26	9,600 to 13,250	7,400 to 11,650

* Eastern cannel. The Utah cannel coal has a much lower heating value.

The U. S. Geological Survey (Bulletin 531) classifies coal as: 1. Anthracite. 2. Semi-anthracite. 3. Semi-bituminous. 4. Bituminous. 5. Sub-bituminous or black lignite. 6. Lignite. The differentiation of the several classes is by the fuel ratio, *i.e.* (fixed carbon ÷ volatile content). Fuel ratios are as follows: Anthracite, not less than 10; semi-anthracite, 6 to 10; semi-bituminous, 2.5 to 6; bituminous, 1 to 2.5. The sub-bituminous coals generally are black and shining, resembling bituminous coal, but having a lower calorific value. They weather more rapidly and have not the prismatic structure of bituminous coal. Lignite is brown in color, and has a woody structure. Its moisture runs from 30 to 40% higher than that of other classes of coal. Table 3 (Prof. N. W. Lord, *Power*, Aug. 18, 1908) gives analyses of representative coals of the six classes.

Table 3.—Analyses of Representative Coals

(U. S. Geological Survey Classification)

Class No.	1	2	3	4a	4b	5	6
Kind	Anthracite Culm	Semi-anthracite	Semi-bituminous	Bituminous, Coking	Bituminous, Non-coking	Sub-bituminous	Lignite
Location	Penna.	Ark.	W. Va.	Pa.	Ohio	Wyo.	Tex.
Moisture	2.08	1.28	0.65	0.97	7.55	8.68	9.88
Volatile combustible	7.27	12.82	18.80	29.09	34.03	41.31	36.17
Fixed carbon	74.32	73.69	75.92	60.85	52.57	46.49	43.65
Ash	16.33	12.21	4.63	9.09	5.85	3.52	10.30
Loss on air-drying	3.40	1.10	1.10	4.20	Undet.	11.30	23.50
ULTIMATE ANALYSIS OF COAL DRIED AT 221°F.							
	2.63	3.63	4.54	4.57	5.06	5.31	4.47
	76.86	78.32	86.47	77.10	75.82	73.31	64.84
	2.27	2.25	2.68	6.67	10.47	15.72	16.52
	0.82	1.41	1.08	1.58	1.50	1.21	1.30
	0.78	2.03	0.57	0.90	0.82	0.60	1.44
	16.64	12.36	4.66	9.18	6.33	3.85	11.43
RESULTS CALCULATED TO AN ASH- AND MOISTURE-FREE BASIS							
Volatile combustible	8.91	14.82	19.85	32.34	39.30	47.05	45.31
Fixed carbon	91.09	85.18	80.15	67.66	60.70	52.95	54.69
	Ultimate Analysis						
Hydrogen	3.16	4.14	4.76	5.03	5.41	5.50	5.05
Carbon	92.20	89.36	90.70	84.89	80.93	76.35	73.21
Oxygen	2.72	2.57	2.81	7.34	11.18	16.28	18.65
Nitrogen	0.98	1.61	1.13	1.74	1.61	1.25	1.47
Sulphur	0.94	2.32	0.60	1.00	0.87	0.62	1.62
CALORIFIC VALUE IN B.T.U. PER LB., BY DULONG'S FORMULA							
Air-dried coal	12,472	13,406	15,190	13,951	12,510	11,620	10,288
Combustible	15,286	15,496	16,037	15,511	14,446	13,235	12,889

CAKING AND NON-CAKING COALS are classifications sometimes applied to bituminous coals. Caking coals soften under heat, and the pieces coalesce. These coals yield a compact, dense coke. Non-caking coals (also called free-burning) do not fuse, and the pieces preserve their form under heat. They produce a coke that is useful only when made from large pieces.

CANNEL COALS are bituminous coals with a higher hydrogen content than ordinary coals. They are used as enrichers in gas making. See Table 5.

GRAPHITIC ANTHRACITE is found in Rhode Island and eastern Massachusetts. It resembles anthracite and is burned with great difficulty. See Table 5.

Analyses and Heating Value of Coals

PROXIMATE ANALYSES determine percentages of moisture, volatile hydrocarbons, fixed carbon and ash in coal. See A.S.M.E. Test Code for Solid Fuels and A.S.T.M. Specification D271–33 for detailed instructions for making such analyses. The following paragraphs summarize these methods.

Moisture is determined by subjecting a sample of approximately 1 gram to a temperature of 220 to 230° F. Place sample in a porcelain or fused silica capsule, 1 3/4 in. diam., 7/8 in. deep, fitted with an aluminum cover, capsule and cover having been previously heated over concentrated H_2SO_4 for 30 min., cooled and weighed. Weigh sample, capsule and cover, place in a preheated (220 to 230° F.) oven, with cover removed and oven closed. Close oven and heat for 1 hr.; then cover capsules, cool in a desiccator and weigh. Loss of weight divided by weight of original sample is percentage of moisture in the coal. Air in the oven should be renewed 2 to 4 times per min., the air being dried by passing through concentrated H_2SO_4.

Samples are air-dried by spreading on tared pans and exposing to air at room temperature or at 18 to 27° F. above room temperature until loss of weight does not exceed 0.1% per hr. Ten-pound samples of dry or air-dried coal are crushed to pass a No. 20 sieve, sampled, reduced on a riffle to pass a No. 60 sieve. Moisture is determined on both No. 20 and No. 60 samples. Moisture in air-dried samples passing No. 20 sieve is corrected to total moisture as received. The following formulas will enable the analysis to be corrected to as received and dry-coal bases:

Air-dried to as-received

$$M_r = \{(100 - L_a)/100\}M_a + L_a$$
$$V_r = V_a\{(100 - L_a)/100\}$$
$$FC_r = FC_a\{(100 - L_a)/100\}$$
$$A_r = A_a\{(100 - L_a)/100\}$$

Air-dried to dry coal

$$V_d = V_a\{100/(100 - M_a)\}$$
$$FC_d = FC_a\{100/(100 - M_a)\}$$
$$A_d = A_a\{100/(100 - M_a)\}$$

M_r = total moisture as received; M_a = moisture in air-dry sample passing No. 20 sieve; L_a = air drying loss; V = volatile matter; FC = fixed carbon; A = ash; subscripts a, d, r = respectively, air-dried, dry coal, and as received; all quantities are in percent.

Volatile Matter is determined by heating a 1-gram sample in a platinum crucible, whose maximum and minimum dimensions are: diam., 35 and 25 mm.; height, 35 and 30 mm.; capacity, 20 and 10 cu. cm. The furnace should be a vertical electric tube furnace or a gas-fired muffle furnace, capable of regulation to maintain the temperature in the crucible at 1740 ± 36° F. Temperature determination is by a thermocouple in the furnace. Place the weighed covered crucible and sample in furnace, previously heated to 1740° F., and, after the more rapid discharge of volatile has ceased as shown by subsiding of luminous flame, expose to this temperature for exactly 7 min. Remove, cool and weigh. The loss of weight minus the moisture, divided by the weight of the sample is the percentage of volatile.

Ash is determined with the capsule and sample from the moisture determination. Place capsule and dried sample in a cold gas or electric muffle furnace, capable of temperature regulation between 1290 and 1380° F., and having good air circulation. Gradually heat to a redness at a rate that avoids loss from too rapid expulsion of volatile matter, and continue at a temperature between 1290 and 1380° F. until the weight is constant (± 0.001 gram). Stir ash with a platinum or nichrome wire before completion of test. Cool in a desiccator and weigh. Deduct weight of capsule, and divide by weight of original sample to obtain percentage of ash.

Fixed Carbon is determined from the previous determinations. Percentage of fixed carbon = (100 − % moisture − % volatile − % ash).

Sulphur is determined and reported separately. See reference above cited for methods.

Relative Value of Constituents.—The heating value of the fixed carbon in coal is about 14,600 B.t.u. per lb. That of the volatile matter depends on its chemical composition, varying with the district in which the coal is mined. In semi-bituminous coals, containing little oxygen, it may be as high as 21,000 B.t.u. per lb. In coals higher in oxygen, it may be as low as 12,000 B.t.u. per lb. The ash has no heating value. The moisture detracts from the heating value, as some of the heat of combustion of the fixed carbon and volatile matter is used to evaporate it and superheat the resultant steam to the temperature of the chimney gases.

THE HEATING VALUE OF COAL depends on its percentage of total combustible and the heating value per pound of the combustible. The latter is related to the percentage of volatile matter. It is highest in the semi-bituminous coals, being nearly constant at about 15,570 B.t.u. per lb. In anthracites, it ranges from 14,800 to 15,500 B.t.u., and from 13,000 to 15,500 B.t.u. in bituminous coals. It decreases with an increase of oxygen in the combustible. In lignites it may be as low as 10,000 B.t.u. per lb. The B.t.u. per lb. of combustible should be stated to permit comparison of different coals. Table 4, showing approximate heating value of coals in relation to volatile matter, was deduced by William Kent from Mahler's tests on European coals.

Table 4.—Approximate Heating Values of Coals

Percent Volatile Matter in Coal Dry and Free from Ash	Heating Value, B.t.u. per lb. of Combustible	Equivalent Water Evaporated, lb., from and at 212° F. per lb. of Combustible	Percent Volatile Matter in Coal Dry and Free from Ash	Heating Value, B.t.u. per lb. of Combustible	Equivalent Water Evaporated, lb., from and at 212 °F. per lb. of Combustible
0	14,580	15.09	32	15,480	16.03
3	14,940	15.47	37	15,120	15.65
6	15,210	15.75	40	14,760	15.28
10	15,480	16.03	43	14,220	14.72
13	15,660	16.21	45	13,860	14.35
20	15,840	16.40	47	13,320	13.79
28	15,660	16.21	49	12,420	12.86

The experiments of Lord and Haas on American coals (*Trans.* A.I.M.E., 1897) practically confirm these figures for all coals in which the percentage of fixed carbon is 60% and over of the combustible, but for coals containing less than 60% fixed carbon or more

(*Continued on p. 4-20*)

Table 5.—Analyses and Heating Values of Coals of the United States

State	County	Proximate				Air Dry Loss	Ultimate					B.t.u. per lb.		
		Moisture	Volatile Matter	Fixed Carbon	Ash		S	H	C	N	O	As Received	Air-dry Ash-free	Ash and Moisture free
I—Anthracite														
Alaska	Bering River	7.43	6.86	71.35	14.36	6.2	0.57	3.71	70.10	1.50	9.76	11,891	14,968	15,203
Colo.	Gunnison	2.70	3.32	88.15	5.83	....	0.80	3.28	85.38	1.12	3.59	14,099	....	15,413
Penn.	Schuylkill	2.80	1.16	88.21	7.83	1.5	0.89	1.89	84.36	0.63	4.40	13,298	14,666	14,882
Penn.	"	3.33	3.27	84.28	9.12	2.6	0.60	3.08	81.35	0.79	5.06	13,351	15,123	15,248
Wash.	Lewis	2.7	7.0	79.6	10.67	2.0	0.62	3.48	79.22	1.32	4.69	13,350	15,287	15,410
II—Semi-anthracite														
Ark.	Johnson	2.36	12.68	72.88	12.08	1.1	1.99	3.82	76.44	1.37	4.30	13,259	15,256	15,496
Penn.	Sullivan	3.38	8.47	76.65	11.50	2.6	0.63	3.58	78.43	1.0	4.86	13,156	15,315	15,457
Va.	Montgomery	4.80	10.12	67.05	18.03	4.1	0.63	3.91	69.27	0.66	7.50	11,961	15,360	15,500
III—Semi-bituminous														
Ala.	Jefferson	3.08	26.86	66.31	3.75	2.0	0.55	5.30	82.04	1.48	6.88	14,681	15,577	15,757
"	"	2.38	25.90	66.84	4.88	1.5	1.46	5.16	81.68	1.53	5.29	14,487	15,474	15,620
Alaska	Bering River	5.14	13.90	75.96	5.00	4.6	1.16	4.50	80.68	1.38	7.28	14,065	15,559	15,651
Ark.	Logan	2.77	14.69	73.47	9.07	2.2	2.79	4.02	78.71	1.46	3.95	13,774	15,523	15,624
"	Sebastian	3.21	14.84	72.66	9.29	2.4	3.12	3.75	78.37	1.52	3.95	13,588	15,387	15,530
Colo.	Pitkin	0.96	21.49	68.93	8.62	0.2	0.52	4.66	79.61	1.83	4.76	14,330	15,716	15,849
"	"	3.07	22.67	65.10	9.16	1.8	0.63	4.96	78.81	1.69	4.75	13,990	15,712	15,939
Ga.	....	3.80	15.88	65.83	14.49	3.2	1.27	4.32	70.59	1.09	8.24	12,791	15,540	15,653
Md.	Allegany	3.2	14.5	75.6	6.7	2.2	0.92	4.51	80.99	1.77	5.11	14,100	15,477	15,640
"	"	2.6	15.5	75.1	6.8	2.0	0.89	4.60	81.34	1.81	4.58	14,360	15,746	15,850
Mont.	Gallatin	2.05	16.42	73.22	8.31	1.5	0.86	4.08	81.03	1.28	4.44	14,092	15,625	15,721
Okla.	Haskell	2.37	19.26	69.54	8.83	1.9	1.03	4.57	79.39	1.55	4.63	13,840	15,504	15,586
"	LaFlore	5.11	13.65	73.21	8.03	4.5	1.18	4.65	78.37	1.60	6.17	13,662	15,619	15,728
Penn.	Cambria	4.25	21.79	66.09	7.87	3.9	1.59	4.72	75.92	1.28	8.62	13,513	15,316	15,376
"	Somerset	1.10	15.80	75.69	7.41	....	1.49	4.20	81.98	1.36	3.56	14,499	....	15,847
Va.	Tazewell	4.0	17.5	74.2	4.31	3.5	0.62	4.76	82.56	1.03	6.72	14,520	15,750	15,840
"	"	4.10	16.5	75.2	3.18	3.5	0.63	4.97	84.29	1.21	5.72	14,740	15,795	15,910
Wash.	Pierce	5.81	12.54	64.61	17.04	4.5	0.38	4.10	68.59	1.67	8.22	11,776	15,008	15,264
W. Va.	Fayette	3.71	23.53	69.37	3.39	2.6	0.80	5.14	82.06	1.47	7.14	14,306	15,218	15,399
"	Mercer	1.75	18.59	75.08	4.58	1.1	0.56	4.65	84.97	1.06	4.18	15,023	15,928	16,038

Table 5.—Analyses and Heating Values of Coals of the United States—(*Continued*)

State	County	Proximate: Moisture	Proximate: Volatile Matter	Proximate: Fixed Carbon	Proximate: Ash	Air Dry Loss	Ultimate: S	Ultimate: H	Ultimate: C	Ultimate: N	Ultimate: O	B t.u per lb.: As Received	B t.u per lb.: Air-dry Ash-free	B t.u per lb.: Ash and Moisture free
						IV—Cannel Coal								
Ky.	Johnson	2.36	48.40	38.75	10.49	1.5	1.20	6.47	71.98	1.16	8.70	13,770	15,646	15,800
"	"	1.70	50.76	38.23	9.31	0.4	1.02	6.83	73.25	1.31	8.28	14,251	15,784	16,013
W. Va.	Fayette	1.80	44.9	49.86	3.44	1.0	0.87	6.96	80.57	1.51	6.65	15,330	16,042	16,176
Utah	Kane	7.35	46.93	22.48	23.24	1.1	1.61	6.18	51.88	1.06	16.03	10,355	13,686	14,918
						V—Bituminous, high grade								
Ala.	Jefferson	2.18	31.71	63.32	2.79	1.0	1.07	4.98	80.86	1.71	8.59	14,816	15,400	15,590
"	Shelby	3.83	32.03	58.66	5.48	2.2	0.97	5.29	77.26	1.25	9.75	13,799	14,947	15,214
Colo.	Delta	2.64	28.84	63.31	5.21	1.7	0.66	5.05	76.81	1.28	10.99	13,529	14,533	14,681
"	Las Animas	2.28	29.81	58.75	9.16	1.5	0.50	5.22	74.24	1.08	9.80	13,781	15,425	15,559
Ill.	Saline	7.81	33.54	50.27	8.38	5.8	2.36	5.31	67.40	1.44	15.11	12,418	14,470	14,818
Kans.	Cherokee	2.50	33.80	51.25	12.45	......	5.68	4.91	69.07	1.20	6.69	12,900	......	15,167
"	Linn	9.04	29.69	45.55	15.72	7.1	3.72	5.01	60.99	1.06	13.50	11,142	14,436	14,809
Ky.	Pike	3.41	32.08	58.78	5.73	2.0	0.53	5.22	77.01	1.18	10.33	13,928	15,095	15,328
New Mex.	Colfax	2.78	34.31	48.34	14.57	1.4	0.61	5.06	68.51	1.51	9.74	12,294	14,630	14,875
"	"	2.45	27.54	52.61	17.40	1.8	0.77	4.82	68.38	0.90	7.73	12,200	15,099	15,221
Ohio	Jefferson	3.53	37.45	49.90	9.12	1.4	3.47	5.15	71.66	1.31	9.29	13,072	14,609	14,965
"	Vinton	5.59	36.86	49.26	8.29	3.2	3.15	4.88	69.76	1.18	12.74	12,773	14,431	14,832
Okla.	Pittsburgh	2.09	27.59	50.25	20.07	1.0	5.73	4.46	63.66	1.33	4.75	11,695	14,817	15,025
"	"	2.81	36.11	52.33	8.75	1.4	1.82	5.12	73.46	1.85	9.00	13,320	14,825	15,061
Penn.	Allegheny	2.61	34.92	56.30	6.17	1.3	1.26	5.21	77.14	1.57	8.65	13,997	15,127	15,345
"	Fayette	5.13	27.87	58.29	8.71	4.2	0.86	4.91	73.13	1.50	10.89	13,365	15,346	15,511
Tenn.	Anderson	6.39	32.32	51.76	9.53	4.7	0.98	5.41	70.16	1.56	12.36	12,578	14,665	14,960
"	Cumberland	3.89	27.61	54.07	14.43	2.9	0.78	4.84	70.04	0.98	8.93	12,514	15,137	15,320
Va.	Lee	4.44	35.99	53.59	5.98	2.1	0.76	5.19	71.95	1.25	14.87	13,363	14,537	14,918
"	Wise	3.31	32.82	60.11	3.76	1.9	0.90	5.55	80.01	1.58	8.20	14,209	15,061	15,291
Wash.	Kittitas	2.32	37.02	47.08	13.58	......	0.60	4.67	66.77	0.61	13.77	12,433	......	14,796
W. Va.	Kanawha	4.21	35.41	53.16	7.22	2.3	0.64	5.16	72.89	1.40	12.69	13,379	14,787	15,107
"	Mingo	2.86	33.23	58.08	5.83	1.6	0.67	5.01	78.38	1.43	8.68	14,105	15,237	15,448
Wyo.	Uinta	5.49	35.88	55.51	3.12	3.5	0.91	5.26	74.15	1.26	15.30	13,570	14,532	14,848
						VI—Bituminous, medium grade								
Ala.	Walker	3.95	30.70	50.76	14.59	1.7	1.12	4.57	66.21	1.49	12.02	11,785	14,078	14,467
Alaska	Peninsula	7.06	31.48	39.68	21.78	5.2	1.30	4.83	55.14	0.61	16.34	9,846	13,484	13,838
Cal.	Monterey	6.95	46.69	40.13	6.23	2.2	4.17	6.28	66.01	1.17	16.14	12,447	13,593	14,336
Ill.	Franklin	8.12	34.46	48.79	8.63	3.6	1.13	5.26	66.44	1.33	17.21	12,064	13,745	14,492
"	Williamson	8.86	31.25	48.23	11.66	5.8	2.46	5.24	64.29	1.29	15.06	11,702	14,177	14,724

(*Table continued on following page*)

Table 5.—Analyses and Heating Values of Coals of the United States—(*Continued*)

State	County	Proximate: Moisture	Proximate: Volatile Matter	Proximate: Fixed Carbon	Proximate: Ash	Air Dry Loss	Ultimate: S	Ultimate: H	Ultimate: C	Ultimate: N	Ultimate: O	B.t.u. per lb.: As Received	B.t.u. per lb.: Air-dry Ash-free	B.t.u. per lb.: Ash and Moisture free
						VI—Bituminous, medium grade—*Continued*								
Ind.	Clay	16.91	26.85	38.87	17.37	13.1	1.89	5.48	52 97	1 01	21.28	9,524	13,698	14,492
"	Vigo	7.88	36.85	41.07	14.20	3.7	5.14	5.22	59 75	0.93	14.76	11,146	13,576	14,305
Iowa	Polk	13.88	36.94	35.17	14.01	9.8	6.15	5.52	54.68	0.84	18.80	10,244	13,445	14,206
"	Wapello	8.24	30.74	45.02	16.00	3.2	5.03	4.81	59.82	0.94	13.40	11,027	13,647	14,555
Kan.	Atchison	6.95	35.70	45.16	12.19	3.5	8.04	5.25	62.74	1.04	10.74	11,905	14,121	14,724
"	Cherokee	2.50	33.80	51.25	12.45	....	5.68	4.91	69.07	1.20	6.69	12,900	....	15,167
Ky.	Hopkins	7.92	36.09	45.93	10.06	2.2	3.52	5.39	65.29	1.40	14.34	12,022	13,702	14,657
"	Webster	5.27	35.07	45.48	14.18	2.8	4.54	4.71	64.65	1 24	10.68	11,950	14,394	14,836
Mich.	Saginaw	11.91	31.50	49.75	6.84	7.9	1.24	5.84	66.56	1.19	18.33	11,781	13,818	14,499
"	"	11.55	31.65	53.55	3.25	6.5	0.95	5.88	69.46	1.25	19.21	12,442	13,786	14,603
Mo.	Adair	17.30	26.43	32.89	23.38	15.2	2.94	5.04	45.41	0 83	22.40	8,240	13,416	13,892
"	Miller	12.67	41.45	41.05	4.83	7.7	5.12	6.18	66.87	0.69	16.31	12,487	14,276	15,134
Mont.	Cascade	3.51	26.39	50.60	19.50	1.6	3.74	4.13	61.51	0.68	10.44	10,881	13,791	14,648
"	Gallatin	5.77	33.14	50.52	10.57	3.8	0.50	5.38	69.23	1.01	13.31	12,281	14,342	14,681
New Mex.	Colfax	5.02	36.78	46.20	12.00	2.4	0.56	5.15	66.01	1 28	15.00	12,064	14,093	14,539
Ohio	Belmont	4.14	39.30	47.18	9.38	2.6	3.96	5.19	69.58	1.20	10.69	12,874	14,626	14,888
"	Jackson	7.71	38.32	42.02	11.95	4.9	4.61	5.41	62 49	1.11	14.43	11,515	13,848	14,332
Okla.	Okmulgee	7.04	34.55	48.40	10.01	3.3	1.92	5 34	67.55	1.25	13.93	12,202	14,075	14,711
Utah	Carbon	5.58	38.92	46.51	8.99	1.1	0.51	5 27	67.84	1.07	16.32	12,170	13,536	14,245
"	"	6.05	42.02	47.06	4.87	3.8	0.55	5.76	72.32	1.38	15.12	13,151	14,399	14,764
Wash.	King	6.02	31.16	43.47	19.35	3.5	0.42	4.90	59.59	1.15	14.59	10,708	13,879	14,348
Wyo.	Uinta	3.96	36.16	55.11	4.77	1.4	0.77	5.17	76.03	1.31	11.95	13,502	14,390	14,792
						VII—Bituminous, low grade								
Alaska	Thompson Valley	10.77	30.37	43.99	14.87	6.5	0.70	4.98	55.27	0.61	23.57	9,641	12,261	12,964
Ill.	Clinton	11.35	34.62	40.63	13.40	5.9	4.76	5.41	57.36	1.05	18.02	10,733	13,300	14,263
"	Montgomery	14.43	29.48	42.81	13.28	9.8	4.01	5.49	54.59	1.11	21.52	10,064	13,084	13,921
Ind.	Pike	12.11	34.19	46.87	6.83	3.5	1.44	5.70	65.57	1.14	19.32	11,952	13,329	14,746
"	Warrick	13.13	31.92	39.27	15.63	8.5	4.79	5.36	54.52	1.08	18.62	10,030	13,220	14,089
Iowa	Appanoose	14.08	35.59	39.37	10.96	4.5	4.26	5.57	58.49	0.90	19.82	10,723	12,684	14,305
Mo.	Adair	15.36	34.81	38.84	10.99	9.5	3.57	5.88	57.09	0.95	21.52	10,460	13,852	14,202
Mont.	Carbon	9.76	27.66	46.16	16.42	3.7	0.63	5.09	56.19	1.01	20.66	10,235	12,813	13,865
"	"	10.88	20.27	41.97	26.88	5.4	1.79	3.72	47.37	0.52	19.72	7,742	11,432	12,438
New Mex.	McKinley	12.29	34.58	46.14	6.99	1.6	0.63	5.82	63.31	1.03	22.22	11,252	13,099	13,939
"	San Juan	15.79	34.99	39.85	9.37	7.6	1.78	5.93	55.74	1.39	25.79	9,970	12,154	13,322

Table 5.—Analyses and Heating Values of Coals of the United States—(*Continued*)

State	County	Proximate				Air Dry Loss	Ultimate					B.t.u. per lb.		
		Moisture	Volatile Matter	Fixed Carbon	Ash		S	H	C	N	O	As Received	Air-dry Ash-free	Ash and Moisture free
VII—Bituminous, low grade—*Continued*														
Okla.	Coal	8.29	30.61	36.05	25.05	2.7	3.95	4.37	50.98	1.19	14.46	9,110	12,604	13,667
Ore.	Coos	20.84	34.04	36.75	8.37	11.3	1.17	......	......	......	......	10,348	12,882	14,618
Utah	Iron	10.35	36.33	43.70	9.62	1.8	5.82	5.13	61.24	0.95	17.24	10,874	12,276	13,586
"	"	14.19	33.39	42.50	9.92	2.8	5.39	5.20	55.27	0.85	23.37	9,927	11,374	13,081
Wash.	King	12.05	36.82	40.72	10.41	6.6	0.34	5.75	58.15	1.37	23.98	10,414	12,548	13,423
VIII—Sub-Bituminous and Lignite														
Ark.	Ouachita	39.43	26.49	24.37	9.71	25.1	0.49	6.98	36.33	0.68	45.81	6,356	9,750	12,497
Cal.	Alameda	18.51	35.33	30.67	15.49	10.4	3.05	5.93	47.34	0.66	27.53	8,507	11,479	12,890
Colo.	Adams	19.65	30.75	43.60	6.00	13.0	0.33	6.02	56.54	1.02	30.09	8,638	10,664	11,619
"	Boulder	19.28	34.61	41.41	4.70	5.5	0.39	5.99	57.94	1.28	29.70	9,064	10,094	13,239
Mont.	Chouteau	30.00	40.10	18.00	11.90	19.3	1.08	5.53	40.39	0.80	40.30	6,914	10,049	11,900
"	Yellowstone	24.59	32.86	27.94	14.61	4.3	0.41	4.83	41.41	0.69	38.05	6,208	7,656	10,211
No. Dak.	McLean	35.96	31.92	24.37	7.75	12.7	1.15	6.54	41.43	1.21	41.92	7,069	8,886	12,557
"	Williams	38.92	25.54	30.15	5.39	31.7	0.48	6.89	39.34	0.68	47.22	6,739	10,712	12,101
Ore.	Coos	16.10	31.10	39.63	13.17	8.1	0.81	5.53	51.07	1.19	28.23	9,031	11,471	12,769
"	"	13.77	32.05	46.72	7.46	7.8	4.35	......	......	......	......	9,054	10,684	11,493
So. Dak.	Corson	30.45	22.97	34.43	12.15	19.7	0.39	......	......	......	......	6,944	10,189	12,098
Texas	Houston	34.70	32.23	21.87	11.20	24.6	0.79	6.93	39.25	0.72	41.11	7,056	10,991	13,043
"	Wood	33.71	29.25	29.76	7.28	25.8	0.53	6.79	42.52	0.79	42.09	7,348	10,980	12,452
Utah	Kane	16.59	32.59	37.38	13.44	3.9	3.41	5.39	46.66	0.85	30.25	7,882	9,535	11,264
Wash.	Lewis	27.17	33.80	28.11	10.92	14.3	0.33	6.27	43.88	0.80	37.80	7,569	10,122	12,226
Wyo.	Carbon	10.26	22.23	57.68	9.83	1.7	0.87	2.72	67.94	0.77	17.87	10,354	11,703	12,956
"	Sweetwater	31.37	29.60	23.91	10.12	14.2	1.27	5.43	37.03	0.79	45.36	5,634	8,445	9,630
IX—Miscellaneous Unclassified Coals of the United States														
R. I.	Newport	23.68	3.01	42.54	30.77	23.1	0.03	3.15	42.36	0.10	23.59	5,976	12,954	13,120
"	Providence	2.41	4.92	73.61	19.06	......	0.07	0.65	76.95	0.17	2.63	11,268	......	14,011
Alaska	Bering River	5.71	13.04	47.10	34.15	2.7	6.47	3.34	46.97	0.81	8.26	8,386	13,279	13,945
Ark.	Sebastian	5.26	14.71	55.22	24.81	0.42	1.00	3.91	59.87	1.23	9.18	10,451	13,978	14,945
Idaho	Cassia	34.28	26.64	25.70	13.38	24.0	2.50	......	......	......	......	8,613	13,754	16,457

than 40% volatile matter in the combustible they are liable to an error in either direction of about 4%. It appears from these experiments that the coal of one seam in a given district has the same heating value per pound of combustible within 1 or 2% (true only of some districts), but coals of the same proximate analysis, and containing over 40% volatile matter, but mined in different districts, may vary 6 or 8% in heating value.

Table 5 is condensed from Bulletin 22, U. S. Bureau of Mines, the coals being classified in accordance with Table 2. The several coals given in the table can be compared on the basis of Col. 2, Table 7, by dividing the percentage of each constituent as given in Table 5 by (100−percentage of moisture). They can be compared on the basis of Col. 3, Table 7, by dividing the percentage of each constituent by (100 − percentage of moisture − percentage of ash).

Table 5 gives analyses of coals, also from Bulletin 22, U. S. Bureau of Mines, but which are not classified. The Rhode Island coals are graphitic and are not used as fuel. The volatile in the Alaska and Arkansas coals would class them as semi-bituminous, but they are higher in oxygen and moisture, and of lower heating value than the semi-bituminous coals. The Idaho coal apparently is a cannel coal but the ultimate analysis is lacking.

Evans's Equations.—The heating value of coal can be determined quite accurately from the percentage of volatile matter, provided the locality of the coal mine is known, as shown by F. C. Evans (Bull. No. 3, Cornell Univ. Engg. Expt. Station). Using the empirical relations given in Table 6, the heating value, hydrogen and carbon content of the combustible can be calculated. Thus the ultimate analysis of the dry fuel can be approximated from the proximate analysis, and then converted to the "as-received" basis by making the proper allowance for moisture. Sulphur may be assumed as from 0.5 to 2% in eastern coals, and as up to 8% in mid-western coals. Nitrogen will average about 1.25%, the range usually being from 0.75 to 1.75%. This table does not apply to lignites, peats, woods or cannel coal.

Table 6.—Empirical Relations for Coals of the United States

Based on Proximate Analyses.

Property of Coal Desired	States	Range of Volatile Matter in Combustible, Percent	Equation *
Heating value of the combustible	Pa., Ohio, W. Va., Md., Va., Ky., Ga., Tenn., Ala., Ind., Ia., Neb., Kan., Mo., Okla., Ark., and Texas.	0–16	$H.V. = 14{,}550 + 7{,}810\,V$
		16–36	$H.V. = 16{,}160 - 2{,}250\,V$
		36 up	$H.V. = 18{,}750 - 9{,}440\,V$
	Ill. and Mich.	All values	$H.V. = 16{,}062 - 3{,}830\,V$
Total carbon in the combustible	Penn., Ohio, W. Va., Md., Va., Ky., Ga., Tenn., Ala.	0–36	$C = 0.943 - 0.242\,V$
		36 up	$C = 1.095 - 0.663\,V$
	Ill., Ind., Mich., Ia., Neb., Kan., Mo., Okla., Ark. and Texas	All values	$C = 0.953 - 0.362\,V$
	Col., Utah, N. Mex., Ariz., Wyo., Mont., Wash., Ore. and Cal.	36–60	$C = \dfrac{(H.V. + 7{,}544)(0.0099 + 0.0208V)}{737.5\,V + 200}$
	All states	60 up	$C = \dfrac{(H.V. + 7{,}544)(0.0099 - 0.0045\,V)}{11.3\,V + 200}$
Hydrogen in the combustible	All states but Ark.	4–16	$H = 0.013 + 0.225\,V$
		16 up	$H = 0.0457 + 0.0206\,V$
	Ark.	All values	$H = 0.0327 + 0.056\,V$

* H.V. = high heating value of the combustible, B.t.u. per lb. V = fraction of volatile matter in the combustible. C = fraction of total carbon in the combustible. H = fraction of hydrogen in the combustible. Combustible = Coal − Ash − Moisture = C + V = Unity.

METHODS OF REPORTING COAL ANALYSES.—Three different methods of reporting coal analyses may be used as shown in Table 7. Combustible always is the sum of only the fixed carbon and volatile matter. Combustible often is called "moisture-free and ash-free coal." The method shown in column 2 of Table 4, percentages based on dry coal, is useful for comparing different lots of coal of the same class, and is better than the method in column 1, based on moist coal. Moisture is a variable constituent, depending on the weather, on condition of the atmosphere at the time of analysis and on the extent to which it accidentally may have been dried in sampling.

Table 7.—Methods of Reporting Analyses of Coal

	Moist Coal	Dry Coal	Combustible
Moisture	10	......	......
Volatile matter	30	33.33	37.50
Fixed carbon	50	55.56	62.50
Ash	10	11.11	......
	100	100.00	100.00

Factors Affecting Combustion of Coal

BURNING CHARACTERISTICS OF VARIOUS COALS (Henry Kreisinger and B. J. Cross, *Trans*. A.S.M.E. FSP-50-52, 1928).—Pennsylvania anthracite is used principally for domestic heating and in very small sizes under power plant boilers, either hand-fired or fitted with traveling grate stokers. It burns with a short, nearly transparent flame, due to burning of CO. Any desired rate of combustion can be obtained by regulating the flow of air. A few installations use pulverized anthracite. In this form it is difficult to ignite, and a small amount of air must be mixed with the coal to keep the temperature above the ignition point. A refractory front furnace wall assists ignition by keeping the incoming coal and air mixture above ignition temperature. Powdered anthracite burns with little visible flame.

Pocahontas semi-bituminous coal when burned on a grate tends to fuse and form a crust over the fuel bed, which must be broken by agitation to permit the flow of air. This coal is best burned in underfeed stokers. It pulverizes easily and burns with a short clear flame, comparatively free from smoke. The volatile is distilled off as permanent gases that do not break down to form soot. The fixed carbon residue is dense and burns slowly without visible flame. A large furnace volume is necessary to burn the carbon to ash. Pocahontas coal usually has a higher percentage of combustible in the flue dust than other bituminous coals.

New River coal is similar to Pocahontas in physical structure and its burning characteristics are similar. It is higher in volatile matter, has a very low ash content, and the highest heating value of any American coal.

The Eastern or Appalachian bituminous coals range in volatile matter from 28 to 35%. The Pittsburgh coals are representative. They are coking, and when burned on hand-fired grates or traveling grate stokers, tend to fuse and form a mass of coke. They are best burned in underfeed stokers. This type of coal does not pulverize easily. When pulverized, it burns with a long, luminous flame, which is smoky with inadequate air supply or inadequate mixing. Comparatively less of the coal is burned as fixed carbon, and there is less combustible in the flue dust than when burning Pocahontas or New River coal under similar conditions.

Illinois coals are free burning, *i.e.*, the pieces do not fuse together, but burn separately. They burn well on traveling grate stokers. The ash is more fusible than the ash of Appalachian coals, and agitation of the fuel bed produces clinker. The coal burns with a long luminous flame, and will produce smoke unless sufficient air is supplied over the fire and thoroughly mixed with the volatile matter as it is distilled. Illinois coal pulverizes more easily than the hard Appalachian coals, but not so easily as the Pocahontas type. When pulverized, it burns with a long, luminous flame. The fixed carbon usually is burned quite completely, and the flue dust contains from 5 to 3% of combustible.

Iowa coal is similar to Illinois coal, but is higher in moisture and ash, with a proportionately lower heating value. Kansas, Arkansas and Oklahoma coals also are similar to the Illinois coals, but contain less ash and moisture. They are free-burning and adapted to the traveling grate stoker.

Western sub-bituminous coals do not coke but burn freely, and have a tendency to crumble in the fire. They burn with a long yellow flame and are apt to produce smoke. These coals are soft and pulverize easily. When pulverized, complete combustion easily is obtained. The flue dust contains from 1 to 2% of combustible.

American lignites, when mined, usually contain from 35 to 40% moisture. They disintegrate to slack on exposure to air. In the fire the lumps crumble to pieces $^1/_8$ to $^1/_4$ in. diameter. The smaller pieces are carried out by forced draft, causing a high cinder loss. They burn with a long yellow flame which does not smoke as readily as the flame of bituminous coal. Lignites can be pulverized when the moisture is reduced to 28%, and complete combustion easily is obtained. The flue dust usually contains less than 1% of combustible.

Peat seldom is used as a fuel in the U. S. It is compressed into briquets and dried to a moisture content of about 20%. Peat burns with a long yellow flame similar to that

of lignite. The preparation of peat for the market is too costly to permit it to compete with good coal.

RELATION OF EXCESS AIR, RATE OF FIRING, RADIATION AND EMISSIVITY.—R. A. Sherman has made a laboratory study of the burning characteristics of pulverized Pocahontas, Hocking, Pittsburgh and Illinois coals (*Trans.* A.S.M.E. FSP–56–6, June, 1934). The characteristics of the coals were as follows:

	olatile	Fixed Carbon	Moisture	Ash	B.t.u. per lb.	Softening Temperature of Ash, deg. F.
Ohio, Hocking No. 6......	34.5	52.3	2.3	10.0	12,420	2590
W. Va., Pocahontas No. 3.	17.2	76.6	0.4	5.8	14,790	2630
Ill., No. 6...............	33.6	50.4	1.1	14.9	11,770	2250
Pa., Pittsburgh No. 8.....	40.0	50.2	0.5	9.3	13,200	1985

The study was to determine the relation of excess air to combustion of Hocking and Pocahontas coals, and for all four coals the relation of fineness of grinding to combustion, rate of firing and furnace temperature to completeness of combustion, and radiation and emissivity of flame to excess air, fineness and rate of firing. From curves in the paper the following data are taken: Relation of excess air to completeness of combustion (Table 8); relation of fineness of grinding to completeness of combustion (Table 9); comparative combustion temperatures, radiation and emissivity of the four coals (Table 10); relation of temperature and completeness of combustion to rate of burning (Table 11).

The study showed that unburned carbon continued to decrease up to 30% excess air with Pocahontas coal, and up to 20% with Hocking coal. Fineness of grinding is increasingly important as combustion space is restricted. The optimum limit of fineness differs for different coals. The rate of combustion varies for different coals. Pocahontas coal burns more slowly than any of the four tested. Illinois coal burns more slowly and Pittsburgh coal more rapidly than Hocking coal. Increased furnace temperature apparently increases rate of combustion. Ignition temperature and temperature in the ignition zone are important factors in the combustion zone. Curves for all the coals show a marked increase in the furnace temperature, a considerable decrease in unburned carbon, and a more rapid rate of combustion in the furnace after operating 7 hr. as compared with similar conditions after operating 2 hr. With respect to radiation, the study showed that temperature and total radiation of the flame were affected by fineness of grinding, excess air and rate of firing, but the emissivity of the flame at any one position was affected to a marked degree only by the type of coal. The emissivity of pulverized coal flames ranges from 0.7 to 0.3, decreasing as the carbon burns from the flame.

Coal Specifications

SPECIFICATIONS FOR PURCHASE OF COAL.—The following is an abstract of specifications and contract for coal for steam power plants, prepared by U. S. Bureau of Mines.

Description of the coal should include percentage of moisture in coal as received, and of volatile matter, ash, sulphur and B.t.u. per lb. in dry coal. The source of the coal should be given by naming the mines, their location and operators, and the name of the coal bed. The commercial name of the coal and the railroads on which the mines are located also should be given.

Comparison of Bids is on the basis of the cost of 1,000,000 B.t.u. The method is: 1. All coals are adjusted to the same ash percentage. That coal containing the lowest ash content is taken as the standard. The price quoted is multiplied by one-half the difference between the ash content of the standard and of any other offered coal; the product is added to the price of the non-standard coal.

2. The price as determined by (1) is divided by the difference between 100% and the percentage of moisture guaranteed. The result is computed to the nearest 0.1 cent. 3. The price as determined in (2) is multiplied by 1,000,000 and the result divided by (B.t.u. per lb. guaranteed) × 2240, or by (B.t.u. per lb. guaranteed) × 2000.

After eliminating undesirable bids, the selection of the lowest bid on the basis of the cost per 1,000,000 B.t.u. may be considered by the consumer as a tentative award. The consumer reserves the right to make tests or practical service tests to determine the final award.

Causes for Rejection are 3% more moisture, 4% more ash, 3% more volatile matter, 1% more sulphur or 4% fewer B.t.u. than the specified guarantees; or coal furnished from mines other than those specified, unless upon written consent of the consumer. The consumer has the option of either accepting or rejecting such coal. The consumer also may reject bids offering coals on which he has information that they possess undesirable physical characteristics, volatile matter, sulphur or ash contents; or because of clinkering or excessive refuse; or because they have failed to meet the requirements of city smoke ordinances; or for any other cause that would indicate them to be of a quality that renders them unsuitable for use in the consumer's storage or power plant equipment.

Price is adjusted for variations in heating value, ash and moisture content, from the guaranteed values, as follows:

1. On a "dry-coal" basis, the adjustment shall be proportioned to the variation in B.t.u. content from the guaranteed standard, when such variation exceeds 2%.

2. On a "dry-coal" basis, the adjustment for variations of more than 2% in ash content,

Table 8.—Relation of Unburned Carbon to Excess Air

Fuel	Percentage of Excess Air					
	5	10	15	20	25	30
	Percentage of Unburned Carbon at 10.5 ft. from Burner					
Hocking coal............	1.18	0.59	0.53	0.35	0.24	0.18
Pocahontas coal.........	4.59	3.41	2.71	2.29	1.76	1.41
	Percentage of Unburned Carbon at 8.5 ft. from Burner					
Hocking coal............	2.00	1.32	1.00	0.88	0.81	0.74
Pocahontas coal	6 65	5.62	4.82	4.18	3.65	3.24

Table 9.—Relation of Unburned Carbon to Fineness of Grinding

Fuel	Percentage through 200-mesh Screen					
	40	50	60	70	80	90
	Percentage of Unburned Carbon at 10 5 ft. from Burner					
Hocking................	....	1.51	1.01	0.63	0.38	....
Pocahontas.............	6.06	4.67	3.54	2.53	1.77	1.58
Illinois.................	3.21	2.27	1.58	1.14	1.89	....
Pittsburgh.............	3.16	1.84	1.01	0.63	0.43	...
	Percentage of Unburned Carbon at 8 5 ft. from Burner					
Hocking................	...	4.00	2.59	1.76	1.14	....
Pocahontas.............	10.0	8.21	6.57	5.18	4.42	4.10
Illinois	4.93	4.00	3.03	2.40	1.84	....
Pittsburgh..........	4.73	3.36	2.17	1.26	0.51	....

Table 10.—Temperature, Radiation and Emissivity of Flame

20% Excess Air

Fuel	Distance from Burner, ft.									
			3		5	6		8	9	10
	Temperature, deg. F, 20% Excess Air									
Hocking coal*....	2345	2405	2475	2495	2485	2485	2485	2450	2405	2370
Pocahontas coal†.	2015	2120	2200	2265	2300	2340	2375	2370	2345	2315
Illinois coal‡.... ..	2200	2295	2375	2430	2425	2420	2415	2400	2360	2355
Pittsburgh coal‡..	2360	2405	2450	2450	2450	2450	2450	2420	2395	2360
	Radiation, 1000 B.t.u. per sq. ft. per hr.									
Hocking coal*....	62.2	60.4	60.1	59.0	55.7	51.9	47.4	44.4	41.2	38.5
Pocahontas coal†.	44.4	47.4	50.4	49.8	45.9	43.9	40.0	38.2	38.2	38.2
Illinois coal‡......	60.0	65.2	69.6	67.3	62.5	57.2	53.3	48.9	46.8	44.1
Pittsburgh coal‡..	66.7	65.2	64.6	60.1	55.7	48.8	43.8	40.0	38.5	35.6
	Emissivity §									
Hocking coal*....	0.574	0.524	0.474	0.441	0.408	0.400	0.353	0.344	.338	0.324
Pocahontas coal†.	.676	.618	.600	.524	.491	.412	.353	.344	.340	.353
Illinois coal‡.....	.676	.632	.618	.600	.524	.500	.441	.412	.406	.398
Pittsburgh coal‡	.603	.559	.529	.485	.441	.400	.353	.332	.309	.309

*80% through 200 mesh; †73% through 200 mesh; ‡85% through 200 mesh; § Emissivity = (Radiation from flame) ÷ (Radiation from black body at flame temperature).

Table 11—Variation of Combustion and Temperature with Rate of Burning

Excess Air, 20%; through 200-mesh, 75 to 79%; rate of heat input 2,650,000–2,710,000 B t u. per hr.

Time, Milli-seconds	Hocking		Pocahontas		Illinois		Pittsburgh	
	Temp., deg. F.	Unburned Carbon, percent	Temp., deg. F.	Unburned Carbon, percent	Temp., deg. F.	Unburned Carbon, percent	Temp., deg F.	Unburned Carbon, percent
25	2260	....	1935	....	2075	....	2170	....
50	2480	....	2240	....	2390	....	2555	....
62	2520 *	13.2	2333	36.4	....	....	....	....
64	....	....	....	....	....	....	2625 *	12.8
68	....	....	....	....	2475*	20.7	....	....
100	2475	6.4	2475	16.9	2450	6.4	2565	6.4
105	....	....	2495*	15.0	....	....	....	....
150	2430	2.9	2475	10.5	2405	7.9	2515	2.9
200	2405	1.3	2475	5.4	2370	1.5	2480	1.5
250	2370	0.9	2450	3.9	2345	1.0	2460	1.0
300	2330	0.7	2405	2.9	2335	0.9	2425	0.9
350	....	....	....	....	....	....	2405	0.7

* Maximum temperature.

above or below the guaranteed standard shall be determined as follows: One-half the difference between the guaranteed ash content and the ash content as determined by analysis is multiplied by the bid price. The result is added to the price adjusted for B.t.u., if the ash content by analysis is below that guaranteed, or subtracted from it if the ash content by analysis is higher than the guaranteed content. Adjustment is to be figured to the nearest 0.1 cent.

EXAMPLE.—Guaranteed ash content, 10%; ash content by analysis, 13.25%; bid price, $3.00 per ton. Adjustment = (0.1325 − 10)/2 × $3.00 = 0.049 per ton deduction.

3. Adjustment for moisture in excess of that guaranteed is determined by multiplying the bid price by the percentage of moisture in excess of the guaranteed percentage, and dividing the product by 100.

EXAMPLE.—Guaranteed moisture, 3%; actual moisture, 4.58%; bid price, $3.00 per ton. Adjustment = (4.58 − 3) × $3.00/100 = 4.7 cents per ton deduction.

Miscellaneous Notes on Coal

ANTHRACITE SILT AND CULM is the product of the wet preparation of standard market sizes of anthracite. It ranges in size from 1/32 in. to dust, and was long considered a waste product. It, however, may be burned under boilers, either as pulverized fuel or in stokers, at a considerable saving in fuel cost. F. H. Dechant states that the design of the combustion chamber is an important element in burning it in stokers. The ignition arch should be short, and the rear arch long to prevent loss of unburned coal lifted from the grates. Baffles should be steeply pitched to prevent accumulation of fly ash. As a pulverized fuel it is best burned on the unit system, using a ball mill for pulverinzing. Data on installations using this fuel are as follows.

Plant No.	1	2	3	4	5	6	7	8	
Type of Stokers *	CG	C	H	C	C	C	C	..	M
Type of Pulverizer †	..	..	..	..	..	BM	..	BM	
Forced Draft, in. of Water	5	5	4	3	3	5	4		
Thickness of Fuel Bed, in.	6–8	6	5	5	5	6	5		

* CG = chain grate; C = Coxe; H = Harrington; M = Metropolitan; † BM = ball mill.

See also paper by F. S. Dechant, *Trans.* A.S.M.E., FSP-52-7, 1930.

SIZES OF ANTHRACITE AND SPACE OCCUPIED.—Table 12 gives the names and screen sizes of anthracite recommended by the Anthracite Operators Conference (now Anthracite Institute) 1929, and included in A.S.T.M. specification D310–31. The holes in the screens are round and staggered.

The space occupied by anthracite varies with the size of the coal. U. S. Bureau of Mines Bulletin 184 gives a table of various coals, from which the following figures are taken: Furnace, 52–55 lb. per cu. ft.; egg, 53–58 lb. per cu. ft.; chestnut, 52.5–56.5 lb. per cu. ft.; pea, 53.5-54.5 lb. per cu. ft.; No. 1 buckwheat, 50.5 lb. per cu. ft.

Table 12.—Screen Sizes of Anthracite

Size	Passes through	Retained on	Size	Passes through	Retained on
Broken	4 7/16 in.	3 7/16 in.	Pea	12/16 in.	8/16 in.
Egg	3 7/16 "	2 8/16 "	No. 1 Buckwheat	9/16 "	5/16 "
Stove	2 8/16 "	1 9/16 "	No. 2 " (Rice)	5/16 "	3/16 "
Chestnut	1 9/16 "	11/16 "	No. 3 " (Barley)	3/16 "	3/32 "

Screening affects the purity of the different sizes as regards ash. Table 13 shows the analyses of samples from one mine.

Table 13.—Effect of Screening of Anthracite upon Ash Content

Size	Screened		Analyses, percent	
	Through	Over	Fixed Carbon	Ash
Egg	2.5 in.	1.75 in.	88.49	5.66
Stove	1.75 "	1.25 "	83.67	10.17
Chestnut	1.25 "	0.75 "	80.72	12.67
Pea	0.75 "	0.50 "	79.05	14.66
Buckwheat	0.50 "	0.25 "	76.92	16.62

SPACE OCCUPIED BY BITUMINOUS COAL varies with the specific gravity of the coal, the relative proportions of lump and slack, the moisture content and the degree of settling to which it has been subjected The U. S. Bureau of Mines (Bulletin 184) reports a reduction in volume, due to shaking of the containers, of 4.2 to 8.34%. Table 14 shows the range of weights of bituminous and semi-bituminous coals given in Bulletin 184. The Bureau of Mines states that, other things being equal, the sample with the higher moisture content will weigh more per cubic foot and occupy greater space per

pound of dry coal than will a sample of lower moisture content. Increase in volume of wet coal is not proportionately as great as the increase in weight per cubic foot. Slack, comprising a mixture up to and including nut size, weighs more than screened nut coal.

Table 14.—Weight of Bituminous and Semi-bituminous Coal

Coal from	Size *	Lb. per cu. ft.	Coal from	Size *	Lb. per cu. ft.
Alabama	D.	45.5	Oklahoma	40–20–20	50.0
"	R.M.	51–54	"	35–45–20	48.5
Arkansas	R.M.	49.5–59.5	Pennsylvania	90– 5– 5	47–49.5
Colorado	Lump	50.5–52.5	"	70–20–10	50.5
"	D.	49.5	"	60–25–10	50.5
Georgia	60–10–30	54	"	20–30–50	52
Illinois	D. Lump	49.5	"	10–15–75	52
"	R.M.	54.5–55.5	"	0–10–90	49.5–53.5
"	Lump	44–48.5	"	0– 0–100	52
Iowa	60–25–15	46.5	"	Lump	46.5
Kansas	95– 5– 0	55.5	Utah	90– 0– 5	44.5
Kentucky	95– 5– 0	43.0–54.5	West Virginia †	75–15–10	55.5
"	Lump	45–47.5	" "	60–30–10	47.0
Montana	90– 5– 5	52	" "	20–10–70	55.0
Ohio	90– 5– 5	49	" "	5–10–85	55.5
"	70–15–15	47.5	" "	4– 2–94	54
"	60–30–10	46.5	" "	3– 5–92	57.5
"	40–20–20	50.0	" "	0– 5–95	56.5

* D = Domestic; R.M. = Run-of-Mine; the figures represent the respective percentages of lump, nut and slack. † Semi-bituminous.

SLIDING ANGLES OF COAL.—Table 15, from data in U. S. Bureau of Mines Bulletin 184, shows angles at which various classes and sizes of coal and other substances will slide on steel chutes. Table 16 from the same source gives sliding angles of coke.

Table 15.—Angle of Sliding of Coal and Ore on Bright Steel

Material	Angle at which sliding starts	Angle at which sliding continues
Penn. anthracite, egg	15° 40′	14° 00′
" " chestnut	16° 40′	15° 10′
Ill. bituminous, 6″ lump	21° 50′	20° 40′
" " 3″ × 6″ egg	21° 00′	19° 10′
" " 3″ × 2″ nut	20° 30′	19° 00′
" " 1 1/4″ × 2″ nut	20° 40′	19° 20′
" " 3/4″ × 1 1/4″ nut	21° 00′	19° 40′
" " slack	25° 40′	22° 30′
Okla. screenings	24° 00′	22° 30′
" chestnut	21° 00′	19° 00′
East. Ky. egg	21° 50′	20° 20′
Bituminous shale	21° 10′	20° 00′
Limestone gangue ores	19° 40′	17° 30′
Sandstone " "	22° 30′	19° 40′
Hematite	22° 40′	20° 40′
Missouri Galena	19° 20′	17° 20′

Table 16.—Angle of Repose and Sliding Angles of Coke on Coke

Size	Angle of Repose, deg.		Sliding Angle, deg.	
	Wet	Dry	Wet	Dry
Furnace	43.0	44.3	45.8	46.9
Stove	39.6	40.6	43.4	43.6
Breeze	37.1	39 9	43.3	41 6

MIXTURES OF HARD AND SOFT COAL for hand-fired boilers as shown in Table 17 are given in Tech. Paper 220, U. S. Bureau of Mines. The coking qualities of the soft coal prevent draft and grate losses of the fine anthracite, while the higher fixed carbon of the anthracite increases the heating value of the mixture.

Table 17.—Proportions of Soft and Hard Coal To Be Used for Hand Firing

		Soft Coal	Hard Coal
No. 1 Buckwheat	Forced draft	30%	70%
	Natural draft	40	60
No. 2 Buckwheat	Forced draft	40	60
	Natural draft	50	50
No. 3 Buckwheat	Forced draft	50	50
	Natural draft	65	35

WEATHERING OF COAL tends to effect a small decrease in heating value, and to produce mechanical disintegration when pyrites are present. Weathering of anthracite is confined to oxidation of the pyrites, but in coking coal it finally may destroy the coking power. Bulletin 136, U. S. Bureau of Mines, shows New River coal under outdoor exposure to deteriorate in heating value 1% in one year, 2% in 2 years and not over 3% in 3 years. Pittsburgh coal deteriorated 1.1% in 5 years; Pocahontas coal deteriorated 0.4% in 6 months and an additional 0.4% at the end of the second year. Illinois coal will deteriorate about 4% in storage. In a closed bin Wyoming coal deteriorated 4.57% at the end of 9 months and 5.26% at the end of 2 3/4 years.

Underwater storage of coal seems to prevent deterioration. S. D. Parr and N. D. Hamilton (Bulletin 17, Univ. of Ill. Engg. Expt. Station) conclude from experiments on 100 lb. samples: 1. [illegible] coal does not decrease appreciably in heating value. 2. Outdoor exposure causes a loss of heating value of 2 to 10%. 3. [illegible] advantage over open storage, except with high sulphur coals. 4. In most cases deterioration appears to be practically complete at the end of 5 months.

SPONTANEOUS COMBUSTION OF COAL is due to slow oxidation in an air supply insufficient to carry away the heat so formed. An ideal condition for spontaneous heating is mixed lump and fine coal, with a high percentage of dust, so piled as to admit a small amount of air. A high percentage of volatile does not of itself increase liability to spontaneous heating. The effect of moisture and sulphur is uncertain. Excessive moisture may cause fine coal to form a blanket, impervious to air, thus restricting the air supply to a point where spontaneous heating will ensue. The U. S. Bureau of Mines states that sulphur has only a minor effect, but a Nova Scotia coal containing 3 to 4% sulphur is reported to have given much trouble from spontaneous fires in storage.

The surfaces of freshly mined or freshly crushed coal oxidize readily. Such coal is quite liable to spontaneous heating in storage. In time, the surfaces are covered with oxide and tendency to heat is abated. If coal that has heated after six weeks or two months storage be rehandled and thoroughly cooled, it seldom will reheat spontaneously.

Tech. Paper 16, U. S. Bureau of Mines gives the following rules for storage of coal to avoid spontaneous combustion: Pile to a maximum depth of 12 ft., with no point in the interior over 10 ft. from an air cooled surface; if possible, store only in lump; reduce handling to a minimum to avoid dust formation; distribute lump and fine as evenly as possible; rehandle and screen after two months; keep away external sources of heat, as hot walls, steam pipes, oily waste, etc.; season coal six weeks after mining before storage; avoid alternate wetting and drying; avoid admitting air to interior of pile through interstices around timbers, irregular brickwork or porous bottoms (coarse cinders, etc.); avoid ventilation by pipes through the pile, which often is dangerous.

Coal should be so stored that its temperature can be ascertained daily. If the temperature rises to 160° F., a destructive temperature almost certainly will ensue, unless the warm coal is removed and cooled. See also Factors in Spontaneous Combustion of Coal, O. P. Hood, *Mech. Engg.*, Dec., 1923.

Ash

Fusibility of Ash.—Ash is a mixture of slate, clay, silica, iron oxide, lime, etc., each with a different fusion point, and all tending to react with each other under high heat to form slag. The temperature range of slag formation varies, depending on the composition of the ash, from 20 to 50° F. to over several hundred degrees. A high percentage of iron or lime causes the ash to melt at from 1800 to 2200° F., while high percentages of alumina and silica cause it to melt at from 2600 to 3000° F. The fusion of ash and formation of clinker were investigated by A. C. Fieldner, W. A. Selvig and P. Nicholls (*Trans.* A.S.M.E. FSP-50-6, 1928). Twenty-one samples of coal, as shown by Table 18 were studied. The analyses of the ash given are in Table 19. The fusion temperatures of the ash were determined by the standard method of the A.S.T.M. (A.S.T.M. Stds., 1924, p. 994). The softening temperature is that at which the test cone fuses down to a spherical lump. The fluid temperature is that at which the ash becomes fluid and spreads out in a flat layer. The study showed that the fusibility of ash depends on the ratio of the silica to the bases, the percentage of alumina, and on the bases present. Ash high in silica and alumina is not readily fusible and, usually, ash low in iron and lime have high silica and alumina. Coals high in sulphur generally will clinker readily.

The formation of clinker in relation to fusion temperatures also was studied. Due to the complexity of the reactions involved no absolute values can be given, but the following general conclusions seem to be true: Quantity of clinker formed decreases as rate of burning increases; density and fusion of the clinker is greater at the higher rates; the rate of decrease of quantity with rate of burning becomes less when ash fusion temperatures are low. The quantity of clinker has a definite relation to the softening temperatures of the ash. It decreases with increase of temperature and has small change for coals with

softening temperatures over 2600° F. The relation becomes more consistent as size of coal decreases. The washed coals (Nos. 9 and 20, Table 18) showed a reduction of clinker formation greater than that due to the proportion of inorganic matter removed.

Table 18.—Analyses and Ash Fusion Temperatures of Various Coals

Coal in as received condition

Sample No.	Grade *	State	County	Bed	Moisture, percent	Fixed Carbon, percent	Volatile Matter, percent	Ash, percent	Sulphur, percent	Heating Value, B.t.u. per lb.	Ash Fusion Temperatures, deg. F. Initial	Softening	Fluid
1	SB	Pa.	Somerset....	B........	1.7	75.4	15.9	7.0	0.8	14,280	2550	2930	a
2	SB	Md.	Allegheny....	Big Vein..	1.0	72.1	19.1	7.8	0.9	14,260	2840	2930	a
3	SB	Pa.	Clearfield....	B........	1.5	65.9	24.2	8.4	1.9	14,120	2450	2520	2580
4	SB	Pa.	Somerset....	C Prime.	1.6	72.8	15.7	9.9	2.0	13,770	2180	2440	2580
5	SB	Pa.	Cambria.....	Miller or B	1.2	72.3	21.2	5.3	1.2	14,670	2520	2650	2710
6	B	Ohio	Meigs	8-A......	5.2	45.0	37.6	12.2	2.4	11,820	2020	2190	2390
7	B	Ill.	Williamson...	No. 6....	9.6	44.4	33.4	12.6	3.6	11,260	1960	2070	2290
8	B	Pa.	Westmoreland	Pittsburgh	1.3	55.4	32.3	11.0	1.5	13,390	2460	2600	2700
9†	B	Pa.	Westmoreland	Pittsburgh	1.3	57.0	33.5	8.2	1.6	13,890	2460	2570	2700
10	SB	W.Va.	New River Coal....		1.6	70.5	20.7	7.2	1.0	14,240	2440	2580	2630
11	SB	W.Va.	Pocahontas Coal....		1.8	71.3	20.9	6.0	0.6	14,480	2160	2300	2440
12	B	Ohio	Jefferson.....	Pittsburgh No. 8...	2.2	53.3	35.6	8.9	2.2	13,280	2090	2210	2330
13	SB	W.Va.	Raleigh......	Beckley...	1.5	73.8	17.6	7.1	0.8	14,310	2630	2800	2850
14	SB	Pa.	Mercer......	Brookville	1.5	70.2	17.7	10.6	1.4	13,640	2390	2640	2830
15	B	Ill.	Williamson...	No. 6....	5.6	50.4	33.4	10.6	2.0	12,130	2110	2280	2460
16	B	Pa.	Westmoreland	Pittsburgh	1.9	54.8	32.5	10.8	1.8	13,200	2360	2520	2640
17	B	Pa.	Allegheny...	Pittsburgh	2.4	54.0	34.2	9.4	1.4	13,280	2100	2270	2430
18	B	Pa.	Westmoreland	Pittsburgh	1.4	58.2	33.8	6.6	0.8	14,080	2580	2730	2840
19	SB	Pa.	Somerset....	Miller or B	1.3	72.8	16.7	9.2	1.9	13,940	2390	2470	2560
20†	SB	Pa.	Somerset ..	Miller or B	1.3	74.1	16.7	7.9	1.4	14,200	2500	2630	2720
21	B	Ill.	Mixture from 7 mines...		12.5	38.3	33.0	16.2	3.2	10,190	1930	1990	2170

* SB = semi-bituminous; B = bituminous. † Washed coal, same as next preceding sample. *a.* Did not attain temperature of fluidity.

Table 19.—Analyses of Ash in Coals of Table 18

Sample	SiO_2	Al_2O_3	Fe_2O_3	TiO_2	P_2O_5	CaO	MgO	Na_2O	K_2O	SO_3
1	40.4	38.6	4.7	1.5	4.0	5.6	0.5	0.4	0.9	2.6
2	54.7	33.7	4.3	1.8	0.50	0.7	0.5	0.7	1.8	0.8
3	43.6	29.3	19.9	1.3	0.14	1.4	0.6	0.3	1.1	2.0
4	43.5	27.1	20.8	1.3	0.09	2.1	0.7	0.4	1.7	1.7
5	43.9	33.4	15.0	1.3	0.13	1.9	0.3	0.3	1.0	2.5
6	45.7	24.2	19.7	1.0	0.17	3.6	0.8	0.4	2.1	1.6
7	41.2	15.9	23.1	0.8	0.12	9.4	0.4	0.6	1.9	7.4
8	55.5	26.6	10.9	1.5	0.38	0.9	0.9	0.7	1.5	0.6
9	53.8	27.5	13.0	1.4	0.30	1.0	0.4	0.7	1.3	0.5
10	50.8	28.5	13.4	1.2	0.13	1.1	1.0	1.0	2.3	0.3
11	51.2	24.0	10.6	1.8	0.09	4.8	2.0	0.6	1.7	2.5
12	47.2	23.2	21.9	1.0	0.22	2.4	0.6	0.4	1.6	1.6
13	53.5	30.4	9.4	1.3	0.18	0.9	1.0	0.9	2.1	0.3
14	47.4	31.1	10.3	1.8	1.9	2.7	0.8	0.5	2.1	1.0
15	55.5	22.7	12.7	1.0	0.05	2.1	1.3	0.5	2.5	1.9
16	54.9	26.3	12.0	1.3	0.17	0.5	0.6	0.9	1.9	0.7
17	52.4	23.3	11.5	1.0	0.27	3.7	1.0	0.6	1.8	3.9
18	55.7	28.3	7.3	1.5	0.62	2.4	0.8	0.6	0.9	1.3
19	38.6	31.1	24.2	1.3	0.30	1.6	0.4	0.5	0.8	1.1
20	41.3	33.1	19.3	1.3	0.35	1.8	0.3	0.4	0.8	1.4
21	41.2	13.1	14.9	0.8	0.22	15.8	0.6	0.8	1.8	10.1

THE EFFECT OF ASH is to reduce boiler efficiency. Table 20 shows the decrease of efficiency with increasing percentages of ash in the coal. See article by G. A. Orrok, *Elec. Wld.*, Oct. 11, 1919.

Table 20.—Average Loss of Boiler Efficiency Due to Ash in Coal

Percentage of Ash.......	10	20	30	40	50
Anthracite coal.........	12	23	45	70	100
Bituminous coal.........	10	20	40	75	100
Western coal...........	5	18	32	98	...

2. PULVERIZED COAL

Pulverized coal is used for firing ing cement and lime in rotary kilns and other metallurgical furnaces. I· , as the heat libera- tion per cubic foot of combustior other method of burn- ing coal. Two general systems .em.

THE BIN SYSTEM

the furnaces.

The raw coal is crushed to a uniform › the pul may or not be dried before entering the claim drying is when the moisture does not exceed 4%. ..ed by the flue gases is used, being passed either through a coal dr· the pulverizer, reducing the moisture to about 2%. Fig. 1 from a paper by ... (*Trans.* A.S.M.E., FSP-53–4, 1931) comparing the performance of a boiler under bin and unit systems at the Calumet station of the Commonwealth Edison Co., Chicago, shows a diagrammatic arrangement of the bin system. For a complete description of the bin system, including various arrangements of equipment, see paper by Henry Kreisinger, *Trans.* A.S.M.E., FSP–52–36, 1930.

THE UNIT SYSTEM prepares the coal as required by the burners. One or more pulverizers are provided for each boiler, coal being fed directly to them from the slack coal bins. The coal is not dried before feeding to the pulverizers, but air heated to about 150° F. is blown into the pulverizer, dries the coal and blows pulverized fuel directly to the burner. At the burner, air from the air preheater, at 400 to 500° F. enters with the coal. While the capital investment is lower with the unit system, the capacity of the boiler is limited by the capacity of the pulverizer equipment, and shutting down the

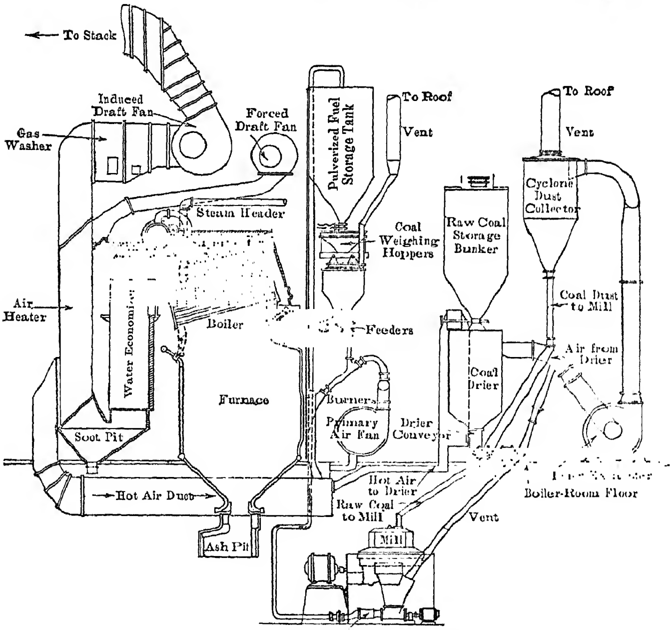


FIG. 1. Bin System of Pulverized Coal

pulverizer means shutting down the boiler also. Compensating advantages are that no storage of pulverized coal is required, eliminating explosion hazards. No preliminary drying of coal is necessary and the transportation system for coal is eliminated. Fig. 2 shows diagrammatically the boiler shown in Fig. 1 as rearranged for the unit system.

COMPARISON OF BIN AND UNIT SYSTEMS.—The equipment required for both bin and unit systems is shown in a paper by A. E. Grunert (*Trans.* A.S.M.E. FSP–53–4, 1931), comparing performance of a boiler of 5940 sq. ft. of heating surface with both bin and unit systems of firing. The equipment required for the bin system was: Raw coal storage bunker; air preheater, with induced draft fan; forced draft fans; gas washer equipment; coal dryer; dryer exhauster; cyclones; drag feeder; pulverizer; pulverized coal transport system; pulverized coal bunker; pulverized coal feeders; pulverized coal weigh bins and scales; a primary air fan; 8 burners. The equipment for the unit system for the same boiler was: Raw coal storage bunker; air preheater; induced draft fan; forced draft fans; air washer; raw coal scales; drag feeder; pulverizer; mill exhauster fan; 6 burners. The evaporative efficiency of the boiler was about the same under both systems, but the auxiliary power consumption of the unit system was lower. On the other hand, the maintenance and labor costs of the unit system were higher. There was but little difference in the total cost of steam generated by the two systems. In a report on pulverized fuel by the Prime-movers Committee of the National Electric Light Assoc., 1931 (now Edison Institute), a number of operating companies report experiences with both systems, tending to confirm the conclusions of Mr. Grunert. Reliability appears to be higher in the bin system while flexibility and efficiency are better with the unit system. Certain companies reported maintenance cost to be lower with the bin systems, while others reported it to be higher. In general, it would appear that so far as general operating characteristics are concerned there is but little to choose between the two systems, and the choice of one or the other should be based on local conditions.

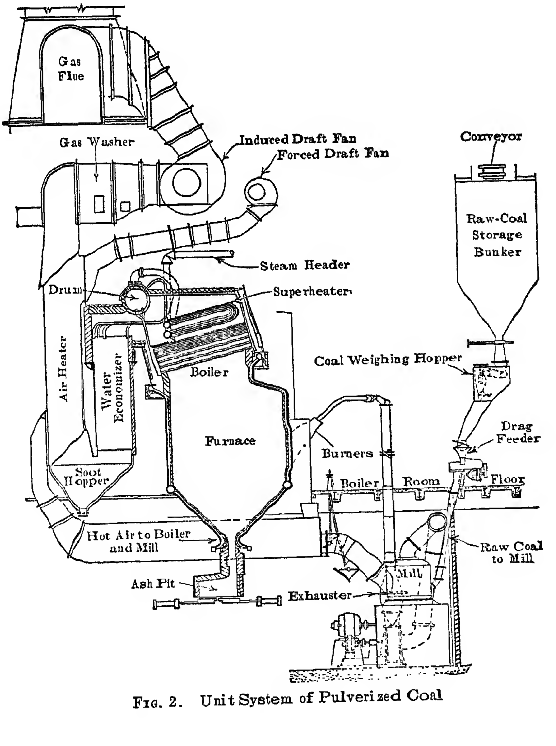


Fig. 2. Unit System of Pulverized Coal

BEST FINENESS OF PULVERIZATION.—The degree to which the coal is pulverized has a direct relation to the efficiency of the steam generating plant. The degree of pulverization varies with grade of coal and size of combustion chamber. If the fuel contains a considerable quantity of relatively coarse particles a larger furnace volume is necessary to permit these particles to be completely burned before striking the furnace walls. However, an excessive degree of pulverization increases the cost of coal preparation more rapidly than the corresponding saving due to increased boiler efficiency. E. H. Tenney (*Trans.* A.S.M.E. FSP–54–7, 1932) shows that with the coal used at the Cahokia Station of the Union Electric Light and Power Co., St. Louis, Mo., the maximum boiler efficiency was obtained when 75% of the coal passed through a 200-mesh screen, but that, due to the lower cost of pulverization, the total cost of steam was lowest when 57% of the coal passed through 200-mesh screen. Henry Kreisinger (*Trans.* A.S.M.E. FSP–54–9, 1932) gives the figures of Table 21 as the most economical fineness for various coals, considering the cost of pulverization and losses from incomplete combustion. Table 22 shows standard sieve sizes for fineness of powdered coal.

Table 21.—Economic Fineness for Various Coals

	Percentage passing through		
	200-Mesh	100-Mesh	50-Mesh
Anthracite	85	97	100
Eastern Low Volatile	75	92	98
Eastern High Volatile	65	85	97
Illinois	55	80	92
Sub-bituminous	45	75	90
Lignites	35	65	88

Table 22.—Standard Sieve Sizes for Fineness of Pulverized Coal

Sizes		Sieve Opening, in.	Wire Diam., in.	Tolerances, percent			
U. S. Standard Sieve Series No.	Microns			In Average Opening	Wire Diam. Under	Wire Diam. Over	In Maximum Opening
16	1190	0.0469	0.0213	± 3	15	30	10
30	590	.0232	.0130	± 5	15	30	25
50	297	.0117	.0074	± 6	15	35	40
100	149	.0059	.0040	± 6	15	35	40
200	74	.0029	.0021	± 8	15	35	60

To determine the fineness of coal delivered by the pulverizers, sampling of the coal stream in the feeder ducts is necessary. The A.S.T.M. tentative standard calls for a thin-walled brass elbow moved through a stuffing box uniformly across the diameter of duct. The accuracy of this method has been questioned, as in horizontal ducts heavier particles tend to flow along the bottom of the duct and do not redistribute themselves uniformly in vertical ducts. The Duquesne Light Co., Pittsburgh, Pa., collects samples in a quart Mason jar moved across the cross-section of a vertical duct connecting the bottom of the cyclone separators with the storage bin. The Detroit Edison Co. has found a double-barreled thief sampler, so arranged that the open slot can be closed before removal from the coal stream, to be satisfactory. Prof. W. A. Willis (*Power*, Dec. 18, 1928) has developed a method of taking samples from the coal and air stream in the duct between pulverizers and burners, 4 ft. above the fan, in which he placed the opening of the sampler 7 1/2 in. from one end on the center line of the duct, and obtained samples of the same fineness as were obtained by a traverse of the duct in twelve equal areas. Horace C. Porter (*Trans.* A.S.M.E. FSP–52–16, 1930) discusses various methods of sampling and gives the following as essential for securing good samples: Samples should be taken from a vertical duct, if taken from the air-coal stream. They should be taken at least 10 ft. away from an elbow, branch or other irregularity, and as far away as possible from disturbing action of fans or blowers. A stuffing box should be used through which a sampling pipe with a right-angle bend facing the air stream and without irregularities on its inner surface, can be moved freely, which tube should be moved systematically to different positions in the cross-section of the duct to correspond in its time schedule with known variations in density of the coal-air mixture. Samples should be taken at several different places in the piping system and mixed to obtain an average.

VELOCITY OF PARTICLES IN FURNACE.—Air velocity in the furnace should be such as to carry the fuel particles in the air stream until they are consumed, otherwise the coarser ones will fall to the bottom of the furnace unburned, with a resulting uncon-d carbon loss. The rate of fall of particles of fuel in still air varies with the diam-

eter. E. G. Bailey (*Trans.* A.S.M.E. FSP-50-72, 1928) presents a curve showing the velocity of fall for various diameters of particles as follows:

Diameter, in......	0.001	.002	.003	.004	.005	.006	.007	.008	.009	.010	.011
Velocity, ft. per sec.	0.16	0.48	0.93	1.41	2.14	3.10	4.19	5.48	6.97	8.48	10.2

The vertical velocity of the gases in the furnace must be higher than the figures above given if the particle is not to fall out of the air stream. R. A. Sherman (*Trans.* A.S.M.E. FSP-54-7, 1932) points out that these figures should not be applied to horizontally moving gases. For horizontal movement of gases the terminal velocity of the particle as above given is the downward component of movement, and should be used as such in calculating the velocity required in ducts and furnaces. See also analysis of this subject by John Blizard, *Jour.* Franklin Inst., vol. 197, 1924.

RATE OF COMBUSTION AND IGNITION TEMPERATURES.—The rate of combustion depends on the character of fuel, the temperature of the furnace and the size of the particles. Volatile matter is consumed rapidly, and the fixed carbon more slowly. Hence, high volatile fuels tend to burn at a higher rate than low volatile. The relation between furnace temperature and rate of combustion is discussed by Griffin, Adams and Smith (*Indust. and Engg. Chem.*, Sept., 1929). They present curves from which the following figures are taken:

Furnace temperature, deg. F...	1300	1400	1500	1600	1700	1800
Size			Time, milli-seconds			
−80 + 90 mesh..............	100	125	145	170	185	205
−50 + 60 "	....	285	330	370	415	455
−45 + 40 "	....	320	385	440	500	560

It should be noted from the above figures that the higher the furnace temperature the slower the rate of combustion. Several theories have been advanced to account for this negative temperature coefficient. See paper by E. H. Tenney (*Trans.* A.S.M.E. FSP-54-7, 1932), which discusses rate of combustion. Mr. Tenney states that the rate of delivering air to the particle of fuel is probably more important than combustibility of the fuel, and since combustion occurs only at the surface of the particle the advantage of great fineness is evident. As each particle of coal dust requires for complete combustion 35,000 times its volume of air, the quick dispersion of the coal particles into the centers of such spheres of air is important, and requires considerable turbulence, which is increased with excess air. Turbulence further scrubs away products of combustion and facilitates delivery of fresh air. Henry Kreisinger (*Trans.* A.S.M.E. FSP-54-9, 1932) also discusses this subject and develops an equation for the rate of contact between oxygen and combustibles. He concludes that intensive mixing in the furnace is the most important factor for producing rapid combustion.

The combustion chamber temperature necessary for the ignition of fuel varies with fineness of the coal. E. H. Tenney (*Trans.* A.S.M.E. FSP-54-7, 1932) shows this relation for one kind of coal to be as follows:

Diameter of particle, in..........	up to 0.002	0.003	0.004	0.0045
Furnace temperature, deg. F......	1300	1500	1900	2200

It should be noted that the above are furnace temperatures and not the ignition temperatures of the fuel.

HEAT LIBERATION.—The rate of heat liberation per cu. ft. of furnace volume per hour with pulverized fuel should not exceed 20,000 B.t.u. with refractory furnaces under continuous operation. Where the ash is fusible, a rate of 15,000 B.t.u. usually is the maximum that should be attempted. Water-cooled furnaces will permit a rate of 30,000 B.t.u. for continuous operation and a peak rate of 35,000 B.t.u. (Henry Kreisinger, *Trans.* A.S.M.E. FSP-54-9, 1932). These figures are in excess of those usually reached in practice. A study of operating practice of 250 central stations, by the Central Station Committee of the Power Division of the A.S.M.E. (C. F. Hirshfeld and G. U. Moran, *Trans.* A.S.M.E. FSP-55-8, 1933) shows the following heat outputs in superheated steam B.t.u. per cu. ft. of furnace volume per hr.: Water-cooled furnaces, maximum, 21,800; minimum, 8100; average, 13,084; air- and water-cooled furnaces, maximum, 12,300; minimum, 7000; average, 9114; air-cooled furnaces, maximum, 13,700; minimum, 5900; average, 8992. Boiler efficiencies are not given in the report, but assuming that the maximum efficiency is only 75%, the heat liberations per cu. ft. of furnace volume would be as follows: Water-cooled furnaces, maximum, 29,065; minimum, 10,800; average, 17,445; air- and water-cooled furnaces, maximum, 16,400; minimum, 9330; average, 12,150; air-cooled furnaces, maximum, 18,266; minimum, 7865; average, 11,989.

COMPLETENESS OF COMBUSTION varies with furnace temperature, fineness of grinding and excess air. See Burning Characteristics of Coal, p. 4–21.

METHODS OF FIRING.—The three general systems of firing pulverized coal are vertical, horizontal and tangential. Modifications of these systems also are used.

The Vertical System fires the coal-air mixture vertically downwards. Burners are located in the arch of the furnace. Primary air enters through the burner and secondary air through ports in the front wall. The two streams impinge at right angles, causing turbulence and mixing. The coal-air mixture takes a U-shaped path through the fur- : the bottom and flowing up through the boiler tubes. Secondary air f 1 to 2 in. of water. Comb

angle.

The Horizontal System fires the fuel and primary air through burners wall of the furnace. The secondary air may be preheated and is supplied sure through the burner. Both primary and secondary air streams are caused to rotate as they leave the burner to intensify mixing and tubulence. A modification of the horizontal is the double opposed system. Rows of burners are placed in opposite walls of the furnace. The coal-air streams impinge at the center, producing high turbulence and xing.

gential System uses four burners, one in each corner of the furnace and in located that the coal-air streams are tangent to the length of one side of the furnace. The four streams impinge on each other at right angles, with resulting violent turb the mixture completely filling the furnace. High rate of heat liberation thus is obtained. From 25 to 50% of the air supply is primary air. The balance is supplied under pressure through the burner around the coal nozzle This system is applicable only to completely water-cooled furnaces.

COAL PULVERIZERS reduce the coal to the desired fineness either by impact or attrition. The impact mill comprises a series of hammers mounted on a shaft. When apidly revolved the hammers break the coal up to the desired degree of fineness. The n mill crushes the coal between balls or rolls and stationary or revolving rings. Typical machines of the two types are described below. Approximate dimensions, capacity and power consumption are given in Tables 23 to 25. In using these tables, it should be noted that capacity, power required, and degree of pulverization will vary with every grade of coal, and the figures are for general guidance only.

Impact Mills.—The Raymond mill, Fig. 3, and Table 23 is a typical impact machine. Swinging hammers *A* on the shaft *B* crush the coal as it is fed from the feeder *C*. The pulverized coal is drawn through the conical outlet *D* by the exhaust fan *E*, whence it is delivered to storage bin or coal burner. The deflector *F* on the shaft *B* rejects the coarser particles and returns them to the hammers for further pulverization. The degree of fineness can be regulated by moving *F* axially along the shaft.

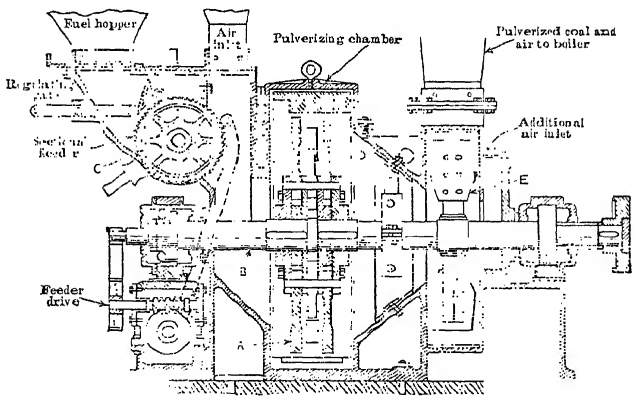


FIG. 3. Raymond Impact Pulverizer

Table 23.—Dimensions and Capacities of Raymond Impact Mills

(Combustion Engineering Corp., New York, 1935)

Capacities are based on fineness of 65% through 200-mesh screen. Power requirements include power for both mill and exhauster.

Size	Overall Dimensions, in.			Eastern Low Volatile			Eastern High Volatile			Western Slack		
	Length	Width	Height	Capacity, lb. per hr. 9% Moisture	Capacity, lb. per hr. 3% Moisture	Kw. at Max. Capacity	Capacity, lb. per hr. 10% Moisture	Capacity, lb. per hr. 4% Moisture	Kw. at Max. Capacity	Capacity, lb. per hr. 16% Moisture	Capacity, lb. per hr. 10% Moisture	Kw. at Max. Capacity
30	65 1/4	38	55 1/8	1500	1800	15.7	1200	1500	15.8	1100	1400	16.0
40	72 3/8	44	57 1/8	2000	2600	21.5	1600	2000	21.8	1600	1900	22.0
47	84 1/4	42	77	3000	3500	29.5	2400	2900	29.5	2400	2700	30.0
50	93 9/16	51	57 3/4	4500	5300	43.0	3600	4300	44.0	3600	4000	44.7
53	101 3/4	47 3/4	126	4500	5500	44.5	3600	4500	45.5	3600	4200	45.7
60	126 1/2	47 1/4	137 5/16	6000	7500	60.9	5000	6200	61.8	5000	6000	62.2
70	142	47 3/4	153 5/16	6500	8000	60.4	5500	6500	61.6	5200	6200	62.0
72	143 1/2	54 3/4	144 5/16	7500	9400	68.3	6400	7500	69.7	6000	6800	70.3
75	147	53 13/16	159 1/4	9000	11200	81.6	7600	9000	82.3	7200	8000	82.6
82	150 1/2	51 1/2	154 5/16	10000	12500	91.0	8500	10000	92.8	8000	9000	93.4
88	159	51	141	13300	16700	122.3	11300	13300	123.4	10700	12000	124.8

Table 24.—Dimensions and Capacities of Raymond Roller Mills

(Combustion Engineering Corp., New York, 1935)

Capacities are based on fineness of 65% through 200-mesh screen. Power requirements include power for both mill and exhauster.

Size	Overall Dimensions, in.			Eastern Low Volatile			Eastern High Volatile			Western Slack		
	Length	Width	Height	Capacity, lb. per hr. 9% Moisture	Capacity, lb. per hr. 3% Moisture	Kw. at Max. Capacity	Capacity, lb. per hr. 10% Moisture	Capacity, lb. per hr. 4% Moisture	Kw. at Max. Capacity	Capacity, lb. per hr. 16% Moisture	Capacity, lb. per hr. 10% Moisture	Kw. at Max. Capacity
Midget	118	68	102	4600	5500	30.0	2800	3500	30.2	2800	3400	30.2
2–2	158	90	126 3/4	6500	7500	40.0	4100	5000	40.0	4100	5000	40.0
2–3	......	......	....	9000	10500	55.2	5700	7000	55.0	5700	7000	55.0
2–4	......	......	....	11500	13500	70.2	7200	9000	70.0	7200	9000	70.0
5	173	105	124	15000	18000	90.0	9500	12000	90.2	9500	12000	90.2
10	206	138	134	23000	30000	139.3	16000	20000	140.0	17500	20000	140.0
15	208	144	131 1/4	36000	45000	203.0	26000	33000	207.0	26000	32000	207.0
25	236	138	201	52000	68000	299.0	39000	50000	307.8	39000	50000	307.8

Attrition Mills pulverize the coal by balls or rollers. Fig. 4 shows a Babcock & Wilcox pulverizer. Two rows of balls *A*, 9 in. diam., are interposed between the stationary rings *B* and *C*, and the rotating ring *D*. Ring *D* floats on and is driven by the main shaft *E*, and in turn propels the balls *A*. Grinding pressure is applied by springs *F* which bear on ring *B*. Coal is fed through the inlet *G* and passes successively through the two rows of balls. Air blown through *H* carries the pulverized coal through the cyclone *J*, whence it passes through the outlet *K* to storage bin or burner.

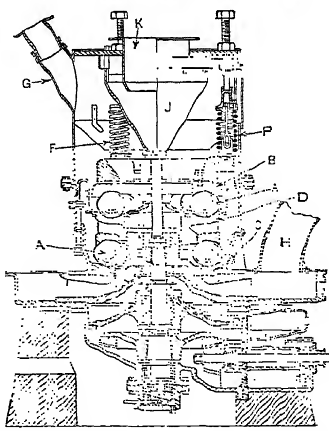


FIG. 4. Babcock and Wilcox Pulverizer

The Raymond roller mill, Fig. 5 and Table 24, comprises a series of vertical rollers *A*, carried on a spider *B*, which is rotated by the shaft *C*. The roller supports are pivoted on the spider at *D*, so that under the influence of centrifugal force they swing out against the bull-ring *E*. Coal enters from feeder *F* and is pulverized between rollers and bull-ring, falling to the bottom of the mill where it is plowed into the currents of entering air by plows *G*. Air blown in from windbox *H* enters the mill through tangential ports in the base casting *J*. The air carries the pulverized coal up to the outlet, first passing it through the separator *K* which rejects

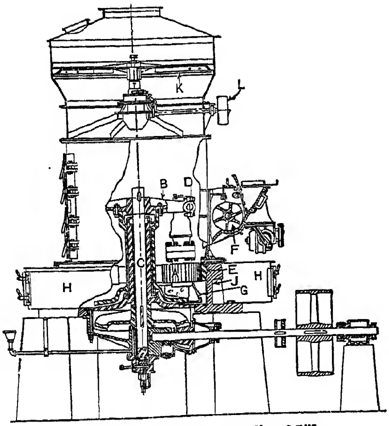


FIG. 5. Raymond Roller Mill

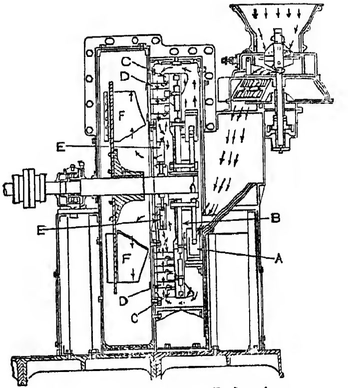


FIG. 6. Riley Atrita Pulverizer

coarse particles and returns them to the rollers. The degree of fineness is governed by the speed of the deflector, which is driven from L through a variable speed transmission.

Combination Impact and Attrition Mill.—The Riley Atrita pulverizer, Fig. 6 and Table 25, is a combination impact and attrition machine. Raw coal from the feeder flows into the first compartment where swing hammers A, carried on the rotor B, reduce it to a granular state by impact. The partially pulverized coal passes around the rotor and is ground by attrition between the pegs C on the rotor and stationary pegs D on the housing. The fuel is drawn through rejector arms E, by the fan F. The heavier particles are thrown out of the rejector by centrifugal force and returned for further attrition.

THE GRINDABILITY OF COAL is an important factor in selecting coal for pulverized fuel installations. Relative grindability may be expressed as the ratio of the total increase of surface produced by grinding to the surface of a standard sample subjected to similar treatment. R. M. Hardgrove (*Trans.* A.S.M.E., FSP-54-5, 1932) describes methods of determining grindability. The original sample consists of material that passes a 16-mesh screen and is retained on a 30-mesh screen. After grinding, the sample is passed through the screens of Table 26. The surface units of the ground sample are calculated by multiplying the percentage remaining on each screen by the surface factor, *i.e.*, the reciprocal of the average opening as given in Table 26. An arbitrary factor of 1000 is used for all material passing 300 mesh. A typical calculation of the grindability of Illinois coal, ground in a ball mill is as follows:

Screen Mesh	Percentage between Screens		Factor		Surface Units
16– 30	24.4	×	29		708
30– 60	36.0	×	61		2196
60–100	13.8	×	129		1780
100–140	7.2	×	202		1454
140–200	3.8	×	285		1084
200–230	1.6	×	377		602
230–300	2.4	×	465		1116
Through 300	10.8	×	1000		10800
			Total surface units	=	19740
			Surface units of original sample	=	2900
			New surface units	=	16840

New surface units of standard sample = 29,961; grindability relative to standard sample = 16,840/29,961 = 56.2%.

This method is based on Rittinger's law that the work done in grinding is proportional to the new surface produced. The grindability of a coal appears to have a relation to its volatile content. Coals of from 16 to 28% volatile have high grindability, this decreasing with increase of volatile. The volatile content cannot, however, be taken as an

Table 25.—Dimensions and Capacities of Riley Atrita Pulverizers
(Riley Stoker Co., Worcester, Mass., 1935)

Mach. No.	Overall Dimensions, in. Length	Width	Height	Capacity, lb. per hr. Kw. input at max. capacity. Percent through 200-mesh	Eastern Low Volatile, Moisture in Coal, percent 4	Eastern Low Volatile 10	Eastern High Volatile 4	Eastern High Volatile 10	Western Slack 10	Western Slack 16
1	33	27 7/16	29	Capacity	1000	530	800	520	......	......
				Kw.	7.2	7.2	7.6	7.6	......	......
				Percent	76	80	65	71	......	......
2	48 5/8	45 1/2	38	Capacity	2000	1060	1400	900	......	......
				Kw.	11.5	11.5	12.7	12.7	......	......
				Percent	75	80	65	72	......	......
3 31 1/2″ fan	48 1/2	63 1/2	57	Capacity	4000	2100	3000	1950	3200	1900
				Kw.	24	24	28	28	24.2	24.2
				Percent	70	80	65	76	60	72
3 27 1/4″ fan	48 1/2	63 1/2	57	Capacity	3000	1750	2500	1650	2600	1520
				Kw.	18.2	18.2	21.7	21.7	19.2	19.2
				Percent	77	80	72	77	68	73
4 48″ fan	75 3/16	87 15/16	87	Capacity	9000	4750	6500	4250	7100	4300
				Kw.	45	45	50	50	47	47
				Percent	70	80	60	74	59	69
4 40″ fan	75 3/16	87 15/16	87	Capacity	6300	3350	5000	3250	5000	3000
				Kw.	31	31	36	36	32.5	32.5
				Percent	78	81	71	76	73	75
4 Duplex 48″ fan	114	92	87 3/4	Capacity	18,000	9050	12,900	8500	14,300	8600
				Kw.	90	90	102	102	92	92
				Percent	70	80	61	74	61	74
4 Duplex 40″ fan	114	92	87 3/4	Capacity	12,600	6700	10,000	6500	10,000	6000
				Kw.	63	63	74	74	65	65
				Percent	79	81	71	76	73	80
5 66″ fan	86	108	107	Capacity	15,000	8000	10,300	6700	11,600	7000
				Kw.	83	83	89	89	81	81
				Percent	70	80	61	74	61	70
5 54″ fan	86	108	107	Capacity	10,500	5700	8000	5200	8200	5200
				Kw.	57	57	65	65	56	56
				Percent	79	81	71	76	68	73
5 Duplex 66″ fan	132 1/4	111	106	Capacity	30,000	16,000	20,600	13,400	23,200	14,000
				Kw.	167	167	188	188	162	162
				Percent	70	80	61	73	61	70
5 Duplex 54″ fan	132 1/4	111	106	Capacity	21,000	11,400	16,000	10,400	16,400	10,400
				Kw.	114	114	130	130	112	112
				Percent	79	81	70	76	68	73

index of grindability. Furthermore, the grindability depends to some extent on the apparatus used to determine it, and on the operating speed. Hardgrove used a ball mill, a mortar and pestle and a tube mill machine. The first two gave fairly consistent results, while the third was so erratic that its results were not comparable. Consistent results were obtained when the ball mill operated at 60 r.p.m. and the mortar and pestle at 1300 strokes per min. The relation between grindability and pulverizer capacity must be determined for each type of pulverizer and the conditions under which it operates.

Table 26.—Screen Sizes for Determining Grindability

U. S. Screen Series Mesh	Opening, in.	Average Opening: Between Screen Sizes	Average Opening: Opening	Reciprocal of Average Opening
16	0.046	16– 30	0.0346	29
30	.0232	30– 60	.01645	61
60	.0097	60–100	.00775	129
100	.0058	100–140	.00495	202
140	.0041	140–200	.0035	285
200	.0029	200–230	.00265	377
230	.0024	230–300	.00215	465
300	.0019	......	......	525 +

Table 27.—Grindability of Representative Coals

State	District	County	Seam	Volatile Matter, percent	Fixed Carbon, percent	Ash, percent	Average Grindability
Ala.	Cahaba	St. Clair	Harkness	32.55	54.55	12.90	58
Colo.	Denver	Weld	....	48.60	45.80	5.57	48
Ill.	Central Ill.	Christian	No. 6	37.70	48.68	13.80	55
	Danville	Vermilion	No. 6	35.46	50.75	13.70	68
	Springfield	Sangamon	No. 5	37.75	46.69	15.22	61
	Murphysboro	Jackson	No. 2	41.00	51.70	7.30	74
Ind.	Sullivan-Linton	Sullivan	No. 5	39.25	44.00	11.75	58
Ky.	Elkhorn	Letcher	Elkhorn	36.45	55.22	8.32	55
Mich.	Saginaw	Saginaw	Saginaw	37.65	52.70	9.65	50
Ohio	Belmont-Pgh	Belmont	Pgh. No. 8	40.80	50.60	8.60	63
	Cambridge	Guernsey	Upper Freeport	34.55	55.55	7.90	49
Okla.	McAlester	Pittsburgh	Wilburton	30.60	55.30	14.00	57
Penn. Anthracite							
	Wilkes-Barre	Luzerne	....	5.63	88.40	5.95	24
	W. Schuylkill	Schuylkill	....	5.50	79.80	13.20	42
Bituminous							
	Clearfield	Clearfield	Upper Kittaning	21.87	63.87	14.25	88
	Indiana	Indiana	Kittaning	26.80	63.42	9.85	106
	Johnstown	Cambria	Freeport	22.00	71.30	6.65	119
	Somerset *	Somerset	Upper Kittaning	17.45	71.88	9.67	100
	Pittsburgh	Allegheny	Pittsburgh	37.25	55.14	7.61	54
Texas	Eagle Pass	Maverick	Eagle Pass	24.10	51.13	26.71	91
Tenn.	....	Overton	Wiler	33.70	53.40	12.90	64
Utah	....	King	Hiawatha	43.05	49.45	7.50	46
Wash.	....	King	McKay	41.70	48.5	9.82	40
	....	Pierce	No. 2	29.05	55.30	15.70	77
W. Va.	Fairmont	Marion	Pittsburgh	39.82	52.85	7.32	67
	New River	Fayette	Sewell	27.40	68.30	4.30	92
	Kanawha	Kanawha	Coalburg	34.90	54.97	10.15	43
	Pocahontas	Mercer	Pocahontas No. 3	21.25	73.35	5.35	107
Canada							
Alta.	Cadomin	....	Cadomin	24.45	60.60	14.95	83
B. Col.	Coal Creek	....	No. 2	25.75	67.40	6.85	108
N. B.	N. Minto	....	Grand Lake	31.30	47.85	20.85	72
N. S.	Glace Bay	....	Emery	31.10	48.05	20.85	73
Ont.	....	....	Lignite	31.15	11.90	5.18	76
Vancouver	Cumberland	....	Comox	28.66	45.70	25.49	65

* This is a 100-grindability coal used as a basis of comparison.

Table 28.—Grindability of Petroleum Coke

Type	Volatile Matter	Fixed Carbon	Ash	Grindability
Petroleum Pitch	59.10	40.70	0.20	175
Cracking Still Coke	12.20	87.75	0.05	123
Pressure Still Coke	15.35	84.05	0.60	120
Coke Still Coke	11.95	87.60	0.45	103
Coke Breeze	10.05	89.39	0.60	102
Hard Lump Coke	9.15	90.65	0.20	99
Coke Still Coke Breeze	10.70	87.85	1.45	73
Coke Still Coke	8.30	91.05	0.65	63
Stack Petroleum Coke	12.30	86.85	0.85	57
Cracking Coil Coke	6.80	92.90	0.60	37
Caustic Tar Butts	13.50	80.10	6.40	25

Table 27 condensed from Hardgrove shows the grindability of a variety of coals. Table 28 shows the grindability of various cokes. See also paper by H. J. Sloman and A. C. Barnhart, *Trans.* A.S.M.E., FSP-56-13, Oct., 1934.

EXPLOSIONS AND FIRE HAZARDS OF PULVERIZED COAL.—A mixture of 30% pulverized coal dust and 70% finely powdered shale is explosive (Bull. 242, U. S. Bureau of Mines, 1925). Hence, coal dust which may have settled on floors and girders of industrial plant buildings is not made inert by the incombustible material mixed with it. Dust collected from the interior of buildings housing furnaces using pulverized coal often contains as high as 23.5% volatile matter. Coal dust containing 10 to 15% volatile matter is explosive.

Spontaneous combustion probably causes most of the fires in storage bins. Experiments by Porter and Ralston (U. S. Bureau of Mines) showed that oxidation of coal increased rapidly at about 150° F. If coal is stored at a fairly high temperature in the presence of sufficient air, the oxidation will raise the temperature to the ignition point. Overheating of coal is especially liable to occur in direct heat driers. The gases of combustion should not be allowed to become so hot as to overheat the coal. Storage bins should be so located that they will not become heated by steam pipes or hot flues.

After a shut-down, delivery of coal from the storage bins should not be allowed until it is certain that the coal has not been heated to a point where it will ignite when brought in contact with air. Fires occur in distribution lines due to back-fires from the heated furnace, caused by air pressure in the distributing line dropping below the secondary air pressure.

When the plant shuts down all furnaces should be cut from the supply line, the gate from the pulverized fuel bin closed, and the transport line blown out free from pulverized coal. In examining bins and other storage places where coal dust may exist open lights should not be used. Electric bulbs should be protected from breakage, as coal dust may be ignited by the hot filament of the lamp.

The maximum quantity of pulverized coal that can be stored, according to the rules of the National Fire Protection Association, is as follows:

Temperature of coal entering mills, deg. F........	250	225	200	175	150
Maximum stored quantity, hours supply for plant.	4	8	12	16	18

FLY ASH FROM PULVERIZED FUEL.—Removal of fly ash from the gases of combustion of boiler plants using pulverized fuel often is necessary to avoid creating a nuisance. Several methods are used: 1. Separators of the cyclonic type, which eliminate the dust by centrifugal action. 2. Separators with baffles that cause abrupt changes of direction of the gases and trap the dust in the baffles. 3. Separators using fans, that throw the dust, by centrifugal action to the periphery of the scroll. 4. Electric precipitation of the dust. 5. Removal of dust by water sprays.

The efficiency of the several methods depends on local conditions. Some factors affecting efficiency are the volume, temperature, relative humidity, chemical analysis and velocity of the gases, the quantity of dust per unit of gas volume, the relative proportions of coarse and fine dust. Comparisons of relative efficiency of different methods of separation, therefore, is useless unless all conditions are known.

Fineness of dust has a marked effect on the dust recovery effected in the separator. The efficiency of a cyclone type separator has ranged from 60% where over 50% of the dust passed a 300-mesh sieve, to 99% where over 90% of the dust passed through 300-mesh. For equal efficiency, the velocity of gas required for the finer dust is about double that required for the coarser. Other types of separators show about the same range of efficiency with variation in dust content, fineness, etc. The manufacturers of equipment should be consulted in regard to selection of equipment to meet any given set of conditions. See paper by K. Toensfeldt (*Trans.* A.S.M.E., FSP–50–59, 1928), for a description of the various systems and results of tests.

The recovery of fly ash from a 12 ft. diam. stack by means of water sprays is described by J. W. Mackenzie (*Trans.* A.S.M.E., FSP-53-5, 1931). The nozzle was located about 20 ft. above the breeching, and required about 30 gal. of water per min. at 10 lb. pressure. A second nozzle 15 ft. above the first, directed its spray upward. The percentage of fly ash to coal fired ranged from 5.63 to 11.16%, and the ash recovered ranged from 46 to 31%. An acid-resisting lining for the stack was necessary.

3. LOW TEMPERATURE CARBONIZATION OF COAL

Low temperature carbonization comprises essentially the destructive distillation of coal at temperatures of not over 1500° F. Its object is to provide a cheap fuel by removing from coal its more valuable constituents as tar, light oils and rich gas, and also to enable bituminous coal to be used as a substitute for anthracite as a domestic fuel. The products of low-temperature distillation are a semi-coke, containing sometimes as high as 15% volatile matter, tar, light oils and gases high in hydrocarbons. The gases most profitably may be used as enrichers for producer water gas. Low temperature carbonization processes have been developed to a considerable extent in Europe, but plants in the U. S. have until now (1935) been largely experimental. The coal used generally is a coking coal, since non-coking coals do not yield the same value in by-products and require special prior processing to produce lumpy coke. A number of low-temperature carbonization processes are described in the Proceedings of the International Conference on Bituminous Coal, Pittsburgh, 1928.

THE CARBOCITE PROCESS treats the coal continuously in slightly inclined rotary furnaces. The coal is crushed, dried and heated to 450° F. in a primary drum, and then distilled at 900–950° F. in a secondary drum or carbonizer. Each drum comprises concentric shells with an annular space 2 in. wide between them, through which flow the heating gases. At a test plant at Philo, Ohio, the coke from West Virginia coal contained about 12% volatile, was dense and strong, and ranged in size from 1/4 to 4 in. diameter. The gas had a heating value of about 800 B.t.u. per cu. ft.

THE TRUMBULL PROCESS comprises a series of retorts heated by superheated steam. The coal is preheated and dried by waste gases from the boiler and superheated, and then is fed intermittently to the retorts, set in batteries of two. The superheated steam passes successively through the several batteries of retorts, being resuperheated between each battery, the temperature being somewhat lower in successive batteries.

THE PLASSMANN PROCESS developed to carbonize low-temperature slightly coking coals, utilizes a vertical retort. The coal is subjected to pressure which is varied according to the compactness of coke desired.

THE TURNER PROCESS uses a vertical retort, with highly superheated exhaust steam as the heating medium. Steam is admitted at the bottom of the retort, which is charged and discharged continuously through valves. Gas is discharged through valves that operate between two fixed pressures, and is condensed in a tubular condenser. The experimental fuels comprised peats, lignites, oil shales, cannel and bituminous coal. The latter presented some difficulties, some requiring treatment for 12 hr. The yield of coke is about 70% by weight. The following yields of oil have been obtained per ton: From peat, up to 20 gal.; from lignite, up to 61 gal.; from oil shale, up to 80 gal.; from bituminous coal, up to 45 gal. One ton of steam at 1200° F. was required to treat 1 ton of bituminous coal.

THE WEISS SYSTEM for low temperature distillation of briquets uses superheated steam at about 1200° F. Coal dust is formed into briquets, with pitch or asphalt binder. The retorts, of 20 tons capacity, are heated progressively with the superheated steam, which then is condensed to recover the primary oils, tars and gases.

THE K. S. G. PROCESS, for the production of domestic fuel (*Trans.* A.S.M.E., FSP–50–28, 1928), utilizes two inclined concentric retorts which rotate as one. Coal is dried at about 400° F. as it passes through the inner retort. It then is fed into one end of the annular space between the two retorts and is discharged as coke at the opposite end. The gases of combustion from the furnace surround the outer retort. Superheated steam is admitted into the coal charge in the zone of highest temperature to prevent carbon deposits and to depress the vapor pressure of the tar as it is formed, thereby reducing thermal decomposition of the tar. By mixing gases of combustion with flue gas at 600° F., recirculated from the base of the stack, four temperature zones surrounding the outer retort are formed, with temperatures of 1320°, 1300°, 1000° and 1000° F. respectively.

Coal containing from 3 to 4% moisture, and of proximate analysis: volatile matter, 23 to 25%; fixed carbon, 59 to 63%; ash, 13 to 17%; calorific value, 12,000 B.t.u. per lb., yielded, per ton (2000 lb.) by weight, coke (12% volatile), 84%; gas, 2000 cu. ft., 5.5%; tar, 14 gal., 6.2%; light oil, 2.5 gal., 0.8%; moisture, 3.5%. The gross heat necessary to carbonize 1 ton of coal was 1,630,000 B.t.u. and 35 I.Hp. was required to drive the apparatus. The coke recovered easily ignites, is free burning and smokeless. A typical screen analysis is: 1 1/2 in. and over 32%; 3/4 to 1 1/2 in., 36%; 1/2 to 3/4 in., 11.8%; 1/4 to 1/2 in., 13.9%; 0 to 1/4 in., 6.3%. The tar recovered has a specific gravity of 1.05 to 1.06 at 60° F. The specific gravity of the light oil recovered is 0.775 at 60° F. Table 29 gives typical distillation analyses of tar and oils. Gas recovered from the same coal from which the foregoing tars and gas were formed analyzed: CO_2, 4.5%; C_nH_m, 5.2%; O_2, 1.6%; CO, 5.9%; H_2, 15%; C_2H_6, 6.3%; CH_4, 46.7%; N_2, 14.8%. Its specific gravity was 0.696 at 60° F.; gross heating value, 814 B.t.u. per cu. ft.; yield, 3200 cu. ft. per ton of 2000 lb.

Table 29.—Distillation Analyses of Tar and Light Oils from Low-temperature Coal Carbonization

Tar			Light Oil	
Fraction, deg. C.	Percent by Volume	Specific Gravity at 15° C.	Fraction, deg. C.	Percent
93–180	7.5	0.856	40–100	43.0
180–230	19.2	.942	100–120	15.0
230–270	13.6	.994	120–150	16.0
270–360	22.9	1.056	150–180	8.0
Pitch	36.8	1.170	180–200	7.0
			over 200	11.0

ECONOMICS OF DISTILLATION OF POWER PLANT COAL.—In a circular, Distillation Products of Coal (Natl. Elec. Light Assoc., now Edison Institute, 1929), the Babcock and Wilcox Co. contributes an analysis of the cost of developing power in

the ordinary steam power plant, and one using coke produced by low temperature distillation, credit being given in the latter case to the value of the by-products from carbonization. The conditions assumed were a plant of 25,000–75,000 kw. capacity, operating at an annual average load of 40% of capacity, using coal of 12,722 B.t.u. per lb. The total power cost was 1.16 cents per kw-hr. The analysis shows that if the carbonization process can be operated at a cost of less than 3 cents per 1,000,000 B.t.u. in coke delivered as a boiler fuel, all fixed charges included, the process will be profitable; otherwise it will be unprofitable when operated as an auxiliary. The statement is made that a coal process will pay, under the assumed conditions, when the sum of its operating and investment increments of cost is equal to or less than 22% of that of the boiler plant in dollars per year. This percentage will vary with conditions other than those assumed in the analysis.

LOW TEMPERATURE CARBONIZATION OF SOUTHERN APPALACHIAN COAL.—Leo Holdridge (*Trans.* A.S.M.E., FSP–51–43, 1929) describes laboratory tests on a coal of analyses: Ash, 6.56%; moisture, 1.68%; volatile, 35.71%; fixed carbon, 56.65%; heating value, 13,916 B.t.u. per lb. The coal was carbonized at a maximum temperature of 1000° F. The yield per ton of coal was: Semi-coke, 1393 lb.; gas, 3613 cu. ft.; dehydrated tar, 37 gal. The tar products per ton of coal were: Motor fuel, 18,480 B.t.u. per lb., 4.4 gal.; burning oil, 18,172 B.t.u. per lb., 4.6 gal.; light lubricating oil, 4.4 gal.; heavy lubricating oil, 2.8 gal.; grease, 12 lb.; pitch, 90 lb.; tar acids, 66 lb. The properties of the dehydrated tar are given in Table 30.

Table 30.—Properties of Dehydrated Tar

	Dehydrated Tar	Dehydrated Tar, Less Oils to 212° F.	Dehydrated Tar, Less Oils to 447° F.	Pitch at 554° F.	Pitch at 617° F.
Viscosity at 100° F.	525				
" " 210° F.	40	49	1570		
Specific gravity	1.04	1.06	1.089	1.120	
Flash point, deg. F.	140	255	312		
Burning point, deg. F.	175	280	350		
Heating value, B.t.u. per lb.	16,600	16,500	16,400		
Melting point, deg. F.					450
Twist point, deg. F				75	

4. COKE

Coke is formed by distilling the volatile matter from coal. Gas coke is formed in a gas-works retort. Oven coke is made in a coke oven, either of the beehive or by-product type. But little beehive coke is now (1935) made. Gas coke is unsuitable for metallurgical purposes, but is used as a domestic fuel. Table 31 from Bulletin 6, U. S. Bureau of Mines, gives analyses and heating values of cokes and the coal from which they were made. Oven coke is used principally as a metallurgical fuel. It has a hard, close texture, is porous and brittle. It retains moisture in quantities up to 20%.

Table 31.—Analyses and Heating Values of Gas Coke

Kind of Coal	Coal: Mois-	Coal: Ash	Coal: Volatile	Coal: Fixed Carbon	Coal: Sulphur	Coal: B.t.u. per lb.	Condi-	Coke: Moisture	Coke: Ash	Coke: Volatile	Coke: Fixed Carbon	Coke: Sulphur	Coke: B.t.u. per lb.
Pittsburgh:													
As received	1.92	6.41	32.82	58.85	1.12	14,026	A	8.54	11.46	0.97	79.03	0.84	11,552
Dry....		6.54	33.46	60.00	1.14	14,301	B		12.53	1.06	86.41	0.92	12,631
Alabama:													
As received	2.71	4.29	29.13	63.87	0.50	13,990	A						
Dry......		4.41	29.94	65.65	0.51	14,380	B		11.40	1.59	81.01	0.52	12,883
Colorado:													
As received	7.17		32.36	45.92	1.00	10,953		21.31	19.93	1.40	57.28	0.68	8,417
Dry......			34.86	49.47	1.08	11,799		.....	25.35	1.78	72.87	0.87	10,706
Kentucky:													
As received	2.46	.25	31.18	60.11	0.43	13,885	A	12.43	10.09	0.92	76.56	0.36	11,210
Dry....		.41	31.97	61.62	0.44	14,234	B		11.52	1.05	87.4[illegible]	0.41	12,802

* Condition—A = 3 days after quench; B = from retorts.

THE HEATING VALUE OF COKE may be calculated roughly from the formula $H = \{14{,}600\,(100 - A)\} \div 100$, where H = heating value, B.t.u. per lb.; A = percentage

of ash. The yield of coke varies widely with different coals, ranging from 90 to 35% of the weight of the coal. Tables 32 and 33, Bulletin 50, U. S. Bureau of Mines, gives the range of analyses of beehive and by-product coke from several localities. Table 34 (Tech. Paper 367, U. S. Bureau of Mines) gives the analyses and heating values of three samples of Clairton (Penn.) by-product coke, weighing 33 lb. per cu. ft.

Table 32.—Range of Analyses of Beehive Coke

Location of Oven	Moisture	Volatile Matter	Fixed Carbon	Ash	Sulphur
Pennsylvania..................	0.23–0.91	0.29–2.26	92.53–80.84	6.95–15.99	0.81–1.87
West Virginia..................	0.07–0.60	0.46–2.35	95.47–84.09	4.00–12.96	0.53–2.26
Southern Appalachian Field....	0.75–1.34	0.75–1.95	91.20–77.81	7.30–18.90	0.58–1.77

Table 33.—Range of Analyses of By-product Coke

Location of Oven	Moisture	Volatile Matter	Fixed Carbon	Ash	Sulphur	Phosphorus
Alabama........	1.92–10.64	0.95–1.58	89.02–84.59	9.40–14.16	0.71–1.08	0.031–0.071
Illinois, Indiana...	0.73– 7.00	0.80–1.45	89.84–87.00	8.63–11.15	0.60–0.83	0.003–0.015
Md., Mass., N. J..	8.00–11.00	1.61–3.30	86.99–85.73	9.75–12.12	1.18–2.37	0.004–0.012
Mich., Minn., Wis.	1.80– 2.64	0.87–2.70	91.93–87.98	7.20– 9.70	0.68–0.97	0.012–0.010
New York........	0.58– 5.50	1.48–2.26	89.49–79.95	8.48–16.26	0.69–1.58	0.010–0.058
Ohio, W. Va......	1.35– 3.00	0.90–1.69	88.10–87.35	8.75–10.92	0.79–0.95	0.012–0.035
Pennsylvania.....	0.20–11.00	0.60–1.95	89.68–84.65	9.52–13.40	0.93–1.40	0.011–0.022

Table 34.—Analyses of Clairton By-product Coke

Sample No.	Moisture	Volatile Matter	Fixed Carbon	Ash	B.t.u. per lb.	
					As Fired	Ash and Moisture Free
1	3.6	2.1	80.0	14.3	11,770	14,330
2	2.3	2.3	81.1	14.3	11,890	14,260
3	0.8	1.1	80.5	17.6	11,720	14,370

BY-PRODUCTS RECOVERED IN COKE MANUFACTURE.—A typical yield of a by-product coke oven per ton of coal charged is: Coke, 73–74%; tar, 4–6%; ammonium sulphate, 1–1.25%; gas, 10,000 cu. ft. The coal tar is split into light and heavy oils and pitch. The tars are the bases of many dyes, medicinal preparations and other industrial products. See also Low Temperature Carbonization of Coal, p. 4–37.

COKE BREEZE is fine coke, 1/2 in. and smaller. Tech. Paper 303, U. S. Bureau of Mines, gives the typical analysis of a by-product coke breeze shown in Table 35. A stack draft of 0.5 in. of water was required to burn a mixture of this breeze, both coarse and fine, with bituminous coal, when burning 8 to 11 lb. per sq. ft. of grate per hour.

Table 35.—Analyses of Coarse and Fine Coke Breeze

	Coarse		Fine				Screen Analysis		
							Percent through Screen		
	As Fired	Moisture Free	As Fired	Moisture Free	As Fired	Moisture Free	Opening	Coarse	Fine
Moisture.......	7.39	....	6.55	....	4.01	....	3/4 in.	78	100
Volatile Matter.	2.49	2.69	3.40	3.64	3.04	3.7	1/2 "	67	100
Fixed Carbon...	66.35	71.64	65.01	69.56	66.25	69.01	1/4 "	50	78
Ash...........	23.77	25.67	25.04	26.80	26.70	27.82	1/10 "	33	60
B.t.u. per lb....	9,748	10,526	9,559	10,229	9,995	10,413	1/30 "	11	9

COKE AS A DOMESTIC HEATING FUEL is discussed in U. S. Bureau of Mines Report of Investigation 2980. Coke of 1 1/2-in. mesh screen size appears to be the most satisfactory, with draft above the fire ranging from 0.02 in. of water when burning 1 lb. per sq. ft. of grate per hr., to 0.09 in. when burning 8 1/2 lb. per sq. ft. per hr.

PETROLEUM COKE, used for carbon electrodes and dry cells, is the residue remaining after refining crude petroleum. It has a range of analysis of about: Fixed carbon, 90–95%; volatile matter, 5–10%; ash, up to 1.5%; sulphur, 0.1 to 0.5%.

LOW TEMPERATURE COKE is produced by carbonizing coal at temperatures below 1400° F. See Low Temperature Carbonization of Coal, p. 4–37.

DRY-QUENCHED COKE, that is, glowing coke quenched without the use of water, by confining it in an atmosphere of inert gases is claimed to have the following advantages over wet-quenched coke: Freedom from moisture, which gives a higher net heating value; more uniform size; lower quantity of fines and breeze; stronger, thus avoiding breakage

under loads in the blast furnace with a consequent more even distribution of the blast; reduction in labor cost of manufacture.

The Sulzer system of dry quenching (W. O. Renkin, *Trans.* A.S.M.E., FSP–53–7, 1931) uses the heat in the coke to generate steam. The glowing coke at approximately 1800° F. is charged into a sealed cooling chamber. A fan draws the inert gas through the coke, where it absorbs heat, and then through a boiler, and discharges it into the cooling chamber to repeat the cycle. The air entrapped in the coke burns a small portion of it, the loss amounting to about 0.04%. When the coke has cooled to about 700° F. it is discharged from the cooling chamber. Fresh coke is charged as each lot of quenched coke is withdrawn and the process is continuous.

The heat available for steam generation is given by the formula $H = Tc_1 - tc_2$, where H = heat given up by coke in cooling B.t.u. per lb.; T and t = respectively, charging and withdrawal temperatures of the coke, deg. F.; c_1 and c_2 = mean specific heats of coke at temperatures T and t, respectively. Values of c are given by Terres and Schaller as follows:

Temp., deg. F.	70–482	70–492	70–943	70–1128	70–1306	70–1492	70–1674	70–1848	70–2025
c	0.230	0.267	0.292	0.315	0.331	0.345	0.354	0.361	0.369

Thus the heat available per pound of coke cooled from 1800 to 700° F. will be

$$H = (1800 \times 0.361) - (700 \times 0.292) = 445.4 \text{ B.t.u.}$$

Table 36 gives a comparison of the analyses of wet- and dry-quenched coke.

Table 36.—Analyses of Wet- and Dry-quenched Coke

Analysis	Original Coal	Wet-quenched Coke		Dry-quenched Coke	
		Moisture Free	As Received	Moisture Free	As Received
B.t.u. per lb.	14,300	13,039	11,463	13,088	13,023
Volatile matter, percent	36.47	1.71	1.49	1.16	1.15
Fixed carbon, percent	58.97	91.29	79.51	91.48	91.03
Ash, percent (calculated on ash and moisture free basis)	4.56	7.00	6.10	7.36	7.32
Moisture, percent	....	....	12.90	....	0.50
Sulphur, percent (separately determined)	0.75	0.75	0.75	0.68	....

5. WOOD

HEATING VALUE OF WOOD.—The specific gravity of wood ranges from 0.3 to 1.2; the heating value of dry wood is approximately proportional to the specific gravity, excepting that the presence of resin increases heating value. Newly-felled wood contains moisture, the amount depending on the species, and averaging 40%. Ordinary air-dry wood contains about 25% moisture. Table 37 shows the amount of moisture expelled from wood at gradually increasing temperatures. Table 38 gives the heating values, and Table 39 the analyses of the more common varieties of woods. The heating value as computed by Dulong's formula (see p. 4-03) usually gives lower values than calorimetric determinations. Taking average composition (Table 39), heating value of the dry wood is

$$(14{,}600 \times 0.4956) + 62{,}000\,(0.0611 - 0.4383/8) = 7645 \text{ B.t.u. per lb.}$$

If the wood contains 25% moisture, the net heating value is (0.75 × 7645) minus the heat required to convert the moisture at atmospheric temperature, say 62° F., to steam at atmospheric pressure and to superheat the steam to the temperature of the chimney gases, say 450°. Net heating value based on Dulong's formula then is

$$(0.75 \times 7645) - \{0.25 \times (212-62) + (0.25 \times 970.2) + 0.25(1204.1-970.2)\} = 5398 \text{ B.t.u.}$$

The average heating value as determined by calorimeter is 8671 B.t.u. per lb.

Table 37.—Proportion of Water Expelled from Wood at Various Temperatures

Temperature	Water Expelled from 100 Parts of Wood			
	Oak	Ash	Elm	Walnut
257° F.	15.26	14.78	15.32	15.55
302° F.	17.93	16.19	17.02	17.43
347° F.	32.13	21.22	36.94 ?	21.00
392° F.	35.80	27.51	33.38	41.77 ?
437° F.	44.31	33.38	40.56	36.56

Table 38.—Analyses of Dry Woods

(Gottlieb)

	Oak	Ash	Elm	Beech	Birch	Fir	Pine	Average
Carbon	50.16	49.18	48.99	49.06	48.88	50.36	50.31	49.56
Hydrogen	6.02	6.27	6.20	6.11	6.06	5.92	6.20	6.11
Nitrogen	0.09	0.07	0.06	0.09	0.10	0.05	0.04	0.07
Oxygen	43.36	43.91	44.25	44.17	44.67	43.39	43.08	43.83
Ash	0.37	0.57	0.50	0.57	0.29	0.28	0.37	0.42

Table 39.—Heating Value of Woods

(Based on U. S. Dept. of Agriculture Bull. No. 753)

	Weight per cord, lb.		Heating Value, B.t.u. per lb.		Equivalent lb. of Coal of 13,500 B.t.u. per lb.	
	Green	Air-dry	Green	Air-dry	Green	Air-dry
Ash, white	4300	3800	4628	5395	0.343	0.400
Beech.......	5000	3900	3940	5359	.292	.397
Birch, yellow.	5100	4000	3804	.5225	.282	.387
Chestnut....	4900	2700	2633	5778	.195	.428
Cottonwood.	4200	2500	3024	6000	.224	.444
Elm, white..	4400	3100	3591	5710	.266	.423
Hickory.....	5700	4600	4053	5391	.300	.399
Maple, sugar	5000	3900	4080	5590	.302	.414
Maple, red..	4700	3200	3745	5969	.277	.442
Oak, red....	5800	3900	3379	5564	.250	.412
Oak, white..	5600	4300	3972	5558	.294	.412
Pine, yellow.	3100	2300	7097	9174	.526	.680
Pine, white.	3300	2200	4226	5864	.313	.434
Walnut, black	5100	4000	4078	4650	.302	.344
Willow................	4600	2300	2370	5870	.176	.435

WOOD REFUSE consisting of sawdust, shavings, trimmings, etc., can be efficiently utilized as boiler fuel, providing that the furnace is properly designed. The requirements are ample combustion space, ranging from 5 to 8 cu. ft. per rated boiler horsepower, ample draft, with preferably automatic control, variation in rate of feeding refuse in accordance with the load, and in rate of air supply. Proper instruments, as draft gages, steam flow meter, thermometers and CO_2 indicator assist the firemen to maintain proper operating conditions. Wood refuse commonly is burned at low efficiency, and the balance of the steam requirements are made up by firing coal. Since 2 lb. of wood is roughly equivalent to 1 lb. of coal, any improvement in efficiency of wood burning decreases the cost of coal. E. Winholt (*Trans.* A.S.M.E.,WDI–50–16 A, 1928) cites a case of burning coal and shavings where less than 2 lb. of steam were produced per 1 lb. of shavings. When the shavings were burned alone in a properly designed furnace, approximately 5 lb. of steam were produced per 1 lb. of shavings. For a complete discussion of wood refuse furnaces, see papers by H. W. Beecher, A. C. Sullivan, and C. L. Young, *Mech. Engg.*, July, 1925; E. Winholt, B. A. Parks, *Trans.* A.S.M.E.,WDI–50–16 A and B, 1928; C. S. Gladden, *Trans.* A.S.M.E., FSP–53–3, 1931.

The use of preheated air with wood fuel increases both boiler capacity and efficiency. H. S. Colby (*Trans.* A.S.M.E., FSP–53–3, 1931) gives results of boiler tests in which the use of air preheated to 456° F. increased the capacity of the boiler from 5.66 lb. of water evaporated per sq. ft. of heating surface per hr. to 6.53 lb., and increased the boiler efficiency from 60.4% to 73%.

Wood refuse comprising about 50% each of chip and sawdust, with about 35% moisture, have been used as gas producer fuel (C. E. Snypp, Louisiana Engg. Soc., May 13, 1912). Cypress refuse, with a heating value of 5540 B.t.u. per lb., gave gas of heating value of 130–135 B.t.u. per cu. ft. Pine refuse, of 7605 B.t.u. per lb., gave gas of heating value of 161 B.t.u. per cu. ft.

A Cord of Wood is a pile 4 × 4 × 8 ft. = 128 cu. ft., comprising about 56% solid wood and 44% interstitial spaces. B. E. Fernow (J.C.I.W., vol. iii) gives the percentage of solid wood in various cords as follows: Timber cords, 74.07%; firewood cords (over 6 in. diam.), 69.44%; billet cords (over 3 in. diam.), 55.55%; brush wood cords (less than 3 in. diam.), 18.52%; roots, 37%.

6. CHARCOAL

THE COMPOSITION OF CHARCOAL depends on the maximum temperature during its production and the duration of that temperature. Black alder charcoal showed the following analyses (M. Violette):

	C	H	O	and loss	Ash
Carbonized at 592° F..........	73.24	4.25	21.96	0.55	24.61
" " 1873° F..........	81.97	2.30	14.15	1.58	15.30

Charcoal also is a by-product of the destructive distillation of wood in retorts at temperatures of from 390 to 625° F. Complete carbonization is not obtained. The yield of 1 cord of wood weighing 4000 lb. by this process (J. S. S. Brame, Fuel, p. 20) is: Charcoal 950 lb.; turpentine, 40 gal.; light oils, 16 gal.; heavy oils, 128 gal., together with large

volumes of gases. A typical analysis of these is: Heavy hydrocarbons, 8.16%; CH_4, 12.32%; CO_2, 31.45%; CO, 35.08%; H, 10.94%; N, 2.05%.

THE HEATING VALUE OF CHARCOAL depends on the percentage of carbon and ash, and ranges from 11,000 to 14,000 B.t.u. per lb. It may be calculated by Dulong's formula.

ABSORPTION OF GASES AND WATER BY CHARCOAL.—Freshly burned charcoal contains practically no water, but will absorb 4 to 8% by weight in 24 hr., and 10 to 15% in a few weeks. The absorption of gases by charcoal is (Saussure):

NH_3....	90.00 volumes	N_2O....	40.00 volumes	N............	6.50 volumes
HCl.....	85.00 "	CO_2....	35.00 "	H (carbureted)..	5.00 "
SO_2.....	65.00 "	CO.....	9.42 "	H............	1.75 "
H_2S.....	55.00 "	O.......	9.25 "		

7. MISCELLANEOUS SOLID FUELS

PEAT, air-dried, contains from 15 to 30% of water. A typical analysis is: C, 58%; H, 6%; O, 31%; ash, 5%. Carbon may range from 63 to 51%, and ash from 11to 2%. The specific gravity is about 0.4 but it may be compressed to much greater density. Peat may be burned in the air-dried condition, or it may be briquetted, powdered or coked. The method of preparation, except coking, has little effect on heating value. Poole gives the following values per pound of moisture-free peat from Black Lake, N. Y.: Raw peat, 8237 B.t.u.; machine peat, 8050 B.t.u.; dry powder, 7971 B.t.u.; briquets, steam-dried, 8203 B.t.u. Peat coke ranges from 12,000 to 14,000 B.t.u. per lb. Table 40 gives analyses and maximum and minimum heating values of peats from various states of the U. S. Bulletin 37, Canadian Dept. of Mines gives the results of boiler tests fired with peat as follows: Calorific value of peat, 12,000 to 14,000 B.t.u. per lb.; moisture content, 15 to 20%; calorific value of dry peat, 8070 to 7590 B.t.u. per lb.; evaporation, 4.70 to 4.96 lb. per lb. of dry peat; efficiency, 51.3% to 54.8%.

Table 40.—Analyses and Calorific Values of Moisture-free Peat

Location		Volatile	Fixed Carbon	Ash	B.t.u. per lb.	Location		Volatile	Fixed Carbon	Ash	B.t.u. per lb.
Conn.....	min.	16.37	6.08	77.55	1,708	Michigan..	min.	42.54	18 03	39.43	5,845
	max.	61.17	31.58	7.25	10,001		max.	60.77	32.22	7.01	10,026
Florida...	min.	11.42	38.53	50.05	1,202	N. H......	min.	31.00	14.24	54.76	4,046
	max.	67.80	30.67	1.53	10,865		max.	66.74	28.67	4.59	10,280
Maine....	min.	29.88	12.31	57.81	3,634	New York..	min.	26.25	10.46	63.29	3,515
	max.	59.95	31.93	8.12	9,779		max.	67.10	28.99	3.91	10,307
Mass.....	min.	54.13	30.69	15.18	8,663	No. Carolina.....		51.88	28.83	19.29	8,249
	max.	57.04	34.61	8.35	9,308	Wisconsin..	min.	23.69	5.91	70.40	2,608
							max.	62.77	27.71	9 52	9,391

SAWDUST AND HOG FUEL.—Hog fuel is the refuse from sawmills which has been reduced to chips of uniform size by a machine called a hog. The unit of hog fuel is 200 cu. ft., equivalent to 1 cord of wood. Variation in voids, moisture and weight of wood will cause the weight of a unit to range from 2500 to 5000 lb., with 4000 lb. as a fair average. Sawdust and hog fuel have the same heating value as the wood from which they are made. (See p. 4–42.)

The evaporative value of hog fuel ranges from 8000 to 13,500 lb. of water evaporated from and at 212° F. per 200 cu. ft. of fuel. A few tests where the fuel has been weighed and the moisture determined have shown evaporations of from 3 to 6 lb. from and at 212° F. per lb. of dry fuel. See papers by H. W. Beecher, C. L. Young and C. C. Simeral, *Mech. Engg.*, July, 1928, for furnace designs and performances with hog fuel. See also p. 4–42.

Sawdust briquets, of a composition of sawdust, 65%, coal dust, 25%, binder, 10% are described in Bulletin 753, U. S. Dept. of Agriculture.

WET TAN BARK, or oak bark, after use in tanning may be burned as fuel in a furnace with very large combustion space. Spent tan contains about 65% moisture, and has an available heating value of about 2665 B.t.u. per lb. (D. M. Myers, *Trans.* A.S.M.E., 1909). The heating value of dry tan was found by six calorimeter tests to be 9504 B.t.u. per lb.; its composition was: C, 51.80%; H, 6.04%; O, 40.74%; ash, 1.42%. Mr. Myers states that with spent tan from 1.5 to 2.08 Hp. may be developed in horizontal tubular boilers per pound of tan.

STRAW from wheat dried at 230° F., and of composition C, 46.1%; H, 5.6%; N, 0.42%; O, 43.7%; ash, 4.1%, had a heating value of 6290 B.t.u. per lb.; with 6% water,

5770 B.t.u.; 10% water, 5448 B.t.u. Dry buckwheat straw showed a heating value of 5590 B.t.u. per lb., and dry flax straw, 6750 B.t.u. Barley straw, of composition C, 36%; H, 5%; N, 0.50%; O, 38%; ash, 4.75%, had a heating value, after deduction of heat lost in evaporating the water, of 5155 B.t.u. per lb.

BAGASSE, or refuse sugar cane after the juice has been extracted, contains 30–50% wood fiber, 0–10% sucrose and 40–60% water. The heating value of dry bagasse is about 8300 B.t.u. per lb. The net heating value is {(1 − moisture content) × heating value of dry bagasse} − (heat required to evaporate the moisture into steam). See Wood, p. 4–41. E. C. Freeland (*Trans.* A.S.M.E., xxxix, p. 611, 1913) gives calorific values of Louisiana bagasse as follows:

Moisture	Gross Calorific Value	Net Calorific Value
56.7%	3620 B.t.u. per lb.	2200 B.t.u. per lb.
48.2%	4800 B.t.u. per lb.	3350 B.t.u. per lb.

According to H. S. Colby (*Trans.* A.S.M.E., FSP-53-3, 1931), the use of air preheated to 400° F. in burning bagasse with 40 to 50% moisture reduced the amount of auxiliary fuel oil from 7 to 2 1/3 gal. per ton of cane crushed, and increased the boiler capacity from 85 to 135% of rating. The relative properties of Louisiana and Cuban bagasse are given by Mr. Freeland in Table 41.

The yield of bagasse from Louisiana cane is 400–600 lb. per ton. The evaporation from and at 212° F. per pound of bagasse is from 2 to 3 1/2 lb.; 4–6 lb. of bagasse are equivalent to 1 lb. of coal (14,000 B.t.u. per lb.); 43–65 lb. are equivalent to 1 gal. of fuel oil (19,000 B.t.u. per lb.). The bagasse from 1 ton of cane will generate 1.16 to 1.44 boiler Hp. per 24 hr. (E. W. Kerr). Drying the bagasse before firing will effect considerable economy. See Bulletin 128, Louisiana Agricultural Expt. Station for designs of driers and of boiler furnaces for bagasse; also *Trans.* A.S.M.E., FSP–53–3, 1931.

Table 41.—Comparison of Heating Values of Cuban and Louisiana Bagasse

Variety of Bagasse	Extraction, percent	Moisture, percent	Fiber, percent	Heating Value, B.t.u. per lb. Total	Net
With 80% Juice Extraction on Weight of Cane					
Cuban	80	32.8	60	5628	4092
Louisiana	80	42.8	50	4816	3345
With Nearly Equal Moisture Contents					
Cuban	75	42.6	48	4807	3335
Louisiana	80	42.8	50	4816	3345

PRESSED FUEL.—Coal briquets comprise fine or slack coal or lignite, mixed with a binder and pressed into small blocks. The most generally used binder is coal tar pitch; other binders that have been used are rosin, asphalt and lime. The binder varies with the coal. A binder that is successful with one coal will be unsatisfactory with another. Briquetting may render fit for use coal that otherwise is valueless, particularly low grade fuel as lignite. Table 42 (U. S. Bureau of Mines, *Bull.* 58, 1912) shows the increase in heating value of lignite due to the removal of moisture in briquetting.

Table 42.—Heating Value of Lignite Briquets

Lignite from	Moisture, percent: In Lignite	In Briquets	Removed	Heating Value, B.t.u. per lb.: Of Lignite	Of Briquets	Increase, percent
Texas	33.0	9.0	24.0	6840	9336	36.5
No. Dakota	40.0	12.0	28.0	6241	9354	50.0
"	42.0	10.0	32.0	6079	9355	54.0
California	40.0	10.0	30.0	6080	9264	52.4

The advantages of briquets, properly made with a suitable binder, as compared with raw fuel, are given in Bull. 58, U. S. Bureau of Mines as: More even and thorough combustion; produce little or no smoke, particularly with smaller sizes; retain their shape in the fire; do not cake and shut off air from upper surface of fire; burn to a fine ash without clinkering; fire requires less care; evaporative power greater than that of raw fuel at all rates of evaporation; weather resisting properties greater than that of raw fuel; higher rates of combustion are possible than with run-of-mine coal; loss from breakage during transportation is less; no danger of spontaneous combustion; block shaped briquets require less storage space than run-of-mine coal; have higher calorific value than the raw fuel. Briquets may be used as a substitute for raw fuel in domestic stoves and heaters and in locomotives. They usually cannot compete with coal in power boilers.

OIL REFINING WASTES may be burned as fuel in special types of burners. These wastes comprise petroleum coke, burned in pulverized form, refinery acid sludge, soda bottoms, neutralized sludge, acid tar, wax tailings and flux bottoms. The heating value of sludge ranges from 17,500 to 8000 B.t.u. per lb. A combined type of burner is used which is capable of burning either gaseous, liquid or pulverized solid fuel without changing any part of the burner. Several such types of burners have been devised, some of which will burn all three types of fuel simultaneously. In one of these, gaseous fuel enters the furnace from a ring-shaped gas chamber, through an annular opening, and meets a rapidly rotating current of air entering the burner through an air register in the outer end. Pulverized solid fuel and air enter tangentially a rifled coal-burner body, coaxial with and adjacent to the gas chamber. The stream of fuel is mixed with and carried by the secondary air from the register, rotating in the same direction. The oil is sprayed from a mechanical atomizer tip, located at the outer end of the coal-burner body. The oil issues in a divergent conical sheet, through which all of the secondary combustion air from the register passes. See paper by R. C. Vroom (*Trans.* A.S.M.E., FSP-53-27, 1931) for description of several types of combined burners, and discussion on burning of refinery waste. Refinery waste is used as fuel in the plant of the Louisiana Steam Products, Baton Rouge, La., together with natural gas and fuel oil, generating about 350,000 lb. of steam per hour. See article by J. F. Muir, *Power*, Jan. 13, 1931, and paper by H. J. Klotz, *Trans.* A.S.M.E., FSP-54-6, 1932.

LIQUID FUELS

By Harry F. Tapp

References.—A.P.I., Petroleum Facts and Figures, 4th Edition, American Petroleum Institute (1928, 1929, 1930, 1931). W. N. Best, Burning Liquid Fuel, U.P.C. Book Company (1921). Cochrane Corporation, Finding and Stopping Waste in Modern Boiler Rooms (1928). D. T. Day, Handbook of the Petroleum Industry (2 vol.), John Wiley and Sons (1922). Haslam and Russell, Fuels and Their Combustion, McGraw-Hill (1926). E. H. Peabody, Oil Fuel, *Trans.* Int. Engg. Congress (1915), and Recent Advance in Oil Burning, *Trans.* Soc. Nav. Arch. and Marine Engrs., vol. 28, 1920. Harry F. Tapp, Handbook of Oil Burning, American Oil Burner Assoc. (1931). W. Trinks, Industrial Furnaces, 2 vol., John Wiley and Sons (1923). Wadsworth, Efficiency in the Use of Oil Fuel, U. S. Bureau of Mines (1919).

1. CHARACTERISTICS OF FUEL OIL

FUEL OIL is defined (A.S.T.M., D288-31 T) as any liquid or liquefiable petroleum product burned for the generation of heat in an industrial or household furnace or firebox, or for the generation of power in a Diesel engine, exclusive of oils with a flash point below 100° F., Tag closed tester, and oils burned in cotton or wool-wick burners. Fuel oils are classed as: 1. Residual oils, *i.e.*, topped crude petroleums or viscous residuums obtained in refinery operations. 2. Distillate oils, derived directly or indirectly from crude petroleum. 3. Crude petroleums and weathered crude petroleums of relatively low commercial value. 4. Blended fuels, *i.e.*, mixtures of two or more of the above classes.

COMMERCIAL STANDARD SPECIFICATIONS for fuel oils cover six grades. They are limited by detailed requirements summarized in Table 1. The several grades are defined as: No. 1.—A light domestic fuel oil; a distillate fuel oil for burners requiring a volatile fuel. No. 2.—Medium domestic fuel oil; a distillate fuel oil for burners requiring a moderately volatile fuel. No. 3.—Heavy domestic fuel oil; a distillate fuel oil for burners requiring low viscosity. No. 4.—Industrial fuel oil, for oil burners requiring low viscosity fuel oil. No. 5.—Medium industrial oil, for burners adapted to medium viscosity fuel oil. No. 6.—Heavy industrial fuel oil, for burners equipped with oil preheaters, permitting use of high viscosity oil.

Flash Point (A.S.T.M., D93-22) is the temperature to which oil must be heated to give off sufficient vapor to form an inflammable mixture with air. It varies with apparatus and procedure used, and both must be specified when flash point is stated. The minimum flash point usually is controlled by law. If no legal requirements exist, minimum values of Table 1 are used. Maximum values are specified for oils 1-3 to insure required ease of ignition.

Water and Sediment (A.S.T.M., D96-28) are excluded almost entirely in oils 1-3, but allowed to limited extent in oils 4-6. Water and sediment are determined together by the centrifuge, except that in oil No. 6, water is determined by distillation and sediment by extraction with benzol.

Table 1.—Commercial Standard Specifications for Fuel Oil
(A.S.T.M. Designation D396–34T.)

Grade	Flash Point, deg. F.		Max. Water and Sediment, per cent	Ash, per cent	Sulphur, per cent	Max. Pour Point, deg. F.	Carbon Residue, per cent	Distillation Temp., deg. F.			Viscosity			
	Min.	Max.						10% Point	End Point	90% Point	Max.	Min.	Deg. F.	Viscosimeter
1	100 *	150	0.05	......	0.5	15	0.02	420 ‡	600 ‡	......	......	......	......	......
2	110 *	190	0.05 *	......	0.5	15	0.05	440 ‡	600 §	620 ‡	......	......	......	......
3	110 *	200	0.1	......	0.75	15	0.15	......	......	620 §	70	......	100	SU **
4	150	......	1.0	0.1 ‡	1.25	‖	......	......	......	......	500	70	100	SU **
5	150	......	1.0	0.15 ‡	¶	......	......	......	......	......	100	25	122	SF ††
6	150	......	2.0 †	......	¶	......	......	......	....	....	300	100	122	SF ††

* Or legal requirement. † A deduction in quality shall be made if sediment exceeds 1%. ‡ Maximum. § Minimum. ¶ No limit. ‖ May be specified when required by conditions of storage or use. ** Saybolt universal. †† Saybolt Furol.

Pour Point (A.S.T.M., D97–28) is the lowest temperature at which oil will flow under prescribed conditions.

Laboratory Distillation (A.S.T.M., D86–27 for oil 1, A.S.T.M., D158–28 for oils 2 and 3) of a sample under prescribed conditions is an index of volatility. The 10% and 90% points represent, respectively, temperatures at which 10% and 90% of the sample are distilled over. The end point is the maximum temperature recorded by the distillation thermometer at the end of distillation. The 10% point is an index of ease of ignition. The 90% point and the end point are specified to insure that the oil will burn completely and produce a minimum of carbon.

Viscosity * (A.S.T.M., D88–26) is the time in seconds in which a definite volume of oil will pass through a tube of specified dimensions at a definite temperature. It is a measure of the resistance of oil to flow. Viscosity decreases as temperature increases; preheating makes possible the use of oils of relatively high viscosities at normal temperatures. Maximum viscosity is limited because of its effect on oil flow in pipe lines, and on the degree of atomization that can be had in given equipment. The Saybolt Universal viscosimeter is used for low-viscosity fuel oils, and the Saybolt Furol viscosimeter for heavier oils. Other types of viscosimeters used with fuel oils are the Engler and Redwood.

The Nat. Elect. Light Assoc. Prime Movers Committee, 1931, gives a curve of the temperature-viscosity relations of a California fuel oil, from which the following figures are taken. Viscosity is in Saybolt universal seconds.

Temp. Deg. F.....	90	100	110	120	130	140	150	170	190	210
Viscosity.........	2300	1600	1150	820	600	450	340	200	110	85

Table 2, from Day's Handbook of the Petroleum Industry, gives approximate factors for converting viscosity as determined by one instrument to viscosity as determined by another. For additional information on viscosity determination and viscosimeters, see Battle's Industrial Oil Engineering (Lippincott) and Day's Handbook of the Petroleum Industry (John Wiley and Sons).

The viscosity-temperature chart (A.S.T.M. Method D341–32T) is convenient for estimating viscosities and temperatures other than standard test temperatures.

Specific Gravity is the ratio between the weight of any volume of oil at 60° F. and the weight of an equal volume of pure water at 60° F. It always is used for solid petroleum products and often for liquids. Except for exact laboratory work, gravity determinations on liquid petroleum are made by hydrometer (A.S.T.M., D287–30T), the depth to which it sinks in the liquid, as shown by the scale, determining specific gravity direct, or the gravity in degrees A.P.I. The A.P.I. gravity of pure water at 60° F. is 10°. The range for fuel oils is approximately 10° to 40° A.P.I.

To overcome the confusion due to the use of two so-called Baumé scales, for light liquids, the American Petroleum Institute, the U. S. Bureau of Mines and the U. S. Bureau of Standards agreed, in 1921, to recommend that only the scale based on the modulus 141.5 be used in the petroleum-oil industry, and that it be known as the A.P.I. scale. The relation of degrees A.P.I. to specific gravity is expressed by

$$\text{Degrees A.P.I.} = \{141.5/(\text{Sp. gr. } 60°/60° \text{ F.})\} - 131.5$$

Liquid fuels are purchased by volume. All gravity readings and volume determina-

* Absolute viscosity is the force required to move a plane surface of 1 sq. cm. at a speed of 1 cm. per sec. with reference to another plane surface, separated from the first by a layer of liquid 1 cm. thick. See p. 4–56.

Table 2.—Approximate Viscosity Conversion Table

Saybolt Uni Time, sec.			Multipliers for Converting Saybolt Universal sec. to Redwood, sec.	Engler, deg.	ybolt Uni Time, sec.			ler D, ime,	Multipliers for Converting Saybolt Universal Seconds to Saybolt Furol, sec.	Red-wood, sec.	Engler, deg.
				0.0328	200		160	5.55	....	0.800	0.0277
				.0325	225		180	6.22	....	.799	.0277
36	31	16	0.856	.0322	250	30	200	6.91	0.111	.798	.0276
38	32	21	.853	.0318	275	28	219	7.59	.110	.798	.0276
40	34	22	.850	.0316	300	33	239	8.27	.108	.798	.0276
45	38	40	.840	.0310	325	35	259	8.96	.108	.798	.0276
50	42	53	.834	.0306	350	38	279	9.64	.107	.797	.0275
55	46	66	.829	.0301	375	40	299	10.32	.107	.797	.0275
60	50	79	.825	.0298	400	43	319	11.01	.106	.797	.0275
65	53	92	.822	.0295	450	47	359	12.38	.105	.797	.0275
70	57	2.05	.819	.0293	500	52	398	13.75	.104	.797	.0275
80	65	2.32	.815	.0289	550	57	438	15.11	.104	.797	.0275
90	73	2.58	.811	.0287	600	62	478	16.49	.103	.797	.0275
100	81	2.85	.808	.0285	700	72	558	19.23	.103	.796	.0275
120	97	3.38	.805	.0282	800	82	637	21.97	.103	.796	.0275
140	112	3.92	.803	.0280	900	92	717	24.71	.102	.796	.0275
160	128	4.46	.801	.0279	1000	102	796	27.46	.102	.796	.0275
180	144	5.00	.800	.0278							

* Engler degrees are obtained by dividing the outflow time in seconds for 200 cc. of the oil, by the outflow time in seconds for 200 cc. of water at 68° F. The later figure is the "water factor" and should be between 51 and 52 seconds.

tions should be corrected to the standard temperature, 60° F. For correction tables, see Circular 154, U. S. Bureau of Standards. The unit of volume is the barrel (42 U. S. gal.) = 5.6 cu. ft. approx. Weight of fuel oil ordinarily is taken as 60 lb. per cu. ft., whence 1 bbl. = 336 lb. at 60° F. The coefficient of expansion of the average fuel oil is approximately 0.0004 per deg. F.

Carbon Residue. (A.S.T.M., D189-30.)—The carbon residue test, in connection with other tests and the use for which the oil is intended, furnishes information and throws light on the relative carbon-forming qualities of an oil. For medium viscosity and blended oils it is used to detect heavy residual products.

Ash. (A.S.T.M. D128-27.)—The ash test determines the amount of non-combustible impurities, which come principally from the natural salts present in the crude oil, or from chemicals used in refinery operations. They also may come from scale and dirt picked up from containers and pipes. Ash in fuel oils causes rapid deterioration of refractory materials in the combustion chamber, particularly at high temperatures. Some ash-producing impurities are abrasive and destructive to pumps, valves, control equipment and other burner parts. Ash specifications are included to minimize these operating difficulties.

Specific Heat of fuel oils varies from 0.4 to 0.55 for the temperature range generally used. Specific heat increases with temperature, and decreases as the specific gravity of the oil increases. The Prime-movers Committee of the Natl. Elect. Light Assoc., 1930, gives the following mean specific heats of a California oil of 18° A.P.I.

Temp. deg. F......	100	120	140	160	180	200	220	240	260	280
Mean Specific Heat.	0.450	.456	.463	.470	.479	.490	.502	.514	.527	541

HEAT CONTENT.—Exact determination of the heat content of fuel oil is made in a bomb calorimeter. Calorimeter determination is unnecessary unless the oil is bought on a B.t.u. basis. The heat content so varies with the gravity that the latter is a reliable index of the former. Approximate heating values for heavy oils may be computed by: B.t.u. per lb. = 18,650 + 40 (A.P.I.° − 10). See Publication 97, U. S. Bureau of Standards, for additional information on heat content. See Table 3 for analysis and heat content of typical oils.

CHEMICAL COMPOSITION OF PETROLEUM.—Petroleum is composed of carbon and hydrogen combined as hydrocarbons, and small quantities of oxygen, nitrogen, and sulphur. The range of ultimate analysis of fuel oils is C, 80-87%, H_2, 10-17%, S, 0-6%, O_2 and N_2, 0-6%.

The ultimate analysis determines the theoretical air required for combustion (See *Air Required to Burn Various Fuels*, p. 4-07), and the maximum CO_2 possible, which depends on the chemical analysis, or on the carbon-hydrogen ratio. The carbon-hydrogen ratio is

Table 3.—Analyses and Calorific Value of Various Fuel Oils

Oil	Chemical Analysis					Specific Gravity	Flash Point, deg. F.	Fire Point, deg. F.	B.t.u. per lb. as Reported	B.t.u. per lb. by Formula
	C	H	O	N	S					
Beaumont, Tex......	84.60	10.90	2.87	.....	1.63	0.92	142	181	19,060	19,142
Colinga, Cal.........	86.37	11.30	.....	1.14	0.60	0.95	162	......	18,720	18,948
Bakersfield, Cal......	85.0	12.0	1.0	0.2	0.8	....	......	......	18,600	....
Penna., crude.......	84.9	13.7	1.4 (O + N)		.....	0.89	......	......	19,210	19,350
Penna., light........	82.0	14.8	3.2 (O + N)		.....	0.83	......	......	17,930	19,809
West Va., crude.....	84.3	14.1	1.6 (O + N)		.....	0.84	......	......	18,400	19,736
Ohio, crude.........	83.4	14.7	1.3	.....	0.6	0.80	......	......	19,580	20,065
Mexican, crude......	82.8	12.19	0.43	1.72	2.83	0.91	77	120	18,493	19,215
Baku, Russia, heavy	86.6	12.3	1.11 (O + N)		.....	0.94	......	......	19,440	19,017

Table 4.—Relation of C-H Ratio, CO_2 and Excess Air

Percent Excess of Air	0	10	30	50	100
	Percent CO_2				
C-H ratio = 6 (light oil)...........	14.9	13.5	11.3	9.7	7.2
" " = 7 (medium oil)........	15.6	14.1	11.8	10.1	7.5
" " = 8 (heavy oil)..........	16.1	14.5	12.2	10.5	7.8

$(C + 0.4\,S)/H$. Table 4 gives relation between C–H ratio, percent of excess air, and CO_2. In practice, burning of fuel oil will give CO_2 as follows: High average, 12–14%; average, 10–12%; minimum (poor), 8%.

SPECIFIC HEAT OF FLUE GAS with oil fuel varies with temperature. F. G. Philo (*Trans.* A.S.M.E. FSP-54-11, 1932) presents a curve which shows it to vary in a straight line from 0.245 at 300° F. to 0.27 at 2000° F.

OIL COMBUSTION THEORIES.—Three theories of the burning of hydrocarbons are: 1. The hydrogen burns with oxygen before the carbon unites with oxygen. 2. The carbon burns in preference to the hydrogen. 3. A preliminary combination of oxygen with the hydrocarbon forms an intermediate hydroxylated compound, which, in turn, burns or is broken down thermally.

Investigations by W. A. Bone and others indicate a combination of hydrocarbons with oxygen, preliminary to final combustion (theory 3). Hydrocarbons combine with oxygen to form alcohol and aldehydes as a preliminary to burning to CO, CO_2, and H_2O. This holds true at all temperatures. Intermediate reactions at high temperatures may vary, but the initial hydroxylated molecule always is formed at the first contact with oxygen. No selective combustion of either carbon or the hydrogen of the hydrocarbon is evident. The initial addition of oxygen to the hydrocarbon molecule forms an alcohol which reacts with more oxygen to form an aldehyde. The aldehyde frequently breaks down into intermediate combustible gases, CO and H_2, or the aldehyde may burn completely to CO_2 and H_2O. This process is termed *hydroxylation.*

The heavier hydrocarbons usually undergo reactions other than simple hydroxylation, as do the lighter hydrocarbons if conditions are unfavorable to forming hydroxylated compounds. Heavy hydrocarbons may be cracked to saturated and unsaturated lighter hydrocarbons, or they may be decomposed completely into carbon and hydrogen. The lightest saturated hydrocarbons are cracked much more slowly than the heavier hydrocarbons.

In ordinary combustion of hydrocarbons no soot will form if conditions favor hydroxylation, *viz.*, premixture with air and ample time for oxygen to enter into the hydrocarbon molecule. If conditions favor cracking, a smoky flame results. For example, if hydrocarbons and oxygen from the air are not thoroughly mixed, the heat due to burning part of the hydrocarbons decomposes or cracks the remainder. For the conditions of ordinary combustion, the hydrocarbon, plus a small amount of oxygen, may be assumed to become a mixture of $CO + H_2$, which mixture burns as if the reactions were $2H_2 + O_2 = 2H_2O$, and $2CO + O_2 = 2CO_2$.

Soot and Scale Deposits from Oil Fuel.—The soot deposited by heavy oil fuel may contain acid-forming material, principally from the sulphur in the oil. If it becomes wet from boiler leakage or other sources, serious corrosion may result. To avoid this, the soot should be blown out at least every 48 hr. Representative soot and scale deposits (Report of Prime-movers Committee, Natl. Elect. Light Assoc., 1930) analyzed as follows:

Soot.—Loss on ignition (carbon, etc.), 51.15%; SiO_2, 1.89%; Fe_2O_3, 12.50%; CaO,

0.60%; MgO, 0.20%; alkalis as Na_2O, 6.24%; Cl, trace; sulphates, as SO_3, 27.4%. The acidity of this soot calculated as H_2SO_4, is 21.13%.

Scale.—Insolubles, as SiO_2, etc., 4.2%; Fe_2O_3, 1.9%; NiO, 2.0%; $CaSO_4$, 2.7%; Na_2SO_4, 89.2%. Some of the sodium sulphate was present as the acid sulphate.

COMPARATIVE FUEL COSTS are estimates of performance based on known and assumed factors, supposed to represent average conditions. Known factors are cost per unit, and heat content per unit. The assumed factor is the efficiency of utilization of the heat content of the fuel. To compare fuel costs intelligently, conditions under which the fuels are used must be studied, and each problem considered separately. The accuracy of an estimate of the probable relative consumption of two fuels will be in direct proportion to the accuracy of information available.

A formula for estimating the relative value of oil and other fuels is

$$X_o = (W_c \times H_c \times E_c) \div (H_o \times E_o)$$

where X_0 = units of fuel oil per unit of competing fuel; W_c = unit of weight or volume of competing fuel; H_c = heat content per unit of weight or volume of competing fuel; E_c = efficiency at which competing fuel is utilized; H_0 = heat content per unit of weight or volume of fuel oil; E_0 = efficiency at which fuel oil is utilized. Unit cost figures of the two fuels then may be used to determine comparative costs.

The actual efficiency of a given heating operation can be determined only by test. If tests are not available, assumed efficiencies must be used, based on knowledge and experience. Factors to be considered are fuel used, air-fuel ratio, furnace volume, combustion temperatures, amount and condition of heating surface, relation between heating surface and flow of gases, the heat absorbing medium (water, air, metal, etc.), load factor, load fluctuation, and temperature of the escaping combustion gases.

Other factors than fuel cost that must be considered to determine real comparative operating cost, are: 1. Cost of installation. 2. Labor. 3. Results obtained, *i.e.*, quality of finished product. 4. Uniformity or flexibility of heat control. 5. Reliability. 6. Maintenance. 7. Depreciation of equipment. 8. Operating cost of power for equipment, including auxiliaries. Installation cost is important, but should not be considered until study of the whole problem is complete. The advantages of a given fuel may so outweigh this factor as to make it negligible in the final analysis.

ADVANTAGES OF FUEL OIL.—(Haslam and Russell, Fuels and Their Combustion.) 1. Weight 30% less and space occupied 50% less than coal of equivalent heat content. 2. No deterioration in storage. 3. Freedom from spontaneous combustion. 4. Storage may be distant from furnaces. 5. Fuel is immediately available and may be stored or removed with practically no labor. 6. High combustion rates per cubic foot of combustion space. 7. Great flexibility in furnaces to readily and economically carry peak and valley loads. 8. Low labor cost to handle oil at the furnace and to clean boiler tubes 9. No labor for cleaning fires or removing ashes. 10. High efficiency and practically no smoke. 11. Absence of wear on machinery due to ash and dust. 12. Low pressure drop through the furnace. 13. A minimum of excess air required for complete combustion.

2. METHODS OF BURNING FUEL OIL

OIL BURNER TYPES.—Oil burners can be classified as: 1. Natural draft vaporizing burner. 2. Natural draft atomizing burner. 3. Mechanical draft vaporizing burner 4. Mechancal draft atomizing burner. They also can be classified, according to most common use made of each class, as: A. Domestic burners, using oils Nos. 1, 2 and 3, full- or semi-automatic, or manually controlled, used in domestic heating systems. B. Commercial burners, using oils Nos. 3, 4, 5 or 6, full- or semi-automatic, or manually controlled. used in heating boilers for apartment houses, office, manufacturing, and public buildings, etc., usually semi-automatic. C. Industrial burners using oil No. 6, used for industrial or power steam generation, and for using any grade of oil from No. 1 to 6, to supply heat for industrial processes.

Mechanical Draft Burners use a fan or blower to supply the air for combustion. Mechanical draft may be used with either vaporizing or atomizing burners. Its advantages are: 1. A more constant supply of air under varying draft conditions. 2. Velocity of the air may be used to increase turbulence or mixing effect. 3. Sufficient air for clean combustion from a cold starting condition is supplied without depending on chimney draft. 4. Can be used to develop high ratings when necessary. Its disadvantages are: 1. Increased first cost. 2. Cost of power. 3. Mechanical wear and noise.

ATOMIZATION breaks the fuel into fine particles that readily mix with the air for combustion. It then burns with a clean hot flame, being vaporized and oxidized by the resulting combustion before cracking takes place. In pressure atomizing burners the

fineness of spray increases as pressure increases and as viscosity decreases. In burning No. 6 oil, a pulsating flame results if viscosity is reduced to a point where the fuel tends to vaporize. Table 5, from U. S. Navy Manual of Engineering Instruction, shows the effect of pressure and viscosity on spray angles. This effect will vary with different burners and must be determined for each design size. If particles of oil or vaporized oil escape from the combustion zone because of improper atomization, combustion chamber design or air control, their heat content will escape with the flue gases. These losses are known as hydrocarbon losses and may be determined with the Burrell-Orsat gas analysis apparatus. Factors affecting hydrocarbon losses are: 1. Insufficient turbulence. 2. Cracking of oil and vapor in hot inert gases of combustion. 3. Cooling effect of excess air.

The advantages of atomization of oil are: 1. Atomizing burners can be used with heavier grades of oil. 2. Can be adapted to large applications because of larger capacity range. 3. Complete combustion is assured by the ability of the small particles to penetrate turbulent combustion. 4. Accurate metering of the fuel is possible, resulting in uniform combustion conditions. Disadvantages are: 1. Necessity of power-driven units to effect atomization. 2. Higher installation cost.

Table 5.—Effect of Pressure and Viscosity on Spray Angle of Pressure Atomizing Burners

Size	88° F. = 340 Seconds Saybolt Universal Viscosity					125° F. = 150 Seconds Saybolt Universal Viscosity					200° F. = 50 Seconds Saybolt Universal Viscosity				
	Pressure, pounds per square inch, gage														
	125	150	200	250	300	125	150	200	250	300	125	150	200	250	300
	Spray Angle, deg.														
5520	35	35	35	35	35	41	40	39	38	37	46	44	42	41	38
5320	36	37	37	38	38	46	45	44	42	41	53	52	50	48	46
5220	37	38	38	39	39	47	45	44	42	41	55	53	51	49	47
5020	41	42	43	43	45	50	49	48	48	47	57	57	57	54	52
4430	47	48	48	49	50	57	56	55	54	53	61	59	59	58	56

Mechanical Pressure Atomizers are designed for capacities ranging from 9 to 3000 lb. of oil per hour per burner. Oil is delivered to the burner, preheated if necessary for pumping and atomization, under pressures ranging from 40 to 250 lb. per sq. in. or more, depending on the quantity and grade of oil used. Pressure is varied to increase or decrease the capacity of the tip. With low-capacity burners, 1.35 to 10 gal. per hr. (diameter less than 0.030 in.), the capacity range generally is determined at pressures of 40 to 150 lb. per sq. in. for the lighter oils Nos. 1 to 4; for the heavier more viscous grades Nos. 5 and 6, the orifice size is increased and the capacities determined at pressures of 100 to 250 lb. per sq. in. The oil leaving the atomizer tip is broken into fine spray by centrifugal force, and by the rapid expansion following a sudden reduction in pressure. A regulating valve, manual or automatic, maintains a uniform pressure at the atomizer. Automatic valves usually are controlled by steam pressure.

Because of the high pressures used, the dimensions of the various parts of the atomizer must be held to limits of 0.001 to 0.005 in., if uniform results are to be expected. Even slight imperfections in the oil passages and orifice will cause faulty atomization. Care must be used in handling and cleaning burner tips. In shipping, storing and handling, they should be individually wrapped and protected against damage. The nose of the atomizer should have a shallow counterbore to protect the orifice.

The principal advantages of mechanical pressure atomization are: 1. Simplicity. 2. Uniform atomization. 3. Accurate and uniform metering of fuel. 4. High efficiency at high ratings. The disadvantages are: 1. Clogging of small orifices and passages (can be reduced materially by providing suitable strainers). 2. Capacity of individual tips is limited to narrow range (the larger the tip the greater the capacity range). 3. High preheating temperature required with heavy grades of oil.

Steam-atomizing Burners are classified as outside mixers or inside mixers. Both types can be designed to produce either round or flat flames. They are designed for capacities ranging from 5 to 1500 lb. of oil per hour per burner. A burner using steam for atomizing also may be used with compressed air. The oil is delivered to the burner oil-regulating valve at pressures ranging from 5 to 50 lb. per sq. in., preheated if necessary for pumping and atomization. The preheating temperature for steam-atomizing burners usually is lower than that required for mechanical pressure atomizers, as the viscosity is decreased by the heat of the steam used for atomization.

Steam usually is delivered to the burner steam-regulating valve at boiler pressure. It should be dry or superheated, as moisture causes the flame to sputter. The amount of steam used for atomization varies with the design of the burner, the skill of the operator,

and the boiler capacity. Under average conditions it ranges from 2 to 4% of boiler output. With competent operators and well-designed burners, the steam consumed may be as low as 1 1/2 to 2%; with careless operators or poor burner design it may be 4 to 6% or more. See Fig. 1 from paper by E. H. Peabody, The Burning of Liquid Fuel (*Trans.* A.S.M.E. FSP-50-13, 1928).

The principal advantages of steam-atomizing burners are: 1. Simplicity of design. 2. Low first cost of installation. 3. Low preheating temperature required. 4. Low pumping pressures. 5. Flexibility and high efficiencies at low and moderate rates of driving. 6. Ability to burn extremely heavy oils. The disadvantages are: 1. Steam consumption of burners. 2. Limitation in boiler capacity. 3. Decreased efficiency at high rates of driving.

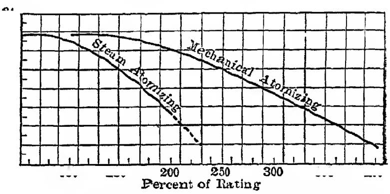


Fig. 1. Comparison of Steam and Mechanical Atomizing Burners

Low-pressure Air Atomizers.—Atomization of light oils can be accomplished satisfactorily with low-pressure air, without depending on oil pressure, if a sufficient quantity of air is supplied. The energy in a large volume of low-pressure air equals the energy in smaller volumes of air at higher pressures. The quantity of air required depends on design of burner, degree of atomization required, grade of oil, its pressure and temperature. Table 6 gives the approximate minimum quantity of atomizing air required at different pressures.

Table 6.—Percent of Combustion Air Required for Atomization

Air pressure, lb. per sq. in.	0.25	0.50		2	5	10	25	60	100
Percent of air required	68	52	42	33	25	19	13	9	7

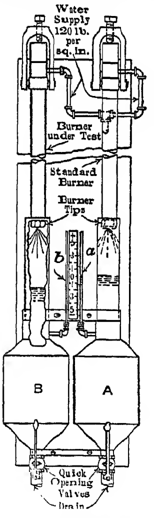


Fig. 2. Testing Device for Mechanical Oil Burner Tips

Mechanical Rotary Atomizers.—Oil is fed by a pump or gravity through a regulating valve to a rapidly rotating tapered cup. The oil enters at the rear, or point of least diameter, is carried forward and spread out in a film on the inner surface, and atomized by centrifugal force as it is thrown off the edge of the cup. The cup is rotated either by a motor or low-pressure air turbine. Rotary cup burners are designed for capacities ranging from 5 to 2000 lb. of oil per hour per burner.

The principal advantages of rotary disc, or cup, burners, are: 1. Flexibility over large capacity ranges. 2. High efficiencies. 3. Ability to burn heavy oils at low preheating temperatures. The disadvantages are: 1. High first cost. 2. Vibration and noise from moving parts. 3. Inefficient combustion resulting from operating burner beyond its rated capacity.

COMBINATION OIL AND GAS BURNERS.—The need for a reliable fuel supply in installations burning natural gas has led to the development of combination oil and gas burners. The necessity of thoroughly mixing natural gas and air for combustion makes the oil burner of any type ideal equipment for such combinations. The oil and gas ratings are such as to permit either fuel to be used for carrying the load.

TESTING OF OIL BURNER TIPS.—To insure equal amounts of oil being fed to each burner in a battery, frequent tests of burner tips are advisable. Tips wear, and leakage between burner parts may occur. Both tend to increase flow, and considerable variation in capacity of individual burners thus may exist. Natl. Elect. Light Assoc. Publication No. 30 illustrates the testing device shown in Fig. 2. The burner under test is compared with a standard burner by admitting water under 120 lb. per sq. in. pressure to both burners simultaneously. The discharge is caught in two receivers A and B fitted with manometers *a* and *b* to show the water level in each. The manometer of the test receiver B is graduated to show its contents in percentage of the standard receiver A. Flow continues through the two burners until the manometer on the standard receiver reads zero, when the inlet valve is closed and the variation of the test burner from standard is read directly on the test receiver manometer. A test of 60 used burners showed a maximum variation from standard of 16%; 40 new burners showed a maximum variation of 7%. All were corrected within 2% of standard.

PREPARATION OF LIQUID FUEL.—Many variations and combinations of the methods used to prepare liquid fuel for combustion, including domestic burners, will be found in Handbook of Oil Burning, by H. F. Tapp, 1931.

For information on various types of oil burners the data published by the following manufacturers is suggested.

Pressure Atomizing Burners.—*a*. Domestic; Automatic Delco Appliance Corp., Rochester, N. Y.; Gilbert and Barker Mfg. Co., Springfield, Mass.; May Oil Burner Corp., Baltimore, Md.

b. Commercial, Industrial and Marine: Anthony Co., Long Island City, N. Y.; The Babcock and Wilcox Co., New York; The Engineer Co., New York; Peabody Engineering Co., New York.

Air and Steam Atomizing Burners.—*a*. Domestic Automatic Air Atomizing: Caloroil Burner Corp., Hartford, Conn.; General Electric Co., Schenectady, N. Y.

b. Commercial and Industrial: W. N. Best Co., New York; National Airoil Burner Co., Philadelphia; Staples and Pfeiffer, Ltd., San Francisco.

Rotary Atomizing Burners.—*a*. Domestic Automatic: Automatic Burner Corp., Chicago; Timken Silent Automatic Co., Detroit; U. S. Burner Corp., Hartford, Conn.

b. Commercial, Industrial and Marine: S. T. Johnson, Oakland, Cal.; Ray Burner Co., San Francisco; Petroleum Heat and Power Co., Stamford, Conn.

Vaporizing Burners.—Domestic Automatic: Franklin Oil Heating, Inc., Columbus, Ohio; McIlvaine Burner Corp., Chicago; Par Appliances, Inc., La Crosse, Wis.

OIL BURNER CONTROLS may be classified as: 1. Manual. 2. Continuous operation (high-low). 3. Continuous operation (graduated control). 4. Intermittent operation (full automatic). 5. Combinations of 2 and 4, or of 3 and 4. With the exception of manual control and operation, the system may respond to changes in temperature or pressure, or both, through electrically or mechanically actuated devices. For data on oil-burner controls the data published by the following manufacturers is suggested.

Electric Controls for domestic, commercial and industrial burners: Minneapolis-Honeywell Co., Minneapolis, Minn.; Mercoid Corp., Chicago; Penn Electric Switch Co., Des Moines, Iowa.

Mechanical Controls for commercial and industrial burners. Fulton-Sylphon Co., Knoxville, Tenn.; Kieley Mueller Inc., New York; Preferred Utilities Mfg. Co., New York; Smoot Engineering Corp., Chicago.

A Full Automatic Control System incorporates a device responsive to changes in temperature or pressure, a sequence control device, and a device to ignite the oil or combustible mixture. If all operations except ignition are performed automatically, the control is *semi-automatic*. The method of control depends on the design characteristic of the burner, grade and amount of oil burned, and character of the load. Manual control largely is used in the industries for metal melting, varnish cooking, forge furnaces, ceramic furnaces, etc., where temperature or pressure control is relatively unimportant or where the operation requires close attention and the temperatures or pressures are changed during critical reactions in the process. In general, however, automatic controls are advantageous in most applications.

Limit Controls to protect boilers or furnaces from excessive operation of the burner respond to the amount of pressure or heat developed in boiler or furnace. These controls usually are designed for electric connection to the control of the burner. They are so connected that if the boiler or furnace control has broken the circuit, due to excess pressure or heat, the burner is inoperative regardless of the demand from the thermostat or other normal control device.

BURNER AND FLAME APPLICATION.—The design and type of burner governs the general shape of the flame, and determines in part the method of installing the burner and the path of flame in the boiler or furnace. Pressure atomizing burners usually have a cone-shaped flame, the included angle varying between 45 and 120 deg., depending on the atomizer tip design and control of air flow. By controlling air flow, it also is possible to obtain a flat flame. Steam- and air-atomizing burners in general can be designed to produce either flat or cone-shaped flames; rotary atomizers are limited to cone-shaped flames, whose included angle may be varied by control of air flow about the atomizer cup. Specific instructions for control of flame shape are given in manufacturers' instructions. Most burners are sufficiently flexible in this respect to be generally adaptable to boiler settings, but the type of boiler or furnace, and also the shape and size of the combustion chamber should be considered in selecting the type and number of burners. No more burners should be used than are needed to utilize the available combustion volume with sufficient flexibility to meet existing load conditions. For example, for a relatively narrow and deep combustion chamber, one burner is preferable, provided the oil rate can be varied to suit both the minimum and maximum load.

Adequate flame distribution requires that the flame conform as nearly as possible to the shape of the combustion chamber, for the reasons: 1. Maximum combustion volume is utilized. 2. Provides for thorough mixing of fuel and air during combustion process, insuring efficient combustion. 3. Permits using a minimum of excess air. 4. Reduces

hydrocarbon losses. 5. Is conducive to quiet combustion. 6. Maximum thickness of flame is obtained, providing maximum transfer of radiant heat. 7. Reduces flame impingement on refractory walls of combustion chamber.

DESIGN OF COMBUSTION CHAMBER.—Refractory combustion chambers are used, principally: 1. To permit maintenance of high temperature in combustion chamber to quickly vaporize raw fuel injected by the burner. 2. To protect parts of boiler or furnace that are not cooled properly. Additional benefit may be obtained by building brickwork to guide and increase flame travel in the combustion chamber.

The more important points to be considered in the design of combustion chambers are: 1. The refractory surfaces provide a radiant heating surface, which aids in combustion of the oil. Hence, it should be of minimum bulk so that it will heat rapidly and bring the combustion chamber to an efficient operating temperature. 2. Usually the combustion chamber must be so designed as to protect metal parts that are not cooled by water. 3. The shape of the combustion chamber should be such as to: *a*, get maximum flame distribution, giving consideration to the size and shape of the flame; *b*, provide maximum flame travel, taking into consideration location of fuel passes to the indirect heating surface. 4. All the direct heating surface of the furnace should, if possible, be exposed to the radiant heat of the flame and should not be covered with refractory material, except to prevent direct flame impingement. 5. For uniformly satisfactory flame distribution, square corners must be avoided; surfaces of the combustion chamber should curve upward to increase the turbulence within the chamber without creating downward eddies from the flame. 6. The center line of the burner should be high enough to prevent flame impingement on the hearth and allow air to circulate below the flame to insure complete combustion.

COMBUSTION VOLUME.—In any installation two combustion volumes are to be considered: 1. Available volume. 2. Effective volume.

Available Combustion Volume is the total volume enclosed by the boiler and its foundation or setting which may be used as combustion space. This is space enclosed by the floor or hearth, side walls, or water legs, up to the crown sheet or equivalent, which is a plane tangent to the bottom of the lowest row of tubes or other water-backed surfaces. A bridge wall should be considered as a side wall.

Effective Combustion Volume is the space actually occupied by the flame and circulating gases. It is controlled by the design, application and adjustment of the burner, and should be as nearly equal to the required available volume as conditions will permit. The required combustion volume varies with different classes of work, and usually is expressed as B.t.u. released per cu. ft. per hr. There are no fixed rules except that, in general, higher temperatures are involved with higher B.t.u. releases, and must be considered in connection with the life and services of refractory materials and structural supports of boiler or furnace. The intensity of combustion noise also tends to increase with high heat releases. Unless the refractory walls of the combustion chamber are cooled, high B.t.u. releases should not be used except in emergencies. The average range of B.t.u. releases for various types of installation is given in Table 7.

Table 7.—Average Range of B.t.u. Releases for Various Classes of Oil-burning Installations

Type of Installation	B.t.u Release per cu. ft. per hr.
Commercial heating boilers (large)	40,000– 70,000
High-pressure steam boilers (medium)...	40,000– 70,000
High-pressure steam boilers (large)	40,000–200,000
High-pressure steam boilers (marine)	40,000–100,000
High-pressure steam boilers (naval).....	60,000–200,000
Industrial furnaces....................	30,000–100,000

FIREBRICK AND REFRACTORY CEMENT.—Firebrick and refractory cement should be selected on the basis of the service in which they are used. A grade higher than absolutely necessary should be chosen because of abuse under extreme operating conditions. The life of refractory material in combustion chambers is shortened by sustained high temperature, rapid changes in temperature, and by panting or vibration from combustion. High temperatures result from: Operation above normal rating; normal operation with insufficient combustion chamber volume; flame impingement; combustion chambers designed for high B.t.u. releases. Rapid temperature changes may be reduced to a minimum by the operating personnel. A cold boiler should be brought up to operating temperature and pressure as slowly as possible. When securing the boiler, registers and dampers must be closed tightly to allow the boiler to cool slowly. Panting usually

is due to improper drafts, faulty atomization, fluctuating oil pressure or high B.t.u. releases. Sputtering results from water in the oil or wet steam supplied to steam-atomizing burners.

FURNACE FLOORS.—The burner manufacturer usually specifies the furnace floor construction. The several layers are as follows: 1. Insulating brick or material. 2. First course of brick, dry, laid $^1/_{16}$ in. apart to provide for expansion, joints broken between adjacent rows. 3. Dry refractory cement, filling all cracks and covering bricks to depth of $^1/_8$ in. 4. Second course of brick similar to first, overlapping joints in first course. 5. Dry refractory cement as in (3). After firing, the bricks take a permanent set and the cement vitrifies to a hard surface. For air ports built into the floor, the bricks may be set in refractory cement mortar.

3. STORAGE OF FUEL OIL

FUEL OIL STORAGE TANKS generally are classified by material, as steel or concrete; by size in gallons, etc.; by location, as exposed or inside, under-ground or buried; by use, as light or heavy oil tanks. The essential requirements for tanks are tightness and durability. The specifications of Underwriters' Laboratories, Chicago, for labeled tanks, or of the National Board of Fire Underwriters, are generally accepted standards. See Tables 8, 9 and 10. Some cities and states require special construction, and local regulations should be studied before installation. Tanks for heavy oil usually have a manhole and provision for a tank preheater, using either steam or hot water. Such tanks should be designed to heat the oil in the vicinity of the suction pipe to not over 100° F. See Table 11.

Table 8.—Specifications for Under-ground Oil Storage Tanks

(National Board of Fire Underwriters, Revised 1931)

Maximum capacity, gal.	285	560	1100	4000	12,000	20,000	30,000
Gage of metal	16	14	12	7	$^1/_4$ in.	$^5/_{16}$ in.	$^3/_8$ in.
Weight of metal, lb. per sq. ft.	2.50	3.125	4.375	7.50	10.00	12.50	15.00

Top of under-ground tanks to be not less than 2 ft. under-ground. Material to be galvanized steel, basic open-hearth, or wrought iron. Joints to be welded, or riveted and caulked.

Table 9.—Specifications for Above-ground Oil Storage Tanks

(National Board of Fire Underwriters, Revised 1931)

Maximum capacity, gal.	60	350	560	1100	Over 1100
Gage of metal	18	16	14	12	See Note 1

NOTE 1. Thickness of metal for above ground tanks to be calculated by the following formula: $t = 2.5\ HDS/TE$, where t = thickness of metal, in.; H = height of tank above bottom of ring under consideration, ft.; D = diam. of tank, ft.; S = specific gravity of liquid; T = tensile strength of plate, lb. per sq. in.; E = efficiency of joint in ring. Minimum values of t are: Tanks not over 6 ft. diam., $t = {}^3/_{16}$ in.; tanks over 6 ft. diam., $t = {}^1/_4$ in.

Table 10.—Spacing of Fuel Oil Tanks

(National Board of Fire Underwriters)

Distance between Tanks, ft.	Maximum Capacity, gal.		Distance between Tanks, ft.	Maximum Capacity, gal.		Distance between Tanks, ft.	Maximum Capacity, gal., Above-ground
	Under-ground	Above-ground		Under-ground	Above-ground		
5	....	750	40	500,000	45,000	100	266,000 †
10	75,000	1,100	50	Unlimited	64,000 *	150	400,000 †
20	100,000	3,000	60	"	80,000 *	250	666,000 †
25	150,000	21,000	75	"	125,000 *	300	1,333,000 †
30	200,000	31,000	85	"	200,000 *	350	2,666,000 †

* May be increased 33% if provided with approved extinguishing apparatus. † Minimum distance may be 175 ft. if provided with approved extinguishing apparatus that also covers other parts of yard and system.

Care of Tanks.—Oil tanks should be cleaned at least once a year. Water and foreign material which settles out of the oil, if allowed to remain on the tank bottom, will accelerate corrosion. Large storage tanks should have a manhole for entrance for periodic examinations and cleaning. To insure that all oil vapors have escaped, before entering the tank, the manhole should be left open for several hours, and air circulation established by steam, compressed air, or a fan. The tank should be examined carefully, the bottom thoroughly cleansed, and all discolored or rusty spots scraped and painted with a compo-

sition paint, insoluble in oil or water, made especially for this purpose. Ordinary paint is unsuitable, being soluble in oil.

OIL PUMPS.—The primary purpose of an oil pump in connection with oil burners is to draw oil from the storage tank by suction and deliver it to the burner. A secondary purpose, sometimes, is to deliver the oil under sufficient pressure to produce the atomization necessary for combustion. (See page 4–50.)

PIPING SYSTEMS depend on the design of burner, storage system, grade of oil and the requirements of local authorities.

Heavy Oil Systems.—The principal difference between the piping for light and heavy oils is the use of preheaters. Consult burner manufacturers for detailed requirements for individual types of burners. The kind and amount of equipment necessary depend on design of burner, grade of oil used, location of tank, character of load, and degree of automatic operation required. Fig. 3 shows a typical heavy oil piping layout.

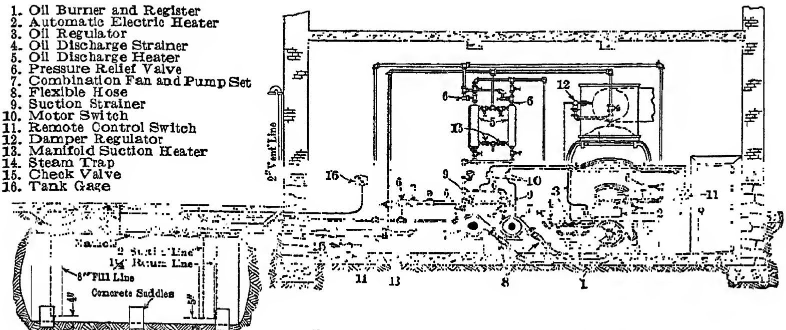


FIG. 3. Typical Heavy Oil Piping Layout

Fill Lines.—Not less than 2-in. pipe should be used for light oils (No. 1); for heavy oils (No. 6), 6-in. or 8-in. pipe should be used. A pipe too large is better than one too small. The fill line for any storage tank should pitch from the fill box to the tank A trap should be provided, either directly inside or outside of the tank, or the fill line sealed by ending it in the tank below the bottom of the suction line. The fill line always should be connected at the low end of the tank, and never cross-connected to the vent pipe.

Vent Pipe.—All fuel oil storage tanks must be vented. The size of the vent pipe should be in proportion to the size of the fill line, and never should be less than 1 1/4-in. pipe. The vent line should be without traps, pitched up from the tank and connected near the high end. It should be installed with swing joints, and should not be cross-connected with any line other than a vent. The vent pipe terminal should be visible from the fill box, should be weather-proof and preferably without screens.

The Suction Line is a most important part of an installation. Between buried storage tanks and the building wall, it should pitch toward the tank, and always should have swing joints. Usually a foot-valve is advisable on the suction line inlet in the tank, about 2 in. above the bottom of the tank for light oil, and 4 in. above for heavy oil. A testing tee should be in the suction line to permit testing at any time.

Strainers.—Every oil line to the burner should have a suitable strainer, either suction or discharge. If the plant is shut down regularly single strainers are satisfactory, but for continuous operation, duplex strainers are advisable. The size of mesh depends on grade of oil used, location of strainer and design of burner. In general, the mesh should not be finer than necessary to protect equipment in the portion of the line where the strainer is located. For light oils, suction strainers are 30 mesh or finer; discharge strainers 40 mesh or finer. For heavy oil, suction strainer meshes range from 8 to 30; discharge strainers from 20 to 30 mesh. The area of strainer surface is of equal importance as the size of mesh. Suction strainers should be at least double the area of the suction line. Discharge strainers should have at least 1 sq. in. area per gallon of oil burned per hour.

PREHEATERS may be classified as: 1. Use, *i.e.*, tank, suction, discharge, auxiliary. 2. Construction, *i.e.*, manifold, coil, multipass, film. 3. Heating medium, *i.e.*, steam, hot water, electric.

Tank Heaters are used only where oil temperature in the tank is so low that oil cannot be pumped. Tank heaters sometimes are called suction heaters. Often, however, for

long suction lines a manifold heater is used to insure a steady flow of oil. Steam or hot water furnishes heat, depending on the grade of oil, the temperature required, and the size of heater. Steam generally is used, but some city ordinances require hot-water heaters to insure control of maximum temperature.

Discharge Heaters reduce the viscosity of the oil to permit atomization. They always should be used for oils Nos. 5 and 6, regardless of burner design. With pressure atomizing burners they often can be used to advantage with No. 4 oil. At least two discharge preheaters should be provided, each of sufficient capacity to carry the normal load, to permit cleaning and inspection without interrupting service. For wide variations in load, several heaters should be installed, to be cut in and out as needed. The best practice uses one heater at maximum capacity rather than several in parallel or partial capacities. To avoid excessive heat losses, discharge heaters should be placed as close to the burner as structural conditions will permit.

Auxiliary Heaters.—Auxiliary electric heaters, used to preheat heavy oils, permit starting when the system is cold. This type of heater also is incorporated in full automatic heavy oil burners. It requires approximately 0.0146 kw-hr. to raise 100 lb. of oil 1° F. in one hour. This type also is incorporated in full automatic heavy oil burners.

Oil Temperature.—The temperature required for the oil depends on the design of burner, and viscosity of the oil. Table 11 gives the range of temperatures commonly used. The table is based on gravity and should be used only as an approximation. The correct temperature required to *reduce the viscosity* of the oil to from 70 sec. to 140 sec. Saybolt Universal should be determined by test, or A.S.T.M. specification D341-32T, and maintained. Gravity is not an accurate index to viscosity. Because of operating difficulties in pumping and atomizing, oil should not be heated above its flash point.

Table 11.—Approximate Preheating Temperatures for Fuel Oil

Gravity, A. P. I.	Temperature, deg. F.	Gravity, A. P. I.	Temperature, deg. F.	Gravity, A. P. I.	Temperature, deg. F.
10–12	275–325	16–18	150–200	22–24	160–200
12–14	220–275	18–20	140–160	24–26	70–80
14–16	175–250	20–22	100–140	....	....

Preheaters designed for steam generally should have from 0.15 to 0.3 sq. ft. of heating surface for each gallon per hour capacity, depending on the grade of oil used and design of heater. So many variables are involved, *i.e.*, types of heaters (manifold, coil, multipass and film), heating mediums (high- and low-pressure steam, hot water), temperature required by service conditions, grade of oil, types of burners, etc., that preheaters either should be selected on burner manufacturers' recommendations, or designed as a heat exchanger on the basis of data applying to specific installation requirements. Heaters should be examined regularly to detect leaks between the oil and steam passages. Such leaks usually can be detected by examining a sample of condensate from the heater. Heaters should be cleaned regularly, using steam in the oil passes and boiler compound in the steam passes. After disconnection or repair, heaters should have a hydrostatic pressure test before being put in service. Good practice provides discharge heaters with a by-pass relief valve to the suction or return line, to prevent damage by abnormal pressures if the oil lines of the heater are cut out and the steam left on.

TEST CODES.—All data, observations and calculations necessary for testing stationary steam generating units are given in the A.S.M.E. Boiler Test Code. See p. 16–13. For low-pressure boilers, use the A.S.H.V.E. Standard Code for Testing Steam Heating Boilers Burning Oil Fuel.

4. FLOW OF OIL*

The flow of oil in pipes is influenced by the viscosity of the oil, which varies with temperature, and by the character of flow, *i.e.*, streamline or viscous, and turbulent.

VISCOSITY is that property of fluids by which they resist an instantaneous change of shape or arrangement of their molecules.

Absolute Viscosity is defined as the force required to move a plane surface of the fluid, of 1 sq. cm. area, parallel to another plane surface of the fluid at 1 cm. distance from the first, and with a relative velocity of 1 cm. per sec. The unit of viscosity is the poise = 1 dyne-cm. per sec. per sq. cm. The centipoise = 0.01 poise usually is used in engineering calculations. The viscosity of water at 68° F. is 1 centipoise. The temperature at which viscosity is measured always should be stated. The symbol of absolute viscosity is μ.

To convert viscosity as measured in poises to viscosity as measured in English units,

* Revised by Robert Thurston Kent.

the following conversion factors are used: 1 poise = 100 centipoises = 0.0672 lb. mass per ft. per sec. = 0.0672 poundal-sec. per sq. ft. = 0.00209 lb. force sec. per sq. ft.

Kinematic Viscosity, whose symbol is y, is the ratio of the absolute viscosity μ, to the density ρ, of the fluid, *i.e.*, $y = \mu/\rho$. Kinematic viscosity is more useful in engineering work, as it can be determined directly from viscosimeter readings (see p. 4–46) by using the equations of Table 12. Kinematic viscosity also is defined as the ratio of absolute viscosity to specific gravity. The former definition, however, is preferred.

Table 12.—Equations for Converting Viscosimeter Readings to Kinematic Viscosity

t = time of flow seconds

Viscosimeter	Metric Units y = sq. cm. per sec.		English Units y = sq. ft. per sec.	
Saybolt Universal *	$y = 0.0022\,t - (1.8/t)$.	[1]	$y = 0.00000237\,t - (0.00194/t)$.	[1a]
Engler	$y = 0.00147\,t - (3.74/t)$.	[2]	$y = 0.00000158\,t - (0.00403/t)$.	[2a]
Redwood	$y = 0.0026\,t - (1.72/t)$.	[3]	$y = 0.00000280\,t - (0.00185/t)$.	[3a]

* Saybolt Furol = (Saybolt Universal/10) Approx. See Table 2.

Relative Viscosity is the ratio of the absolute viscosity of a fluid to that of water. The symbol of relative viscosity is Z.

Density, ρ, is defined as the mass per unit volume. It also is defined as the weight per unit volume, as determined by balancing against standard weights, and not as measured on a spring balance.

Variation of Viscosity with Temperature.—The absolute viscosity of petroleum oils varies with temperature, decreasing as temperature increases. Between 32° and 400° F. the variation will be in a straight line when plotted on logarithmic paper.

STREAMLINE AND TURBULENT FLOW.—In streamline or viscous flow, each particle of fluid moves in a direction parallel to the axis of the channel or pipe. In a pipe, the velocity of flow increases from a minimum at the circumference to a maximum at the center. The average velocity over the cross section of the pipe is one-half the maximum velocity. The velocity at any radius r is

$$V = \{(P_1 - P_2)/4L\mu\} \times \{(D^2/4 - r^2\} \quad . \; . \; . \; . \; . \; . \; . \; . \quad [4]$$

where in any homogeneous system of units V = velocity; r = radius; P_1 and P_2 = higher and lower pressures, respectively; L = length; μ = absolute viscosity; D = diameter. If velocity distribution be plotted with radii as ordinates, and velocities as found by equation [4] as abscissas, the resulting curve is a parabola with its vertex in the axis of the pipe.

When the velocity increases beyond a certain value the flow ceases to be streamline and becomes **turbulent.** The point at which streamline flow ceases is determined by the Reynolds number (see below). Streamline flow usually exists with values of Reynolds numbers below 2000. For values between 2000 and 3000, the flow is unstable, and this range is called the *critical velocity.* The point at which flow changes from streamline to turbulent is the *critical velocity region.* Flow is turbulent when the Reynolds number is above 3000. Flow in the critical range produces excessive pressure drops without corresponding increases in volume discharged. Oil pipes, therefore, should be so designed as to avoid flow in the critical region.

Reynolds Number.—Prof. Osborne Reynolds in 1874 showed that the critical velocity varies directly as the absolute viscosity of the fluid and inversely as the pipe diameter and the density of the fluid. These relations are expressed in the Reynolds number. For a discussion of Dimensional Analysis, on which Reynolds numbers are based, see Eshbach, Handbook of Engineering Fundamentals, Sect. 3 (John Wiley & Sons), and Barnard, Ellenwood and Hirshfeld, Elements of Heat-Power Engineering, Part III, p. 795 (John Wiley & Sons).

$$\text{Reynolds Number} :: \frac{\text{diameter} \times \text{velocity} \times \text{density}}{\text{absolute viscosity}} = \frac{D\,V\,\rho}{\mu} \quad . \; . \; . \quad [5]$$

$$= \frac{\text{diameter} \times \text{velocity}}{\text{kinematic viscosity}} = \frac{DV}{y} \quad . \; . \; . \; . \; . \; . \; . \; . \quad [6]$$

In computing a Reynolds number a homogeneous system of units should be used if formulas [5] and [6] are to be applied directly, for example, metric dimensions and viscosities in poises or centipoises. If dimensions are expressed in the English system, the conversion factors given above under definition of absolute viscosity must be introduced.

Example.—Given a liquid of viscosity 30 centipoises (=0.3 poise), density 0.9 gram per cu. cm., flowing at a velocity of 300 cm. per sec. in a pipe 10 cm. diam.; Reynolds number = $(10 \times 300 \times 0.9)/0.3 = 9000$.

In English units, density = $0.9 \times 62.43 = 56.187$; velocity = $300 \times 0.03281 = 9.843$ ft. per sec.; diam. = $10 \times 0.03281 = 0.3281$ ft. Conversion factor = 0.0672. Reynolds number = $(0.3281 \times 9.843 \times 56.187)/(0.3 \times 0.0672) = 9000$.

VOLUME OF FLOW.—In the following discussion, based on article by Prof. W. F. Durand in Day's Handbook of the Petroleum Industry (John Wiley & Sons), the units are all in the English system, *viz.*, pounds, feet, and for viscosity, poundals.

Notation.—μ = absolute viscosity, poundal-sec. per sq. ft. = 1488 centipoises; ρ = density, lb. per cu. ft.; D, d = diameter, ft. and in., respectively; L = length, ft.; V = velocity, ft. per sec.; P_1, P_2 = higher and lower pressures, respectively, lb. per sq. ft.; h = head, ft. = $(P_1 - P_2)/\rho$ lb. per sq. ft.; g = acceleration due to gravity = 32.2; R = Reynolds number = $DV\rho/\mu$; K = a constant.

The head causing the flow of oil is

$$h = \phi \frac{[illegible]}{\mu} \times \frac{[illegible]}{D} \frac{[illegible]^2}{2g} \quad . \quad . \quad . \quad . \quad . \quad . \quad . \quad . \quad . \quad [7]$$

If $\phi(DV\rho)/\mu$ be put equal to f, equation [7] corresponds to the familiar D'Arcy equation for the flow of water

$$h = f(L/D)(V^2/2g) \quad . \quad . \quad . \quad . \quad . \quad . \quad . \quad . \quad . \quad . \quad [8]$$

For a given value of $R = (DV\rho/\mu)$ the value of f in equation [8] will be constant, irrespective of the individual values of D, V, ρ and μ. Table 13 gives values of f for various values of R. If R be expressed as $DV \times (\rho/\mu)$, the value of (ρ/μ) can be obtained from the readings of a Saybolt Universal viscosimeter by means of Table 14. This table gives the reciprocals of the values of $y = \mu/\rho$ as computed by formula [1a]. The value of R, and consequently of f, for any combination of values of D and V thus is easily determined from the viscosimeter reading.

The volume of oil flowing depends on the area of the pipe and the velocity. From equation [8]

$$\sqrt{[illegible]}$$

where K is a constant depending on the diameter of the pipe and the value of f. Values of K are given in Table 15. Values intermediate to those given in the table can be obtained by plotting on logarithmic paper any two values of K for a given pipe diameter and drawing a straight line between the points. Intermediate values of both K and f then can be read. Volume, in cu. ft. per sec., then is

$$Q = VA = \frac{[illegible]}{4} \quad [10]$$

Values of $A = (\pi D^2/4)$ may be taken from Table 15. The volume flowing measured in any other units is

$$[illegible] \quad [11]$$

where C is a constant whose value is as follows.

Unit...	Cu. ft. per min.	Cu. ft. per hr.	U. S. gal. per min.	U. S. gal. per hr.	Barrels (42 gal.) per hr.	Barrels (42 gal.) per day
C......	60	3600	448.8	26,928	641.1	15,386

If L is taken in miles instead of feet, the constant C becomes

$$C_1 = C/\sqrt{5280} = 0.013762C.$$

The only uncertain factor is the value of f. Emory Kemler (*Trans.* A.S.M.E., HYD-55-2, 1933) in an exhaustive study of the work of previous investigators, concludes: 1. Value of f is independent of the fluid flowing for identical values of R. 2. For brass pipe, f varies only about ±5% from an average value and is substantially independent of pipe size and fluid flowing for identical values of R in the range usually met in practice. 3. For steel pipe, f varies about ±10% from an average value, and varies with size because roughness increases with pipe diameter. 4. If roughness is not large enough to cause contraction and enlargement losses, f never exceeds 0.054. Kemler further states that the effect of diameter on flow is important, since f varies as D^5. Hence, a small change in D produces a large change in f. For example, if a value of $D = 1$ is used in design, whereas the actual value is 0.9, the pressure drop will be about 70% greater than the value based on $D = 1$.

Streamline Flow.—The value of f for streamline flow in circular horizontal pipes has been determined experimentally to equal $64/R = 64\mu/DV\rho$. If this is substituted in equation [8] we have

$$h = (32\mu LV)/gD^2\rho \quad . \quad . \quad . \quad . \quad . \quad . \quad . \quad . \quad . \quad . \quad [12]$$

which is known as Poiseuille's law. It indicates that for streamline flow in horizontal

pipes the drop in pressure varies directly as the absolute viscosity, average velocity and length of pipe, and inversely as the square of the pipe diameter.

Turbulent Flow.—The value of f in turbulent flow depends almost entirely on the roughness of the pipe. There has been no rational method of evaluating roughness, and heretofore the selection of f for a given set of conditions has been largely a matter of experience and judgment. R. J. S. Pigott (The Flow of Fluids in Closed Conduits, *Mech. Engg.*, Aug. 1933) has evolved a method which, while empirical, is well borne out by experimental data. Basing his method on Kemler's studies (see above) Pigott

Table 13.—Values of f Corresponding to Various Reynolds Numbers

Streamline Flow		Turbulent Flow					
$DV\rho/\mu$	f	$DV\rho/\mu$	f	$DV\rho/\mu$	f	$DV\rho/\mu$	f
100	0.6400	2,500*	0.0442	14,000	0.0292	70,000	0.0195
200	.3200	3,000	.0426	16,000	.0280	80,000	.0190
400	.1600	3,500	.0412	18,000	.0271	90,000	.0185
600	.1067	4,000	.0400	20,000	.0264	100,000	.0180
800	.0800	4,500	.0390	25,000	.0249	150,000	.0168
1000	.0640	5,000	.0382	30,000	.0238	200,000	.0158
1200	.0533	6,000	.0364	35,000	.0228	250,000	.0150
1400	.0457	7,000	.0350	40,000	.0219	300,000	.0144
1600	.0400	8,000	.0340	45,000	.0213	350,000	.0140
1800	.0356	9,000	.0330	50,000	.0208	400,000	.0137
2000	.0320	10,000	.0320	60,000	.0200	450,000	.0134
2400*	.0267	12,000	.0304				

* In critical region; flow unstable, and may be either streamline or turbulent.

Table 14.—Values of ρ/μ Corresponding to Various Saybolt Universal Viscosities

Saybolt Seconds	ρ/μ	Saybolt Seconds	ρ/μ	Saybolt Seconds	ρ/μ	Saybolt Seconds	ρ/μ
40	21,600	90	5216	200	2154	1000	422
50	12,550	100	4596	300	1419	1500	281
60	9,100	125	3562	400	1060	2000	211
70	7,236	150	2919	500	847		
80	6,050	175	2477	750	564		

Table 15.—Value of K in Formula for Flow of Oil

f	Nominal Diameter of Pipe, in.							
	12	10	8	6	5	4	3	2
	D = Actual Diameter of Pipe, ft.							
	1.000	0.8349	0.6650	0.5054	0.4205	0.3355	0.2555	0.1726
	A = Actual Area of Pipe, sq. ft.							
	0.7854	0.5476	0.3474	0.2006	0.1389	0.0884	0.0513	0.0234
	Values of K							
0.020	56.71	51.82	46.24	40.32	36.78	32.85	28.67	23.56
.021	55.34	50.57	45.14	39.35	35.88	32.06	27.99	22.99
.022	54.07	49.40	44.09	38.44	35.06	31.32	27.32	22.46
.023	52.88	48.32	43.12	37.59	34.29	30.64	26.73	21.97
.024	51.77	47.30	42.22	36.80	33.57	29.99	26.17	21.51
.025	50.72	46.35	41.37	36.07	32.89	29.38	25.64	21.07
.026	49.74	45.45	40.57	35.36	32.25	28.81	25.14	20.65
.027	48.81	44.60	39.80	34.70	31.65	28.28	24.67	20.27
.028	47.93	43.79	39.08	34.08	31.09	27.76	24.22	19.91
.029	47.10	43.04	38.41	33.48	30.54	27.28	23.80	19.56
.030	46.30	42.31	37.75	32.91	30.03	26.82	23.41	19.23
.031	45.55	41.62	37.14	32.38	29.55	26.38	23.03	18.93
.032	44.83	40.97	36.56	31.87	29.07	25.96	22.66	18.63
.033	44.15	40.34	36.00	31.38	28.62	25.57	22.31	18.34
.034	43.50	39.74	35.47	30.92	28.21	25.20	21.98	18.08
.035	42.87	39.17	34.96	30.48	27.80	24.84	21.67	17.81
.036	42.27	38.62	34.47	30.05	27.41	24.49	21.37	17.56
.037	41.69	38.10	34.00	29.64	27.04	24.16	21.08	17.33
.038	41.14	37.59	33.55	29.25	26.68	23.84	20.79	17.09
.039	40.61	37.11	33.11	28.87	26.33	23.52	20.52	16.87
.040	40.10	36.64	32.71	28.51	26.00	23.23	20.27	16.67
.050	35.87	32.78	29.25	25.50	23.26	20.77	18.13	14.89
.060	32.82	29.92	26.71	23.28	21.24	18.96	16.55	13.61

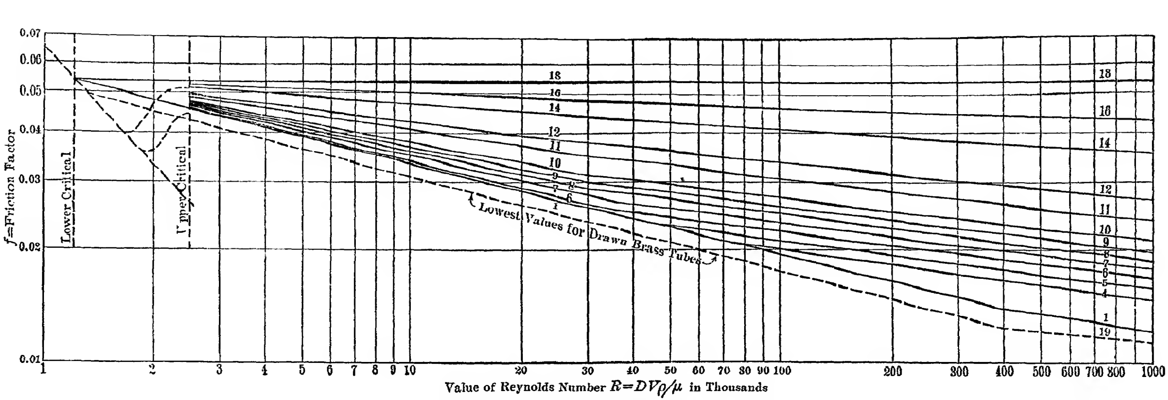


Fig. 4. Value of Friction Factor f for Various Classes of Pipe. See Table 16

Table 16.—Relation of f to Character, Size and Roughness of Pipe

See Curves Fig. 4

Percent Roughness	Value of f. See Curve No.	Drawn Tubing, Brass, Tin, Lead, Glass	Clean Steel, Wrought-iron	Clean Galvanized	Best Cast-iron	Average Cast-iron	Heavy Riveted, Spiral Riveted
		Diam., in. (Actual of drawn tubing; nominal of standard weight pipe)					
0.2	1	0.35 up	72	......	......	......	......
1.35	4	......	6 to 12	10 to 24	10 to 48	42 to 96	84 to 204
2.1	5	......	4 to 5	6 to 8	12 to 16	24 to 36	48 to 72
3.0	6	......	2 to 3	3 to 5	5 to 10	10 to 20	20 to 42
3.8	7	......	1 1/2	2 1/2	3 to 4	6 to 8	16 to 18
4.8	8	......	1 to 1 1/4	1 1/2 to 2	2 to 2 1/2	4 to 5	10 to 14
6.0	9	......	3/4	1 1/4	1 1/2	3	8
7.2	10	......	1/2	1	1 1/4	......	5
10.5	11	......	3/8	3/4	1	......	4
14.5	12	......	1/4	1/2	......	......	3
24.0	14	0.125	......	3/8	......	......	......
31.5	16	......	......	1/4	......	......	......
37.5	18	0.0625	......	1/8	......	......	......

derived the curves, Fig. 4, which give the value of f for various classes of pipe with various degrees of roughness. Kemler has shown that roughness varies with diameter and character of pipe. From his data, Pigott prepared Table 16, which shows the effect of these two variables and refers the reader to the proper curve in Fig. 4, from which to select a value of f consistent with the given conditions.

Pigott bases his method on the assumption that the roughness of the surface has the effect of entangling a film of the fluid somewhat thicker than the depth of the projecting rough portions, flowing at low speed close to the wall of the pipe, and thereby virtually reducing the diameter of the pipe. Since the flow varies as D^5, a small increase in roughness will cause a large variation in flow. For a given pipe diameter, the thickness of the film would decrease with increased speed or density, and would increase with increase of viscosity and roughness. Pigott concludes that the values of f approach the values for smooth pipes as the diameter increases, and must lie between the smooth pipe curve and a constant value of 0.054 for entirely rough surfaces.

5. GASOLINE AND KEROSENE

PETROLEUM NAPHTHA is a generic term applied to refined, partly refined or unrefined petroleum products, and liquid products of natural gas, not less than 10% of which distills below 347° F., and not less than 95% of which distills below 464° F., when subjected to distillation in accordance with the A.S.T.M. standard method. (A.S.T.M. designation D86.)

Gasoline is defined (A.S.T.M. designation D288-31T) as a refined petroleum naphtha which by its composition is suitable for use as a carburant in internal combustion engines. For gasoline specifications see Federal Specification Board requirements. Also see Edgar, Boyd and Hill, The Meaning of the Gasoline Distillation Curve, *Trans.* A.P.I., 1930.

While gasoline and kerosene are not entirely uniform in make-up, there is substantial agreement among producers and consumers as to their general characteristics.

The fuel-air ratio for gasoline and kerosene varies between 14 and 15 lb. of air per pound of fuel.

Commercial Gasoline is a mixture of liquid hydrocarbons distilling between about 100° and 400° F. It usually is a composite of: *a*. Straight run gasoline, *i.e.*, the portion of crude oil boiling up to 400° F.; *b*. Cracked gasoline, *i.e.*, the same portion of the product made by thermally decomposing heavy oils; *c*. Casinghead or natural gasoline, *i.e.*, the liquid hydrocarbons carried as vapor in natural gas. The sulphur content usually is limited by specification to 0.1%, and the proportion of elements other than carbon and hydrogen is negligibly small. The elementary composition by weight is, in general, not far from carbon, 85.7%, hydrogen, 14.3%. The refined product on the market usually is free of water, acid compounds, gum or other deleterious constituents.

Gasoline ordinarily is graded by volatility and anti-knock value (octane number; see p. 14–68) into motor, premium and aviation grades, the latter being used in airplanes. Typical samples of these grades may have characteristics given in Table 17.

Typical significant properties of gasoline are: Volume as vapor at 32° F. and 760 mm., 3.6 cu. ft. per lb., 22.3 cu. ft. per gal.; air required for combustion of 1 lb. of gasoline, 15.3 lb.; calorific value = 18,254 + (39.6 × deg. A.P.I.) = 20,600 B.t.u. (approx.); vapor pressure at 100° F. (Reid method, A.S.T.M. D323-32T), 7–8 lb. (summer), 9–10 lb. (winter); surface tension against air, 25–28 dyne-cm.;

$$\text{specific heat} = \{(t^\circ\text{ F.} + 670)(2.10 - \text{sp. gr.})\}/2030 = 0.52 \text{ (approx.)};$$

heat of vaporization (approx.), 130 B.t.u. per lb., 150 for light fractions, 100 for heavy ends; electrical conductivity, (2 to 4) $\times 10^{-14}$; coefficient of thermal expansion per deg. F., 0.0006–0.0007 (see Bur. Stds. Circular 154 for tables of calculated results).

Table 17.—Characteristics of Typical Gasolines

	Motor	Premium	Aviation
Distillation,* per cent evaporated			
Initial Boiling Point, I. B. P....	100° F.	85° F.	125° F.
10......................	148° F.	125° F.	165° F.
50......................	276° F.	235° F.	200° F.
90......................	358° F.	350° F.	250° F.
End Point, E. P............	400° F.	385° F.	310° F.
Specific gravity †...............	0.753	0.732	0.716
Gravity, deg. A. P. I............	56.4	61.8	66.0
Octane number ‡....................	69 §	76 §	75 ¶

* A.S.T.M. Specification D86–30. † The specific gravity will be heavier (as much as 0.77) for products made from naphthenic or aromatic crude oils. ‡ A.S.T.M. Specification D357–34T. § With lead tetraethyl. ¶ Up to 85–90 with lead tetraethyl.

EXPLOSIVE MIXTURES OF GASOLINE.—Mixtures of air and gasoline vapor containing from 1.5 to 2.5% of gasoline are explosive. (Tech. Paper 115, U. S. Bureau of Mines.)

SPECIFIC VOLUMES of petroleum products completely vaporized at one atmosphere pressure are given in publication No. 97 of the U. S. Bureau of Standards. Table 18 gives values at 60° F. for products having a specific gravity range of 0.6690 to 0.7796 to include both gasolines and kerosene.

Table 18.—Specific Volume of Petroleum Products at 60° F. and 1 Atmosphere Pressure
(U. S. Bureau of Standards)

Gravity, A. P. I.	50	55	60	65	70	75	80
" specific	0.7796	0.7587	0.7389	0.7201	0.7022	0.6852	0.6690
Vapor, cu. ft. per lb.	2.8	3.1	3.4	3.7	4.0	4.3	4.6

Petroleum Naphthas for Special Purposes, as dry cleaning and paint thinning, usually are made up of materials corresponding to the higher-boiling half of gasoline. The boiling range of dry-cleaning solvent may be from 250° or 280° F. to 400° F., and the flash point is around 100° F. The physical properties correspond to those of gasoline, making allowance for the shift in molecular weight.

KEROSENE is defined (A.S.T.M. designation D288–31T) as a refined petroleum distillate having a flash point not below 73° F. as determined by the Tag closed tester (see A.S.T.M. designation D56), and suitable for use when burned in a wick lamp.

Kerosene is a higher boiling product than gasoline or the special naphthas. The significant physical properties of a typical product are, in general, as follows: Specific gravity, 0.8155; A.P.I. gravity, 60° F., 42.0; flash point, closed tester, 132° F.; coefficient of thermal expansion, per deg. F., 0.0005; viscosity, centipoises, 1.9; heat of vaporization, 85 B.t.u. per lb.; surface tension, 30 dyne-cm.; assay distillation: overpoint, 347° F.; 10%, 381° F.; 50%, 436° F.; 90%, 500° F.; end, 539° F.

Specifications for Kerosene for general illuminating purposes for U. S. Government are given in Tech. Paper 323, U. S. Bureau of Mines as: Flash point, min. 100° F.; flock test, negative; sulphur, max., 0.125%; distillation end point, max., 625° F.; cloud point, negative at 5° F.; the oil to be free from water, glue and suspended matter, and to burn readily and freely for 16 hr.

Kerosene Distillates are used to some extent as fuel oils for small furnace installations. The product employed for lamps and wick burners must be much more highly purified; the extent of purification and suitability are verified by long-time burning tests.

ALCOHOL.—Denatured alcohol is a grain or ethyl alcohol, C_2H_5OH, mixed with a denaturant in order to make it unfit for beverage or medicinal purposes. The various governments provide definite formulas for completely denatured alcohols. For information refer to U. S. Internal Revenue Department, or other authorities having jurisdiction. The denaturants generally used in varying proportions are methyl or wood alcohol (CH_4O),

Table 19.—Specific Gravity of Ethyl Alcohol at 60° F. Compared with Water at 60° F.
(Smithsonian Tables)

Sp. Gr.	Percent Alcohol		Sp. Gr.	Percent Alcohol		Sp. Gr.	Percent Alcohol	
	Weight	Volume		Weight	Volume		Weight	Volume
0.834	85.8	90.0	0.826	88.9	92.3	0.818	91.9	94.5
.832	86.6	90.6	.824	89.6	92.9	.816	92.6	95.0
.830	87.4	91.2	.822	90.4	93.4	.814	93.3	95.5
.828	88.1	91.8	.820	91.1	94.0	.812	94.0	96.0

Table 20.—Calorific Value of Denatured Alcohol
(Bulletin No. 43, U. S. Bureau of Mines)

Specific Gravity at 60° F.	Percent by Weight				Calorific Value, B.t.u. per lb.	
	Water	H	C	O	High	Low
0.8181	8.3	12.64	47.17	40.19	11,628	10,530
.8188	8.5	12.74	47.47	39.79	11,750	10,643
.8191	8.7	12.73	47.28	39.99	11,612	10,508
.8192	8.7	12.75	47.30	39.95	11,592	10,485
.8196	8.9	12.75	47.02	40.23	11,605	10,498
.8198	8.9	12.73	46.92	40.35	11,547	10,440
.8206	9.2	12.71	47.72	39.57	11,561	10,447
.8225	9.9	12.60	45.97	41.43	11,473	10,379
.8241	10.5	12.47	46.87	40.66	11,479	10,395

benzol and pyridin. The alcohol which it is proposed to manufacture under the present law (1935) is ethyl alcohol. This material seldom, if ever, is obtained pure, it being generally diluted with water and containing other alcohols when used for engines.

The products of complete combustion of alcohol are H_2O and CO_2. Under certain conditions, with an insufficient supply of air, acetic acid is formed, causing rusting of the parts of an alcohol engine. This may be prevented by addition of benzol to the alcohol.

SHALE OIL is obtained by the destructive distillation of oil-shale. This is a compact rock of sedimentary origin, with an ash content of more than 33%. It contains organic matter yielding oil when destructively distilled, but not appreciably when extracted with the ordinary solvents for petroleum. At present (1935) shale oil is not important as a commercial product.

COAL TAR AND TAR OIL. (Industrial Furnaces, W. Trinks, John Wiley & Sons, N.Y.).—Coal tar is a product of the destructive distillation of bituminous coal carried out at high temperature. A typical composition of tar is: C, 86.7%; H, 6.0%; N, 0.1%; S, 0.8%; O, 3.1%; ash, 0.1%; water, 3.2%. The black color is due to free carbon in suspension (about 4%). The high heating value equals 16,340 B.t.u. per pound. The viscosity is about 140 Saybolt seconds at 140° F. Coal tar weighs 9.5 lb. per gal. The above analysis shows tar to have almost the same chemical composition as the combustible matter of the coal from which it is made. Tar principally is used in reheating and open-hearth furnaces of steel works. It is not easily obtainable in the open market. Since it is a by-product, its cost price is more or less arbitrary.

Liquefied Petroleum Gas as an industrial fuel for heating furnaces and similar purposes is described by R. W. Thomas (Trans. A.S.M.E., FSP-52-21, 1930). Such gas is the vapor of propane or butane, which are obtained in the cracking of oil for motor fuel and other products. The physical properties of commercial grades of the two gases are given in Table 21.

Table 21.—Properties of Propane and Butane.

	Vapor Pressure, lb. per sq. in. at			Specific Gravity		Cu. Ft. of Gas		Boiling	Heating Value, B.t.u.		
	70° F.	100° F.	130° F.	Liquid	Gas	Per lb.	Per gal.	Point, deg. F.	Per lb.	Per gal.	Per cu. ft.
Propane..	120	195	300	0.509	1.523	8.49	36	—45	21,650	91,800	2550
Butane...	33	65	110	0.576	1.95	6.7	32	12	21,500	102,400	3200

Commercial butane is less volatile than propane and is better adapted to industrial use. It will evaporate at temperatures down to 32° F. when supplied with its latent heat of evaporation, 170 B.t.u. per lb. Ordinarily it is vaporized in a multi-tubular vaporizer, by hot or cold water or steam. The vapor may be piped any distance at moderate pressure if it is not cooled below the dew point, 26° F. It is burned in burners that either premix air and vapor or which draw the vapor in by flow of the air supply. The advantages of this fuel are ease of temperature regulation and furnace atmosphere control, the absence of sulphur compounds, dust and water in the furnace, together with smaller pipe sizes than are required for manufactured or natural gas. It is competitive with manufactured gas for practically all uses but not with natural gas.

Vegetable Oils as Internal Combustion Engine Fuel. (*Engineer*, Mar. 11, 1921.)—Tests of internal combustion engines using palm oil or cotton seed oil showed the following fuel consumption: 265 grams per cheval-heure (1 cheval-heure = 0.9863 Hp.-hr.) in a 4-cycle, 25-Hp. engine; 320 grams per cheval-heure in a 2-cycle, 168 Hp. engine; 285 grams per cheval-heure in a 2-cycle, 33-Hp. engine. The calorific value of palm oil and cotton seed oil averaged 9350 calories per kg. Thermal efficiencies were 25.9, 21.45 and 24.1%.

GASEOUS FUELS *

The gaseous fuels commonly used in industry are producer gas, blast furnace gas, coke oven gas, acetylene and hydrogen. For heats of combustion of gases, see p. 4-05.

PRODUCER Gas.—See p. 13-03.

ILLUMINATING Gas.—See p. 4-67.

GAS ANALYSES BY VOLUME AND WEIGHT.—To convert an analysis of a mixed gas by volume into analysis by weight: Multiply the percentage of constituent gas by its relative density, *viz;* CO_2 by 11, O by 8, CO and N each by 7, and divide each product by the sum of the products. Conversely, to convert analysis by weight into analysis by volume, divide the percentage by weight of each gas by its relative density, and divide each quotient by the sum of the quotients.

EXPLOSIVE MIXTURES OF GASES (Tech. Paper 39, U. S. Bureau of Mines.)—Mixtures of methane and air containing less than 5.5% or more than 12.8% methane

* Staff Revision.

will not explode. The explosive power is greatest when the proportion of methane is about 9.5%. The limit of inflammability is about 30% methane. The mixture between this proportion and explosibility burns quietly. A mixture of one volume methane and 955 volumes of dry air will ignite at 1200° F. The pressure developed by this mixture ignited in a closed space is 102.5 lb. per sq. in

NATURAL GAS.—The principal constituent of natural gas is methane CH_4, but the composition and calorific value varies between wide limits, depending on the locality. See Table 1.

Table 1.—Analyses of Natural Gas Collected in 31 Cities in the U. S.

(Tech. Paper 158, U. S. Bureau of Mines)

Town			Ethar C_2H	Carbon Dioxide, CO_2	Nit N	Calculated Gross Heating Value, B.t.u. per cu. ft. (760 mm. pressure) 0° C.	60° F.	Calculated S Gravity (ai
Fayette, Ala...........	97.6	97.6	0.0	0.3	2.10	1039	983	0.57
Alma, Ark.............	99.2	99.2	0.0	0.20	0.6	1057	1000	.56
Little Rock, Ark.......	96.7	96.7	0.0	1.00	2.3	1030	974	.57
Los Angeles, Cal.......	93.5	77.5	16.0	6.50	0.0	1123	1062	.70
Olney, Ill.............	97.1	37.5	59.6	0.0	1.7*	1591	1505	.86
Palestine, Ill.........	95.6	95.6	0.0	0.5	3.9	1018	963	.58
Geneva, Ind...........	98.8	75.4	23.4	0.0	1.2	1238	1171	.68
Coffeyville, Kans......	98.0	98.0	0.0	1.2	0.8	1044	988	.57
Pittsburg, Kans.......	93.0	90.5	2.5	0.4	6.6	1010	955	.60
Ashland, Ky...........	79.0	75.0	24.0	.0	1.0	1245	1178	.68
Lexington, Ky.........	99.0	76.4	22.6	.0	1.0	1234	1167	.67
Kansas City, Mo......	90.8	84.1	6.7	.8	8.4	1025	965	.63
Elmira, N. Y..........	99.0	84.0	15.0	.0	1.0	1174	1111	.63
Bolivar, N. Y..........	97.4	59.8	37.6	.4	2.2	1336	1264	.75
Buffalo, N. Y..........	99.6	88.1	11.5	.0	0.4	1152	1090	.61
Pavilion, N. Y.........	98.7	91.9	6.8	.0	1.3	1105	1045	.59
Wellsville, N. Y.......	98.0	78.1	19.9	.0	2.0	1202	1137	.65
Ashtabula, O..........	98.7	82.2	16.5	.0	1.3	1182	1118	.65
Lima, O..............	96.3	83.5	12.8	.0	3.7	1127	1066	.63
Piqua, O..............	90.9	78.3	12.6	.2	8.9	1068	1010	.66
Sandusky, O...........	96.0	83.5	12.5	.2	3.8	1122	1061	.63
Utica, O..............	93.9	74.8	19.1	.3	5.8	1152	1090	.68
Guthrie, Okla..........	91.9	73.5	18.4	.0	8.1	1125	1064	.68
Sapulpa, Okla.........	98.8	93.1	5.7	.4	0.8	1098	1039	.59
Altoona, Pa...........	99.0	90.0	9.0	.2	0.8	1126	1065	.60
Oil City, Pa..........	97.7	64.3	33.4	.0	3.3	1306	1235	.74
St. Marys, Pa.........	99.2	88.0	11.2	.0	0.8	1146	1084	.61
Sharon, Pa............	99.3	32.3	67.0	.0	.7	1591	1505	.89
Charleston, W. Va.....	99.3	76.8	22.5	.0	.7	1236	1169	.67
Clarksburg, W. Va.....	99.3	66.6	32.7	.0	.7	1318	1247	.72
Fairmont, W. Va	99.0	82.0	17.0		.9	1189	1125	.64

* Contained also 1.2% hydrogen sulphide (H_2S).

BLAST-FURNACE GAS AND COKE-OVEN GAS are given off in the operations of blast furnaces and coke ovens. Typical analyses of these gases (F. E. Leahy, *Yearbook*, Am. I. & S. Inst., 1929) are as follows:

	CO_2	O_2	CO	H_2				H_2S	per cu. ft.
Blast-furnace gas....	12.4	0.0	26.6	2.2	0.4	58.4	...	...	97.1
Coke-oven gas......	1.4	0.2	5.9	57.3	26.9	4.7	3.0	0.6	537.0

These gases, in steel plant operation, are mixed to give a gas of the desired heat content for various operations. Table 2 gives the heat content of various mixtures.

Leahy gives the heat requirements of gas for the various furnaces, B.t.u. per cu. ft., as follows: Billet-heating, 350; box-anneal, 200; coke ovens, 100–250; open-hearth, 250–350; sheet and pair, 250; soaking pits, 150; tin pots, 250.

A mixture of coke-oven and blast-furnace gas often can be substituted profitably for producer gas or other fuels in steel plants. According to Leahy, the order of replacement, in general, should be first the lower-temperature furnaces, excepting welding, melting and

Table 2.—Heat Content of Mixtures of Blast-furnace and Coke-oven Gas

Percentages by Volume

Blast-furnace Gas, percent	Coke-oven Gas, percent	B.t.u. per cu. ft.	Blast-furnace Gas, percent	Coke-oven Gas, percent	B.t.u. per cu. ft.	Blast-furnace Gas, percent	Coke-oven Gas, percent	B.t.u. per cu. ft.	Blast-furnace Gas, percent	Coke-oven Gas, percent	B.t.u. per cu. ft.
95	5	119.1	70	30	229.1	45	55	339.1	20	80	449.1
90	10	141.1	65	35	251.1	40	60	361.1	15	85	471.1
85	15	163.1	60	40	273.1	35	65	383.1	10	90	493.1
80	20	185.1	55	45	295.1	30	70	405.1	5	95	515.1
75	25	207.1	50	50	317.1	25	75	427.1			

open-hearth, and the higher-temperature furnaces last. If only the fuel replacement value is considered, especially if the gas producer plant cannot be operated economically, the open-hearth should be supplied first with the mixed gas. In determining desirability of substituting mixed gas for present fuel, the replacement value of the latter should be measured against its heat value, by a study of present cost against expected cost. An example of such a study is given in Leahy's paper above cited. B. F. Entwisle, in a discussion of this paper, states that as blast-furnace gas is produced uniformly, it is best burned in furnaces that require fuel regularly and uniformly. Purchased fuels should be replaced by blast-furnace and coke-oven gas in the order of their cost, the most expensive first. Based on these considerations, the order of use of the mixed gas would be: 1. Base steam load, including blower and power load of blast furnaces and coke ovens (Sunday load only). 2. Coke oven heating. 3. Open-hearth furnaces. 4. Soaking pits. 5. Reheating furnaces. 6. Week-day steam and power load.

Leahy states that open-hearth furnaces using liquid fuel, natural gas or coke-oven gas require alteration if they are to use a mixture of blast-furnace and coke-oven gas. The checkers must be sealed to prevent leakage through walls and roof. Preheating the blast-furnace gas to 1200 deg. F. and enriching it with coke-oven gas may eliminate necessity of sealing. Preheating of the gas gives a higher flame temperature than preheating the air to the same temperature. Blast-furnace gas is non-luminous, and must be combined with other fuels to be given luminosity.

C. R. Meissner (*Yearbook* Am. I. & S. Inst., 1932) describes an extremely flexible arrangement for the utilization of blast-furnace gas and coke-oven gas at the plant of the Weirton Steel Co. Of 198,000,000 cu. ft. of blast-furnace gas produced daily, 55,000,000 cu. ft. are used in the blast-furnace stoves, 43,000,000 cu. ft. are used for under-firing the by-product coke ovens, the balance for general metallurgical furnaces. Of 26,300,000 cu. ft. of coke-oven gas, produced by 111 ovens, 16,300,000 cu. ft. are available for mill use when the coke-oven gas is used for under-firing the coke ovens. When blast-furnace gas is used under coke ovens, 23,000,000 cu. ft. of coke-oven gas are available for mill use. The heat consumption per pound of coal coked was: With coke-oven gas, 1101 B.t.u.; with blast-furnace gas, 1118 B.t.u. The blast-furnace gas gave a more uniform quality of coke.

A steam plant using blast-furnace gas at the Fairfield Works of the Tennessee Coal, Iron and R.R.Co. developed a boiler efficiency of 83% (F. G. Cutler, *Trans.* A.S M.E., FSP–54–2, 1932). The gas is cleaned in 11 Cottrell-type electric precipitators, which reduce the dust content (grams per cu. ft.) from 3.04, leaving dust-catcher, to 0.02, entering stoves, and 0.54 entering boiler furnaces. The average temperature of the gas entering the boiler was 470° F. The boilers used powdered coal as an auxiliary fuel.

The following average data are condensed from a table in the above paper:

B.t.u. per hr. in top gas, 739,150,000; gross boiler Hp. generated, 14,324; powdered coal burned per 24 hr., 31 tons; B.t.u. per hr. to steam at 82% efficiency, 572,980,000; B.t u. per hr. in powdered coal, 31,200,000; B.t.u. per hr. in blast furnace gas, 541,780,000; percent of top gas to boilers, 73.34; steam pressure, 324 lb. per sq. in.; superheat, 189° F.; boiler Hp. in service, 4383; percent of rating, 326. Surplus power generated in the plant is sold to the Alabama Power Co.

ACETYLENE, C_2H_2, consists of 92.3% carbon and 7.7% hydrogen It is a colorless, tasteless gas, of specific gravity 0.92. The commercial gas contains a small percentage of phosphoretted and sulphuretted hydrogen. It becomes liquid at a pressure of 700 lb. per sq. in. at 70° F. The pressure necessary for liquefaction varies directly with temperature, up to 98° F., which is the critical temperature. It cannot be liquified by any pressure at temperatures in excess of 98° F.

The heat of combustion of acetylene is 1475 B.t.u. per cu. ft. at 70° F. and 30 in. of mercury. Of this, 227 B.t.u. are endothermic. For complete combustion, 1 cu. ft. requires 11.91 cu. ft. of air. Its ignition temperature in air is 804° F., and in oxygen, 782° F.

Mixtures of air and acetylene in the proportions of from acetylene 3%, air 97% to acetylene 24%, air 76%, are explosive. The point of maximum explosibility is acetylene 7.7%, air 92.3%. Acetylene will explode of itself when ignited under a pressure of 15 lb. per sq. in. Danger of explosion is removed by absorbing the acetylene in acetone, which is itself absorbed by asbestos, kieselguhr or other non-inflammable substance.

Acetylene is produced by the combination of calcium carbide and water, in the proportions of 64 lb. CaC_2 and 36 lb. H_2O, to produce 26 lb. of acetylene. The chemical reaction is $CaC_2 + 2\,H_2O = C_2H_2 + Ca(OH)_2$. Calcium carbide is produced in the electric arc furnace from a mixture of finely divided and intimately mixed calcium oxide, or quicklime, and coke. The chemical reaction is $CaO + 3\,C = CaC_2 + CO$.

IGNITION TEMPERATURES OF GASES vary with the conditions under which they are burned. Table 3, compiled from the International Critical Tables, give ignition temperatures under several different conditions.

Table 3.—Ignition Temperatures of Gases

At 1 Atmosphere Pressure

Gas	Burned with	Method. See Note	Ignition Temperature, deg. F.	Gas	Burned with	Method. See Note	Ignition Temperature, deg. F.
Hydrogen				Propane			
H_2	O_2	A	1121	C_3H_8	O_2	D	1013–1018
H_2	O_2	C	1092	C_3H_8	O_2	E	914–1058
15% H	Air	A	1148	C_3H_8(1.25%)	Air	B †	1090
60% H	Air	A	1313	C_3H_8(2.5%)	Air	B †	1026
Hydrogen Sulphide				C_3H_8(6.5%)	Air	B †	961
$2(H_2S)$	$3O_2$	A	599–608	*n*-Butane			
H_2S	$3O_2$	C	482–518	C_4H_{10}(1.25%)	Air	B †	1049
H_2S	Air	E	655–714	C_4H_{10}(2%)	Air	B †	1013
Ammonia				C_4H_{10}(3.65%)	Air	B †	959
NH_3	O_2	A	1436	C_4H_{10}(7.65%)	Air	B †	905
NH_3	Air	E	1292–1580	Isobutane			
Carbon Monoxide				C_4H_{10}	O_2	D	1013–1022
CO	$2(CO+O_2)$	B	1193–1202	*n*-Pentane			
CO	O_2	B	1202–1256	C_5H_{12}(1.5%)	Air	B †	1018
CO	Air	E	1337	C_5H_{12}(2.75%)	Air	B †	968
Methane				C_5H_{12}(7.65%)	Air	B †	889
CH_4	$2O_2$	A	1202–1346	C_5H_{12}(6.7%)	Air	H	608–637
CH_4	$2O_2$	B	1112–1202	Benzene			
CH_4	$2O_2$	C	1121–1202	C_6H_6(5%)	Air	B	1089
CH_4	$2O_2$	D	1213–1252	C_6H_6	O_2	J	1051
CH_4(10% by vol.)	Air	B	1346–1454	C_6H_6	Air	J	915
Acetylene				*n*-Hexane			
C_2H_2(45–55%)	Air	B	635	C_6H_{14}(6.7%)	Air	H	572–583
C_2H_2(20%)	Air	B	752	*n*-Heptane			
C_2H_2(10%)	Air	B	932	C_7H_{16}(6.7%)	Air	H	545
C_2H_2	O_2	E	752–824	C_7H_{16}(5%)	Air	H	536
C_2H_2	Air	E	763–824	*n*-Octane			
Ethylene				C_8H_{18}(6.7%)	Air	H	527–536
C_2H_4	$3O_2$	C	986–1121	Ether			
C_2H_4	O_2	E	932–1006	$C_4H_{10}O$	Air	A	374
C_2H_4(4.5–6.5%)	Air	B*	909	$C_4H_{10}O$	Air	B	365–379
C_2H_4	Air	E	1008–1017	$C_4H_{10}O$(4.8%)	Air	B	352–363
Ethane				$C_4H_{10}O$(6.6%)	Air	H	414
C_2H_6(1.9%)	Air	B †	1101	$C_4H_{10}O$	O_2	J	374
C_2H_6(4.85%)	Air	B †	1031	C_2S	O_2	E	225–313
C_2H_6(10.6%)	Air	B †	993	C_2N_2	O_2	E	1477–1504

* Volume of vessel, 275 c.c. † Volume of vessel, 85 c.c.

The methods used in determining the temperatures given in the table are as follows: A. Mixture passed through tube held at known temperature. B. Mixture rapidly admitted to bulb held at known temperature. C Bulb containing mixture rapidly heated to definite temperature. D. Mixture passed through small reservoir while temperature was raised until flame at exit tube ran back into it. E. Constituent gases heated separately in concentric tubes, gas from inner tube then being passed into gas in outer tube. F. Constituent gases heated separately and mixed in open. G. Mixture adiabatically compressed, and temperature calculated from final volume. H. Mixture adiabatically compressed, and temperature calculation based on final pressure, experimentally determined. Correct temperature lies between values calculated by G and H. J. Small drop of inflammable liquid dropped into air or O_2 at known temperature.

ILLUMINATING GAS

By Alfred E. Forstall

1. COAL-GAS AND WATER-GAS

COAL-GAS is made by distilling bituminous coal in either retorts or by-product coke ovens. Retorts are made of fire-clay or silica material, and are set either horizontally or vertically.

Horizontal retorts are usually long, semi-cylindrical, or ⌂-shaped, chambers and are either stop-end, *i.e.*, one end permanently closed, the other fitted with a cast-iron mouthpiece and lid; or throughs, with both ends open except for mouthpieces and lids. Internal cross-section varies from 14 × 24 in. to 16 × 28 in. Stop-end retorts usually are about 9 ft. long and hold 250 to 400 lb. of coal per charge of from 4 to 6 hr. duration. Through retorts are 11 to 22 ft. long and hold 400 to 1000 lb. per charge of from 4 to 12 hr. Retorts of either type are set in groups of 6 to 12. Such a group is called a bench, and is heated by a generator furnace or gas producer, usually using coke as fuel.

Vertical retorts are either rectangular or elliptical chambers; the internal cross-section at the top ranges from 33 × 10 in. to 103 × 10 in.; at the bottom it widens out to provide for expansion of the coal as it carbonizes in passing down through the retort. Length usually is 25 ft. Retorts are set in benches of from 2 to 8 retorts. Vertical retorts may be continuous or intermittent. Intermittent retorts are filled with from 2000 to 2400 lb. of coal and emptied of coke at regular intervals of 12 hr. In continuous retorts coal is fed at the top and coke discharged at the bottom continuously. Intermittent retorts have mouthpieces and lids at both ends. Continuous retorts have a coal hopper at the top and a coke chamber at the bottom.

By-product Coke Ovens are rectangular chambers, 13 ft., 6 in. to 42 ft. long, 8 to 13 ft. high, 12 to 18 in. wide. They carbonize charges of 4000 to 30,000 lb. of coke in 12 to 18 hr.

The volatile matter driven off when coal is exposed to heat consists of a mixture of fixed gases and vapors. During their passage to the outlet of the retort or oven some of these vapors are converted into fixed gases either by contact with the highly-heated walls of the retort or by exposure to heat radiated from the walls. For horizontal retorts or ovens this outlet consists of stand, bridge and dip pipes. The lower end of the latter is sealed in ammoniacal liquor in a horizontal pipe called a hydraulic main. From vertical retorts the gas is taken off at the upper end through a short pipe dipping into either a hydraulic, or a dry main. Coke ovens have take-off pipes at each end.

Cooling of the gas begins in this main, with a consequent condensation of a portion of the steam and hydrocarbon vapors to water and tar, respectively. The water, in forming, absorbs ammonia, which is present as a gas, and becomes ammoniacal liquor. Part of the tar is deposited in the hydraulic main, but the rest remains in the gas as a fog. To remove this the gas is passed through a frictional tar extractor, before its temperature is lowered below 100° F., and then goes to a condenser for further cooling. The condenser is a vessel traversed by iron tubes through which water flows, causing the deposition as light tar and ammoniacal liquor, respectively, of the excess of condensible hydrocarbon vapors and steam. The cooled gas passes through a washer, where it bubbles through water or ammoniacal liquor to remove the remaining light oils and a part of the ammonia and sulphuretted hydrogen, and through a scrubber, in which it travels in thin streams over wetted surfaces, where removal of the ammonia is completed. The gas then passes into the purifiers where the sulphuretted hydrogen is removed by means of hydrated ferric oxide. The gas then is measured and passed into storage holders.

COAL USED FOR GAS MANUFACTURE.—Coal generally used in gas making is caking-bituminous. When high illuminating value was important, it was necessary to enrich the gas produced from this coal by adding to it gas made from cannel coal, or carburetted water-gas. At the present (1935), illuminating value is not considered important in town gas, and coal-gas without any enrichment meets the ordinary requirements of Public Service Commissions.

Table 1 shows proximate analyses of various coals used for the manufacture of gas in the United States.

Table 1.—Proximate Analysis of Various Gas Coals Used in the U. S.

Name of Coal	Moisture	Volatile Matter	Fixed Carbon	Ash	Sulphur
	Percent	Percent	Percent	Percent	Percent
Pennsylvania-Westmoreland	0.93–1.51	33.02–36.33	56.09–58.87	5.99–9.07	0.76–1.81
West Virginia	0.80–1.35	36.92–38.60	53.70–55.36	6.37–6.90	0.90–1.40
Kentucky	1.17–4.25	35.60–36.89	56.84–60.30	2.02–2.93	0.53–1.74
Ohio-Hocking Valley	6.72	37.13	50.32	5.83	1.67
Oklahoma	3.87	35.73	50.05	10.35	1.99

THE PRODUCTS OF DISTILLATION from 100 lb. of average gas-coal vary according to quality of coal and temperature of distillation, but are about as follows: Coke 64 to 70 lb.; gas 15 to 19 lb.; tar 5 to 6 lb.; virgin liquor and ammonia 7 to 9 lb.; impurities and loss 3 to 5 lb.

WATER-GAS is obtained by passing steam through a bed of carbonaceous fuel heated to a temperature of at least 1800° F. The steam is decomposed, its oxygen being seized by the hot carbon, forming carbon monoxide and a small amount of carbon dioxide, and liberating hydrogen. The chemical reaction is $C + H_2O = CO + H_2$, or $2C + 2H_2O = C + CO_2 + 2H_2$, followed by a splitting up of the CO_2 as it comes into contact with more carbon. By weight, the theoretical gas, $CO + H_2$, consists of $C + O + H_2 = 28$ parts CO and 2 parts H_2, or 93.33% CO and 6.67% H_2. By volume, it consists of equal parts of CO and H. This theoretical gas never is made in practice and commercial water-gas always contains from 3% to 5% of CO_2, a small percentage of N, and a trace of CH_4. Water-gas, produced as above described, has a calorific value of about 300 B.t.u. per cu. ft., but no illuminating value. It may, however, be used for lighting by causing it to heat to incandescence some solid substance, as a Welsbach or other incandescent mantle.

CARBURETTED WATER-GAS.—An illuminating-gas, or a gas of higher calorific value, is made from water-gas by adding to it hydrocarbon gases, or vapors, which usually are obtained from the distillate of crude petroleum, known as gas-oil. With apparatus properly designed and operated, heavy fuel oil can be used as enricher.

A history of the development of carburetted water-gas processes, together with a description of the form of apparatus which at that time had been recently developed and which still remains the standard form, is given by Alex. C. Humphreys, in a paper Water-gas in the U. S. (Mech. Sec., British Assoc. for Advancement of Science, 1889).

The water-gas, or carburetted water-gas, process has two periods; (1) The "blow," during which air is blown through the fuel bed in the generator and the lean producer gas thus formed is completely burned in the carburetter and superheater, the other two vessels making up the generating apparatus, giving up a great portion of its heat to the firebrick checker-work contained in them, before passing out through a stack valve to the open air. (2) The "run," during which the air blast is stopped, the stack valve closed, and steam is passed through the incandescent bed of fuel. The resulting water-gas passes into the top of the carburetter where it meets a spray of a distillate of petroleum, as gas-oil, which is vaporized and carried by the water-gas through the carburetter and superheater, where the hydrocarbon vapors become converted into fixed illuminating-gases. From the superheater the combined gases are passed into a relief holder, which compensates for the intermittent nature of the process. From this holder it passes through condensers, tar extractors, purifiers, etc., into storage holders. No ammonia is formed, and no apparatus for its removal is required.

The specific gravity of carburetted water-gas increases with increase of the hydrocarbon gases, which give increased calorific value. It will vary between 0.54 and 0.70. For further description of both coal-gas and carburetted water-gas apparatus and operation see Morgan's American Gas Practice.

ANALYSES OF COAL-GAS AND CARBURETTED WATER-GAS.—No two samples of either coal-gas or carburetted water-gas have exactly the same composition. Table 2 gives percentage compositions by volume and by weight, which may be considered typical for coal-gas made in retorts and in coke ovens, and for carburetted water-gas of calorific value of 560 B.t.u. per cu. ft.

Characteristics of Constituents.—In burning the gas the CO_2, O and N are inert, and are so classed. The olefines, or illuminants, give the gas any illuminating power it may

Table 2.—Typical Composition of Coal-gas and Carburetted Water-gas

Constituent	Coal-gas				Carburetted Water-gas	
	Retorts		Coke Ovens			
	Volume	Weight	Volume	Weight	Volume	Weight
Carbon dioxide, CO_2	1.5	5.6	1.2	5.0	2.6	6.1
Olefines—Illuminants, C_nH_{2n}	2.2	9.7	2.4	9.3	10.1	18.6
Oxygen, O_2	0.3	0.9	0.9	2.9	0.9	1.6
Carbon monoxide, CO	8.6	19.7	5.6	14.6	30.8	46.3
Methane, CH_4	31.4	47.2	29.0	43.9	11.0	9.6
Ethane, C_2H_6	......	......	0.4	1.1	0.9	1.4
Hydrogen, H_2	52.5	8.8	55.7	10.7	35.4	3.9
Nitrogen, N_2	3.5	8.1	4.8	12.5	8.3	12.5
Total	100.0	100.0	100.0	100.0	100.0	100.0
Specific gravity	0.42		0.37		0.64	
Calorific value, B.t.u. per cu. ft.	575		550		560	

have and contribute materially to its calorific value, as do also the CO, CH_4, C_2H_6, and H. Since the calorific value of blue water-gas is only about 300 B.t.u. per cu. ft., almost half the calorific value of carburetted water-gas comes from hydrocarbons added as oil gas.

CALORIFIC VALUE STANDARDS.—Neither coal-gas nor carburetted water-gas now is used in general practice for purposes in which its inherent illuminating value is important. Regulations for quality of gas that must be supplied deal only with its calorific value. The first calorific value standards, established in the U. S., called for gas of 600 B.t.u. per cu. ft., but at the present time (1935) the customary requirement is from 525 to 535 B.t.u. per cu. ft. The standard in Canada is 450 B.t.u. per cu. ft.

EFFICIENCY OF A WATER-GAS PLANT.—The practical efficiency of an illuminating water-gas setting is discussed in a paper by A. G. Glasgow (*Proc.* Am. Gaslight Assoc., 1890) from which the following is abridged:

The results refer to 1000 cu. ft. of unpurified carburetted gas, reduced to 60° F. The total anthracite charged per 1000 cu. ft. of gas was 33.4 lb., ash and unconsumed coal removed, 9.9 lb., leaving total combustible consumed, 23.5 lb., which is taken to have a fuel value of 14,500 B.t.u. per lb., or a total of 340,750 B.t.u.

The heat-energy absorbed by the apparatus is $23.5 \times 14{,}500 = 340{,}750$ heat-units $= A$. Its disposition is as follows: B, the energy of the CO produced; C, the energy absorbed in the decomposition of the steam; D, the difference between the sensible heat of the escaping illuminating-gases and that of the entering oil; E, the heat carried off by the escaping blast products; F, the heat lost by radiation from the shells; G, the heat carried away from the shells by convection (air currents); H, the heat rendered latent in the gasification of the oil; J, the sensible heat in the ash and unconsumed coal recovered from the generator.

The heat equation is $A = B + C + D + E + F + G + H + J$; A being known. A comparison of the CO in Table 3 shows that (280/434); or 64.5% of the volume of carburetted gas, is pure water-gas, distributed thus: CO_2, 2.3%; CO, 28.0%; H, 33.4%; N, 0.8%; total = 64.5%. 1 lb. of CO at 60° F. = 13,531 cu. ft. CO per 1000 cu. ft. of gas $= 280 \div 13.531 = 20.694$ lb. Energy of the CO $= 20.694 \times 4395.6 = 91{,}043$ heat-units $= B$. 1 lb. of H at 60° F. = 189.2 cu. ft. H per M. of gas $= 334 \div 189.2 = 1.7653$ lb. Energy of the H per lb. (according to Thomsen, considering the steam generated by its combustion to be condensed to water at 75° F.) = 61,524 B.t.u. In Mr. Glasgow's experiments the steam entered the generator at 331° F.; heat required to raise products of combustion of 1 lb. of H, *viz.*, 8.98 lb. H_2O from water at 75° to steam at 331° must be deducted from Thomsen's figure, or $61{,}524 - (8.98 \times 1140.2) = 51{,}285$ B.t.u. per lb. of H. Energy of the H, then, is $1.7653 \times 51{,}285 = 90{,}533$ heat-units $= C$. Heat lost due to sensible heat in the illuminating-gases, (temperature 1450° F.), and that of the entering oil (235° F.), is 48.29 (weight) $\times$ 0.45786 (sp. heat) $\times$ 1215 (rise of temperature) = 26,864 heat-units $= D$. The specific heat of the entering oil is approximately that of the issuing gas. The heat carried off in 1000 cu. ft. of the escaping blast products is

$$86.592 \text{ (weight)} \times 0.23645 \text{ (sp. heat)} \times 1474° \text{ (rise of temp.)} = 30{,}180 \text{ heat-units.}$$

The temperature of the escaping blast gases is 1550° F., and that of the entering air 76° F. But the amount of blast gases, by registration of an anemometer, checked by a calculation from the analyses of the blast gases, was 2457 cu. ft. for every 1000 cu. ft. of carburetted gas made. Hence the heat carried off per M. of carburetted gas is $30{,}180 \times 2.457 = 74{,}152$ heat-units $= E$.

Table 3.—Operation of a Water-gas Plant

		Composition by Volume	Weight per 100 cu. ft.	Composition by Weight	Specific Heat
I. Carburetted water-gas	$CO_2 + H_2S$	3.8	0.465842	0.09647	0.02088
	C_nH_{2n}	14.6	1.139968	.23607	.08720
	CO	28.0	2.1868	.45285	.11226
	CH_4	17.0	0.75854	.15710	.09314
	H	35.6	.1991464	.04124	.14041
	N	1.0	.078596	.01627	.00397
		100.0	4.8288924	1.00000	0.45786
II. Uncarburetted gas	CO_2	3.5	0.429065	0.1019	0.02205
	CO	43.4	3.389540	.8051	.19958
	H	51.8	.289821	.0688	.23424
	N	1.3	.102175	.0242	.00591
		100.0	4.210601	1.0000	0.46178
III. Blast products escaping from superheater	CO_2	17.4	2.133066	0.2464	0.05342
	O	3.2	0.2856096	.0329	.00718
	N	79.4	6.2405224	.7207	.17585
		100.0	8.6591980	1.0000	0.23645
IV. Generator blast-gases	CO_2	9.7	1.189123	0.1436	0.031075
	CO	17.8	1.390180	.1680	.041647
	N	72.5	5.698210	.6884	.167970
		100.0	8.277513	1.0000	0.240692

Experiments made by a radiometer covering 4 sq. ft. of the shell of the apparatus gave figures for the amount of heat lost by radiation = 12,454 heat-units = F, and by convection = 15,696 heat-units = G.

The heat rendered latent by the gasification of the oil was found by taking the difference between all the heat fed into carbureter and superheater and total heat dissipated therefrom to be 12,841 heat-units = H. Sensible heat in ash and unconsumed coal is 9.9 lb. × 1500° × 0.25 (sp. heat) = 3712 heat-units = J.

$B + C + D + E + F + G + H + J$ = 327,295 heat-units, which subtracted from the 340,750 heat-units of the combustible consumed, leaves 13,455 heat-units, or 4% unaccounted for.

Of the total heat-energy of the coal consumed, or 340,750 heat-units, the energy wasted is the sum of items, D, E, F, G, and J, amounting to 132,878 heat-units, or 39%; the remainder, or 207,872 heat-units, or 61%, being utilized. Efficiency of the apparatus as a heat machine is 61%.

Five gallons, or 35 lb. of crude petroleum, were fed into the carbureter per 1000 cu. ft. of gas made; deducting 5 lb. of tar recovered, leaves 30 lb. × 20,000 = 600,000 heat-units as the net heating value of the petroleum used. Adding this to the heating value of the coal, 340,750 B.t.u., gives 940,750 heat-units, of which there is found as heat-energy in the carburetted gas, as in the table below, 764,050 heat-units, or 81%, which is the commercial efficiency of the apparatus, *i.e.*, ratio of energy contained in finished product to total energy of coal and oil consumed.

The heating power per M. cu. ft. of the carburetted gas is

C_3H_6*	146.0 × 0.117220 × 21222.0 =	363200
CO	280.0 × 0.078100 × 4395.6 =	96120
CH_4	170.0 × 0.044620 × 24021.0 =	182210
H	356.0 × 0.005594 × 61524.0 =	122520
		764050

The heating power per M. cu. ft. of the uncarburetted gas is

CO	434.0 × 0.078100 × 4395.6 =	148991
H	518.0 × 0.005594 × 61524.0 =	178277
		327268

* The heating value of the illuminants C_nH_{2n} is assumed to equal that of C_3H_6.

The candle-power of the gas is 31, or 6.2 candle-power per gallon of oil used. The calculated specific gravity is 0.6355, air being 1.

Since the efficiency of oil-gas production is greater than that of production of blue water-gas, the reduction in the percentage of oil-gas contained in carburetted water-gas as made at present as compared with that made in 1890, reduces the overall efficiency of the apparatus. On the other hand, the plant tested by Glasgow was not equipped, as such plants are now, with a waste heat boiler which recovers much heat otherwise carried off by escaping blast gases. Some English tests indicate an increase of efficiency of about 8.5% by use of waste heat boilers.

OIL-GAS.—Almost from the beginning of the business of manufacturing illuminating-gas hydrocarbon oils have been used for this purpose. These were originally the Scotch shale oils, and later, petroleum or some of its distillates, distilled in retorts similar to those used for the manufacture of coal-gas. This gas at first was employed as a substitute for and in competition with, other illuminating-gases. Later, for a number of years, it was used merely for the enrichment of coal-gas, the lighting of railway cars, etc., and the amount so used formed but a small proportion of the total amount of illuminating-gas made and distributed. In the last years of the 19th century the development of large oil fields caused petroleum to become in most of the Pacific Coast States, the cheapest available source of carbon and hydrocarbons. These conditions led to the development of a process, using crude petroleum or one of its heavier distillates as the sole raw material, for making a gas similar in composition to coal-gas but very different from the gas produced by the old method of distilling oil in retorts.

The form of apparatus most largely used for the manufacture of oil-gas consists of two cylindrical steel shells of equal diameter but different heights, lined with firebrick and filled to a greater or less extent with firebrick checker-work, similar to that used in the carburetter and superheater of a carburetted water-gas set. The shells are connected at the bottom by a throat piece. The heat required to gasify the oil is obtained by burning oil, sprayed into the apparatus by means of steam through sprays or burners entering the shorter vessel through the side, a short distance below its top. The oil is consumed by an air-blast entering the vessel through its top. A secondary air supply enters at the bottom of the taller vessel, and the products of combustion escape through a stack valve opening at the top of that vessel. When the apparatus has been heated to the proper gas-making temperature, that is, about 1800° to 2100° F., at the top of the shorter vessel, and about 1800° F. in the taller vessel, the blast is shut off, the stack valve closed, and oil is turned in through so-called gas-making burners at the top of the shorter vessel and either at the top or the bottom of the taller one, according to the type of apparatus. The " run " is continued until the temperature of the checker-brick has been reduced below that at which gas economically can be made. The oil then is shut off, and the apparatus purged with steam blown in through the oil burners.

Practically all of the illuminating-gas now (1935) made in the states of California and Oregon is made by this process, a more complete description of which can be found in Morgan's American Gas Practice.

CONSIDERATIONS AFFECTING THE CHOICE BETWEEN THE MANUFACTURE OF COAL-GAS OR CARBURETTED WATER-GAS.—Now that the illuminating value standard of quality has been abandoned and the calorific value standard can be met by coal-gas, the decision of the question of the kind of gas to be manufactured depends solely on conditions of prices of gas coal, coke, tar, and ammonia, anthracite coal, gas oil, and labor in any given locality. As a general proposition where there is a fair market for the residuals—coke, tar, and ammonia—of coal-gas manufacture, coal-gas can be produced at a lower material cost but with a higher labor cost than carburetted water-gas. The ground area required is much larger for coal-gas, and generally the investment in plant per 1000 cu. ft. of daily capacity is also greater for coal-gas. Before an intelligent decision can be reached as to the gas which can be most economically manufactured, it is necessary to study all the local conditions, and from this study determine the cost of each kind of gas, including both investment and operating charges.

The amount of gas to be made also has a great influence upon the relative economy of the two processes. Coal-gas must be manufactured continuously and, therefore, requires 24-hr. operation of the plant, while the operation of a carburetted water-gas plant can be suspended without detriment to the apparatus or undue increase in the cost of operation. For small plants it is, therefore, frequently more desirable to manufacture carburetted water-gas, making the total 24-hr. supply in a few hours, even under conditions where for manufacture on a large scale coal-gas would be more economical.

As far as generating apparatus only is concerned, Tables 4 and 5 show for continuous vertical retort coal-gas plants and for carburetted water-gas plants, respectively, the floor space required per 1000 cu. ft. of gas manufactured per day of 24 hr. The figures given for the carburetted water-gas plants include also that part of the condensing plant which would be immediately connected to the generating apparatus.

Table 4.—Space Required for Coal-gas Plant

Capacity of Plant, cu. ft.	Capacity of Unit, cu. ft.	Floor Space per 1000 cu. ft., sq. ft.	Capacity of Plant, cu. ft.	Capacity of Unit, cu. ft.	Floor Space per 1000 cu. ft., sq. ft.
500,000	125,000	4.00	2,500,000	312,500	2.60
750,000	125,000	3.75	3,000,000	312,500	2.60
1,000,000	250,000	3.25	3,750,000	375,000	2.50
1,500,000	250,000	3.00	4,500,000	375,000	2.50
2,000,000	250,000	2.75			

Table 5.—Space Required for Carburetted Water-gas Plant

Plant Capacity per Day, cu. ft.	Nearest Size Set	Capacity of Set per Day, cu. ft.	Ground Space One Set, sq. ft.	Square Feet per M. cu. ft. Generating Capacity	Ground Space, Spare Set, sq. ft.	Square Feet per M. cu. ft. Generating Capacity, spare set
200,000	4′ 0″	210,000	470	2.240	302	1.440
400,000	5′ 0″	480,000	630	1.310	490	1.020
800,000	6′ 0″	810,000	852	1.050	685	0.845
1,400,000	7′ 6″	1,470,000	1190	0.810	918	.625
2,000,000	8′ 6″	2,020,000	1465	.725	1160	.575
3,500,000	11′ 0″	3,500,000	2480	.710	1980	.566

In addition to the space required for the generating apparatus each type of plant requires space for additional condensing and for purifying, measuring and storage apparatus, as well as for the storage of manufacturing materials and residuals. The space required for the latter purposes is much larger in a coal-gas plant than in a carburetted water-gas plant.

FUEL VALUE OF ILLUMINATING-GAS.—The adoption of a calorific value standard of quality has made it important to have accurate means of determining the calorific value of illuminating-gas. This determination is made by a calorimeter, in which the heat of combustion of a measured quantity of gas is transferred to a measured quantity of water flowing at a constant rate through the calorimeter. The calorimeter most frequently used in the United States was originally designed by F. W. Hartley, but has been greatly modified and is now known as the Hinman–Junkers calorimeter. For a description of this calorimeter and the method of installing and operating it, see the Reports of the Committee on Calorimetry of the American Gas Institute (*Proc.* Am. Gas Inst.,

vol. iii, p. 285; vol. iv, p. 148), and also the Report of the Joint Committee on Calorimetry ... of the Second District. New York State and

B.t.u. per cu. ft.

At the price for which sold for industrial purposes, it furnishes only about 600,000 to 900,000 B.t.u. for $1.00, w th coal having a calorific value of only 12,000 B.t.u. per lb., and selling at $7.00 per ton of .b., there is furnished 3,429,000 B.t.u., or approximately four to six times as much as from the for the same cost. However, the greater efficiency with which gas can be utilized, together with the convenience, and in many operations, the increased capacity of apparatus and the decrease in the amount of spoiled work attending its use, makes gas for many industrial purposes a much cheaper fuel than coal.

2. FLOW OF GAS IN PIPES

The rate of flow of gases of different densities, diameter of pipes required, etc., are given by Pole in King's Treatise on Coal Gas, vol. ii, 374, as follows:

If d = diameter of pipe, in.;
Q = quantity of gas, cu. ft. per hour;
: length of pipe, yd.;
in.
of

$$d = \sqrt[5]{Q^2 sl/(1350)^2 h}$$

$$h = Q^2 sl/(1350)^2 d^5$$

Molesworth gives $Q = 1000\sqrt{d^5 h/sl}$.

Table 6.—Factors for Reducing Volumes of Gas to Equivalent Volumes at 60° F. and 30 Inches Barometer *

(Multiply th observed volume by the factor to obtain the equivalent volume)

	Barometer, in.										
	30.0	29 8	29.6	29.4	29.2	.0	28.8				
	1.2095	2014	.1934	.1853	.1772	.1692	.1611	.1530	.1450	.1369	.1288
	1.1956	.1876	1.1796	.1716	.1637	.1557	.1476	.1398	.1318	.1238	.1159
	1.1820	.1741	1.1662	.1583	.1505	.1426	.1347	.1268	.1189	.1111	.1032
	1.1687	1609	1531	.1453	.1375	.1297	.1219	.1141	.1064	.0986	.0908
	1.1557	.1480	1403	.1326	1.1249	.1172	.1095	.1018	.0941	.0863	.0786
	1.1430	.1354	.1277	1.1201	1.1125	.1049	1.0973	.0896	1.0820	.0744	.0668
0	1.1306	.1230	.1155	1.1079	1.1004	1.0929	.0853	1.0778	.0703	1.0627	.0552
5	1.1184	.1109	.1035	1.0960	1.0885	.0811	.0736	1.0662	.0587	1.0513	.0438
10	1.1065	.0991	.0917	.0843	1.0770	1.0696	.0622	1.0548	.0474	1.0401	.0327
15	1.0948	0875	.0802	1.0729	1.0656	.0585	.0510	1.0437	.0364	1.0291	1.0218
20	1.0834	.0762	.0689	1.0617	1.0545	1.0473	.0401	1.0328	.0256	1.0184	1.0112
25	1.0722	0651	.0579	1.0508	1.0436	1.0365	.0293	1.0222	0150	1.0079	1.0007
30	1.0613	.0542	.0471	1.0401	1.0330	1.0259	.0188	1.0118	1.0047	0.9976	0.9905
35	1.0506	.0435	.0365	1.0295	1.0225	1.0155	.0085	1 0015	0.9945	.9875	.9805
40	1.0400	.0331	.0261	1.0192	1.0123	1.0053	0.9984	0.9915	.9845	.9776	.9707
45	1.0297	.0229	.0160	1.0091	1.0023	0.9954	.9885	.9817	.9748	.9679	.9611
50	1.0196	.0128	.0060	0.9992	0.9924	.9856	.9788	.9720	.965	.9584	.9516
55	1.0097	1.0030	0.9962	.9895	.9828	.9761	.9693	.9626	.9559	.9491	.9424
60	1.0000	0.9933	.9867	.9800	.9733	.9667	.9600	.9533	.946	.9400	.9333
65	0.9905	.9838	9772	.9706	.9640	.9574	.9508	.9442	.9376	.9310	.9244
70	.9811	.9746	.9680	.9615	.9550	.9484	.9419	.9353	.9288	.9223	.9157
75	.9719	.9655	.9590	.9525	.9460	.9395	.933	.9266	.920	.9136	.9071
80	.9629	.9565	.9501	.9437	.9373	.9308	.924	.9180	.9116	.9052	.8987
85	.954	.9477	.941	.9350	.9286	.9223	.9159	.9096	.903	.8968	.8905
90	.945	.939	.9328	.9265	.9202	.913	.9076	.9013	.895	.8887	.8824
95	.9369	.930	.924	.918	.911	.9056	.8994	.893	.8869	.8807	.8744
100	.9285	.9223	.916	.909	.9037	.8976	.891	.885	.879	.8728	.8666
105	.9203	.914	.9080	.901	.8957	.889	.8835	.877	.871	.8651	.8589
110	.9122	.906	.9000	.894	.887	.8818	.8757	.869	.863	.8575	.8514
115	.9043	.8982	.8922	.8862	.880	.874		.862	.856	.8500	.8440
120	.896	.890	.8845	.878	.8726	.866	6	.854	.848	.842	.8367

* Formula: Equivalent volume = observed volume × {519.6/(t + 459.6)} × (B/30).

J. P. Gill (*Am. Gas-light Jour.*, 1894) gives $Q = 1291\sqrt{d^5h/s(l + d)}$. This formula is said to be based on experimental data, and to make allowance for obstructions by tar, water, and other bodies tending to check the flow of gas through the pipe.

Practically all problems concerning flow of gas in pipes are solved by gas flow computers of the slide rule type. A commonly used computer, is based on the Spitzglass formula, $Q = 3550\sqrt{\dfrac{hD^5}{SL\ \{1+(3.6/D)+0.03D}}$ where Q = quantity of gas, cu. ft. per hr.; h = loss of pressure, in. of water; S = specific gravity of gas, air being 1; L = length of pipe, ft.; D = internal diameter of pipe, in. The value of the coefficient of friction, used in deriving this formula, is $0.00285\ \{(1 + (3.6/D) + 0.03D)\}$, which was obtained by experiments on pipes of various diameters.

This formula gives slightly larger rates of flow than those given by Pole's formula under similar conditions. The figures in Table 7 were obtained by the slide rule flow computer mentioned above. All of the above formulas apply to low pressure gas, that is gas at a gage pressure of less than 1 lb. per sq. in. For gas at higher pressures, the Spitzglass formula is $Q = 66.5\sqrt{\dfrac{PAD^5}{SL\ \{1 + (3.6/D) + 0.03D\}}}$, where Q = quantity of gas, cu. ft. per hr. reduced to standard conditions of 30 in. pressure and 60° F.; P = drop of gage pressure, lb. per sq. in.; A = mean absolute pressure in pipe line, lb. per sq. in.; D = internal diameter of pipe, in.; L = length of pipe, miles; S = specific gravity of gas, air being 1.

For Natural Gas at High Pressure the formula generally accepted is that of T. R. Weymouth, $Q = 37\sqrt{\{(P_1^2 - P_2^2)\ d^{5.33}\}/L}$, where Q = quantity of gas, cu. ft. per hr.

Table 7.—Maximum Supply of Gas through Pipes in Cu. Ft. per Hour

Specific gravity taken at 0.65. For any other specific gravity y multiply by $\sqrt{0.65/y}$

LENGTH OF PIPE = 50 FT.

Diam. of Pipe, in.	Loss of Pressure, in. of Water Gage									
	0.1	0.2	0.3	0.4	0.5	0.6	0.7	0.8	0.9	1.0
3/4	52	74	91	104	117	128	138	148	157	165
1	105	148	182	210	235	257	279	298	315	332
1 1/4	231	326	398	462	516	565	611	654	693	729
1 1/2	357	506	620	714	794	876	943	1015	1075	1132
2	721	1022	1250	1442	1615	1768	1910	2042	2163	2283

LENGTH OF PIPE = 100 FT.

Diam. of Pipe, in.	Loss of Pressure, in. of Water Gage								
	0.1	0.2	0.4	0.6	0.8	1.0	1.5	2.0	2.5
1 1/4	163	231	326	400	462	515	630	729	815
1 1/2	253	358	506	619	716	797	982	1,130	1,265
2	511	723	1022	1253	1449	1,612	1,973	2,282	2,555
2 1/2	837	1185	1674	2057	2373	2,643	3,235	3,741	4,185
3	1523	2158	3046	3735	4317	4,815	5,895	6,820	7,615
4	3192	4515	6384	7817	9030	10,090	12,360	14,280	15,960

LENGTH OF PIPE = 1000 FT.

Diam. of Pipe, in.	Loss of Pressure, in. of Water Gage						
	0.5	1.0	1.5	2.0	2.5	3.0	4.0
2	360	511	624	720	805	882	1,022
3	1,081	1,523	1,872	3,162	2,417	2,648	3,046
4	2,252	3,192	3,901	4,504	5,036	5,516	6,384
6	6,675	9,451	11,562	13,350	14,926	16,350	18,900
8	13,630	19,250	23,600	27,250	30,470	33,370	38,500
12	38,010	53,810	65,840	76,020	84,990	93,110	107,620

LENGTH OF PIPE = 5280 FT. = 1 MILE

Diam. of Pipe, in.	Loss of Pressure, in. of Water Gage					
	1.0	2.0	3.0	4.0	5.0	6.0
2	222	314	385	444	496	544
3	664	939	1,150	1,328	1,485	1,626
4	1,388	1,962	2,404	2,776	3,104	3,400
6	4,108	5,810	7,115	8,216	9,186	10,060
8	8,370	11,840	14,500	16,740	18,700	20,500
10	14,930	21,110	25,860	29,860	33,390	36,570
12	23,410	33,110	40,550	46,820	52,350	57,350

reduced to standard conditions of 14.65 lb. absolute pressure and 60° F.; P_1 = absolute initial pressure, lb. per sq. in.; P_2 = absolute terminal pressure, lb. per sq. in.; d = inside diameter of pipe, in.; l = length of pipe, miles. This formula being for natural gas, the specific gravity has been assumed as 0.60 and merged into the coefficient 37.

Pole's formula translated into the form of the common formula for the flow of compressed air or steam in pipes, is $Q = c\sqrt{(p_1 - p_2)d^5/wL}$, in which Q = cu. ft. per min.; $(p_1 - p_2)$ = difference in pressure, lb. per sq. in.; w = density, lb. per cu. ft.; L = length, ft.; d = diameter, in. This gives 56.6 for the value of the coefficient c, which is nearly the same as that commonly used, *i.e.*, 60, in calculations of the flow of air in pipes.

An experiment made by Mr. Clegg, in London, with a 4-in. pipe, 6 miles long, pressure 3 in. of water, specific gravity of gas 0.398, gave a discharge into the atmosphere of 852 cu. ft. per hour, after a correction of 33 cu. ft. was made for leakage.

Substituting this value, 852 cu. ft., for Q in the formula $Q = C\sqrt{d^5h \div sl}$, we find C, the coefficient, = 997, which corresponds nearly with the formula given by Molesworth.

Wm. Cox (*Am. Mach.*, March 20, 1902) gives the following formula for flow of gas in long pipes:

$$\sqrt{\;(p_1^2 - p_2^2)/L.}$$

Q discharge, cu. ft. per hour at atmospheric pressure; d = diameter of pipe, in.; p_1 = initial and p_2 = terminal absolute pressure, lb. per sq. in.; l = length of pipe, ft.; L = length, miles. For $p_1^2 - p_2^2$ we may substitute the product $(p_1 + p_2) \times (p_1 - p_2)$. The specific gravity of the gas is assumed to be 0.65, air being 1. For fluids of any other specific gravity, s, multiply the coefficients 3000 or 41.3 by $\sqrt{0.65/s}$. For air, $s = 1$, the coefficients become 2419 and 33.3. J. E. Johnson, Jr.'s, formula for air, p. 1–14, translated into the same notation as Mr. Cox's, makes the coefficients 2449 and 33.5.

It is difficult to accurately check the tables, on account of the extra friction introduced by rough pipes, bends, etc. For bends, one rule is to allow $^1/_{42}$ of an inch pressure for each right-angle bend.

Another method of allowing for the effect of bends is to consider each one as equivalent to so many feet of pipe. The number of feet equivalent to one standard elbow, as given by J. M. Spitzglass, increases continuously from 1.7 ft. for 1-in. pipe to 49 ft. for 12-in. pipe.

The size to be selected for any particular service pipe often depends more upon other conditions than upon the rate at which gas is to be carried. For instance, in the United States, it is not considered good practice to lay any services for lamps with pipe less than 1 in. diameter, nor any house services with pipe of less than 1 $^1/_4$ in. to 1 $^1/_2$ in. diameter, although in the case of the lamp services a $^3/_8$-in. pipe, and in the case of the house services a 1-in. pipe would be sufficient to carry the gas without an undue loss in pressure.

Table 8.—Standard Flanged Cast-Iron Pipe for Gas

(U. S. Cast & Foundry Co., 1932; Am. Gas Inst. Std., 1932)

Nominal Diam., in.	Thickness, in.	Flange* Diam., in.	Flange Thickness, in.	Bolt Circle Diam., in.	Bolts No.	Bolts Size, in.	Weight Single Flange, lb.	Approx. Weight, lb. Foot	Approx. Weight, lb. Length
4	0.40	9.00	0.72	7.125	4	0.625	8.19	18.62	223
6	.43	11.00	.72	9.125	4	.625	10.46	29.01	348
8	.45	13.00	.75	11.125	8	.625	12.65	40.05	481
10	.49	16.00	.86	13.75	8	.625	22.53	54.71	656
12	.54	18.00	.875	15.75	8	.625	25.96	71.34	856
16	.62	22.50	1.00	20.00	12	.75	39.68	108.61	1303
20	.68	27.00	1.00	24.50	16	.75	51.10	147.95	1775
24	.76	31.00	1.125	28.50	16	.75	65.00	197.38	2369
30	.85	37.50	1.25	35.00	20	.875	96.70	273.45	3281
36	.95	44.00	1.375	41.25	24	.875	132.26	366.67	4400
42	1.07	50.75	1.56	47.75	28	1.00	186.83	483.48	5802
48	1.26	57.00	1.75	54.00	32	1.00	235.23	647.36	7768

* Flanges are Am. Gas Inst., and are different from the "American 1928" standard for water and steam pipe. All flanged pipe faced to the exact dimension specified.

Section 5
STEAM

TABLE OF CONTENTS

STEAM

Revised by A. G. Christie

1. PROPERTIES OF STEAM

The definitions on p. 3–79 for saturated heat of liquid, heat of vaporization, dry and saturated vapor, superheat, enthalpy, quality, etc., apply to water and steam.

The British Thermal Unit (B.t.u.) is defined on p. 3–72.

THE TEMPERATURE OF STEAM in contact with water depends upon the pressure under which it is generated. At the ordinary atmospheric pressure (14.7 lb. per sq. in.) its temperature is 212° F. As the pressure is increased, as when the steam is being generated in a closed vessel, its temperature, and that of the water in its presence, increases. See the Steam Tables pp. 5–04 to 5–11 for pressure-temperature relations.

THE ENTHALPY, formerly called the total heat in the steam tables is calculated above 32° F. and includes three elements: 1. The *total heat of saturated liquid* h_f, which is the heat required to raise the temperature of the water from 32° F. to the boiling temperature at the given constant pressure at which it boils. 2. The heat required to completely evaporate the water at that pressure and temperature. This heat is called the *internal latent heat* h_i. 3. The *external work* done by the steam in making room for itself against the pressure under which it is generated, APv_{fg}, where $A = 1/778.6$; P = absolute pressure, lb. per sq. ft.; v_{fg} = change in volume of vapor, cu. ft. per lb. The sum of the last two elements is the *latent heat* of steam h_{fg} or the *total heat of evaporation.* Thus the heat required to generate 1 lb. of steam at 212° F., from water at 32° F. and 14.696 lb. per sq. in. absolute pressure is:

Total heat of saturated liquid at 212° F., h_f	=	180 B.t.u.
Internal latent heat of steam, h_i	=	897.8
External work, $APv_{fg} = \{14.696 \times 144\,(26.82 - 0.017)\}/778.6$	=	72.4
Total heat of evaporation h_{fg}	=	970.2
Enthalpy or total heat of saturated vapor, h_g	=	1150.2 B.t.u.

The enthalpy or total heat of 1 lb. of wet steam at quality x is $h_x = h_f + xh_{fg}$.

The enthalpy or total heat of superheated steam is found by adding to the total heat of saturated vapor the energy added in the superheat. This is found by the calculus from

$h_s = \quad c_p dt = \bar{c}_p (T_2 - T_1)$, where h_s is the energy added; $\bar{c}_p$ = mean specific heat per lb. of superheated steam between T_1 and T_2 at constant pressure; T_1 = absolute temperature of evaporation corresponding to the given pressure and T_2 = absolute temperature of the superheated steam.

PROPERTIES OF SATURATED AND SUPERHEATED STEAM are given in Tables 1 and 2 which are condensed, by permission, from Keenan's Steam Tables and Mollier Diagram. See note on Tolerances, p. 5–07.

The Volumes of 1 lb. of saturated liquid, v_f, of the increase during evaporation v_{fg}, and of the saturated vapor v_g are given in Table 1 for steam at various pressures. To find the volume of 1 lb. of wet steam, $v_x = v_f + xv_{fg}$, where x is the quality. This formula must be used for pressures above 250 lb. per sq. in. v_f is relatively small as compared to v_g at low pressures and may be neglected in commercial calculations as, for instance, on exhaust steam volumes. Then $v_x{}^1 = xv_g$ is approximately correct.

The Volume of Superheated Steam is given in Table 2.

The Entropies of saturated liquid s_f, of evaporation s_{fg}, and of saturated vapor s_g, are given in Table 1. To find the entropy of 1 lb. of wet steam at x quality, $s_f + xs_{fg}$

The Entropies of Superheated Steam are given in Table 2.

The total heat of saturated liquid h_f as given in these tables does not include the work of the feed pump. It is the heat added to 1 lb. of water to raise it to the boiling point from 32° F., during which process the pressure remains constant and is that of evaporation.

Table 1.—Properties of Saturated Steam

Condensed by permission from Keenan's Steam Tables

Vacuum, in. of Mercury	Absolute Pressure, lb. per sq. in. p	Temp., deg. F. t	Specific Volume, cu. ft. per lb. Of Liquid v_f	Specific Volume, cu. ft. per lb. Of Steam v_g	Weight of Steam, lb. per cu. ft.	Enthalpy or Heat Content, B.t.u. per lb. In Liquid h_f	Enthalpy or Heat Content, B.t.u. per lb. Latent h_{fg}	Enthalpy or Heat Content, B.t.u. per lb. Total h_g	Entropy Of Liquid s_f	Entropy Of Vaporization, s_{fg}	Entropy Of Steam s_g
27.885	1	101.76	0.01614	333.9	0.00299	69.69	1035.3	1105.0	0.1326	1.8442	1.9769
25.849	2	126.10	.01623	173.96	.00575	93.97	1021.6	1115.6	.1750	1.7442	1.9192
23.813	3	141.49	.01630	118.86	.00841	109.33	1012.7	1122.0	.2009	1.6847	1.8856
21.777	4	152.99	.01636	90.74	.01102	120.83	1005.9	1126.8	.2198	1.6420	1.8618
19.741	5	162.25	.01641	73.61	.01359	130.10	1000.4	1130.6	.2348	1.6088	1.8435
17.705	6	170.07	.01645	62.05	.01612	137.92	995.8	1133.7	.2473	1.5814	1.8287
15.669	7	176.85	.01649	53.70	.01862	144.71	991.7	1136.4	.2580	1.5582	1.8162
13.633	8	182.87	.01652	47.39	.02110	150.75	988.1	1138.9	.2674	1.5379	1.8053
11.597	9	188.28	.01656	42.44	.02356	156.19	984.8	1141.0	.2758	1.5200	1.7958
9.561	10	193.21	.01658	38.45	.02600	161.13	981.8	1143.0	.2834	1.5040	1.7874
7.525	11	197.75	.01661	35.17	.02843	165.68	979.1	1144.8	.2903	1.4894	1.7797
5.489	12	201.96	.01664	32.42	.03084	169.91	976.5	1146.4	.2968	1.4760	1.7727
3.453	13	205.88	.01666	30.08	.03324	173.85	974.1	1147.9	.3027	1.4636	1.7663
1.417	14	209.56	.01669	28.06	.03564	177.55	971.8	1149.3	.3082	1.4521	1.7604
0	14.696	212.00	.01670	26.82	.03729	180.00	970.2	1150.2	.3119	1.4446	1.7564
	15	213.03	.01671	26.31	.03801	181.04	969.6	1150.6	.3134	1.4414	1.7548
	16	216.32	.01673	24.76	.04039	184.35	967.4	1151.8	.3184	1.4312	1.7496
	17	219.43	.01676	23.40	.04274	187.48	965.4	1152.9	.3230	1.4218	1.7448
	18	222.40	.01678	22.18	.04509	190.48	963.5	1154.0	.3274	1.4127	1.7402
	19	225.23	.01680	21.08	.04744	193.34	961.7	1155.0	.3316	1.4042	1.7358
	20	227.96	.01682	20.095	.04976	196.09	959.9	1156.0	.3356	1.3960	1.7317
	21	230.56	.01684	19.197	.05209	198.72	958.2	1156.9	.3395	1.3883	1.7278
	22	233.07	.01685	18.380	.05441	201.25	956.6	1157.8	.3431	1.3809	1.7240
	23	235.49	.01687	17.630	.05672	203.70	955.0	1158.6	.3466	1.3738	1.7204
	24	237.82	.01689	16.941	.05903	206.05	953.4	1159.5	.3500	1.3670	1.7170
	25	240.07	.01690	16.306	.06133	208.33	951.9	1160.2	.3533	1.3604	1.7137
	26	242.25	.01692	15.718	.06362	210.54	950.4	1161.0	.3564	1.3542	1.7106
	27	244.36	.01694	15.172	.06591	212.67	949.0	1161.7	.3594	1.3481	1.7075
	28	246.41	.01695	14.664	.06819	214.75	947.7	1162.4	.3624	1.3422	1.7046
	29	248.40	.01697	14.190	.07047	216.77	946.3	1163.1	.3652	1.3365	1.7018
	30	250.34	.01698	13.745	.07275	218.73	945.0	1163.7	.3680	1.3310	1.6990
	31	252.22	.01700	13.330	.07502	220.64	943.7	1164.4	.3707	1.3257	1.6964
	32	254.05	.01701	12.940	.07728	222.50	942.5	1165.0	.3732	1.3206	1.6938
	33	255.84	.01703	12.573	.07954	224.32	941.2	1165.6	.3758	1.3156	1.6914
	34	257.58	.01704	12.226	.08179	226.09	940.0	1166.1	.3783	1.3107	1.6890
	35	259.28	.01706	11.898	.08405	227.82	938.9	1166.7	.3807	1.3060	1.6866
	36	260.94	.01707	11.587	.08630	229.51	937.7	1167.2	.3830	1.3014	1.6844
	37	262.57	.01708	11.294	.08854	231.17	936.6	1167.8	.3853	1.2968	1.6822
	38	264.16	.01710	11.015	.09079	232.79	935.5	1168.3	.3876	1.2925	1.6800
	39	265.72	.01711	10.750	.09302	234.38	934.4	1168.8	.3898	1.2882	1.6780
	40	267.24	.01712	10.497	.09527	235.93	933.3	1169.2	.3919	1.2840	1.6759
	41	268.74	.01713	10.257	.09749	237.45	932.3	1169.7	.3940	1.2799	1.6739
	42	270.21	.01715	10.027	.09973	238.95	931.2	1170.2	.3961	1.2759	1.6720
	43	271.65	.01716	9.808	.1020	240.42	930.2	1170.6	.3981	1.2720	1.6701
	44	273.06	.01717	9.599	.1042	241.86	929.2	1171.1	.4000	1.2682	1.6683
	45	274.45	.01718	9.399	.1064	243.28	928.2	1171.5	.4020	1.2645	1.6665
	46	275.81	.01719	9.207	.1086	244.67	927.2	1171.9	.4039	1.2608	1.6647
	47	277.14	.01720	9.023	.1108	246.03	926.3	1172.3	.4057	1.2572	1.6630
	48	278.45	.01722	8.846	.1130	247.37	925.4	1172.7	.4076	1.2537	1.6613
	49	279.74	.01723	8.676	.1153	248.68	924.4	1173.1	.4093	1.2503	1.6596
	50	281.01	.01724	8.514	.1175	249.98	923.5	1173.5	.4111	1.2469	1.6580
	51	282.26	.01725	8.357	.1197	251.26	922.6	1173.9	.4128	1.2436	1.6564
	52	283.49	.01726	8.206	.1219	252.52	921.7	1174.3	.4145	1.2404	1.6549
	53	284.70	.01727	8.060	.1241	253.76	920.9	1174.6	.4162	1.2372	1.6533
	54	285.90	.01728	7.919	.1263	254.99	920.0	1175.0	.4178	1.2340	1.6518
	55	287.07	.01729	7.783	.1285	256.19	919.1	1175.3	.4194	1.2309	1.6504
	56	288.23	.01730	7.653	.1307	257.38	918.3	1175.7	.4210	1.2279	1.6489
	57	289.37	.01731	7.527	.1329	258.55	917.5	1176.0	.4226	1.2249	1.6475
	58	290.50	.01732	7.405	.1350	259.71	916.6	1176.4	.4241	1.2220	1.6461
	59	291.62	.01734	7.287	.1372	260.86	915.8	1176.7	.4256	1.2191	1.6447

Table 1.—Properties of Saturated Steam—*Continued*

Absolute Pressure, lb. per sq. in. p	Temp., deg. F. t	Specific Volume, cu. ft. per lb.		Weight of Steam, lb. per cu. ft.	Enthalpy or Heat Content, B.t.u. per lb.			Entropy		
		Of Liquid v_f	Of Steam v_g		In Liquid h_f	Latent h_{fg}	Total h_g	Of Liquid s_f	Of Vaporization, s_{fg}	Of Steam s_g
60	292.71	0.01735	7.172	0.1394	261.98	915.0	1177.0	0.4271	1.2162	1.6434
61	293.79	.01736	7.062	.1416	263.09	914.2	1177.3	.4286	1.2134	1.6420
62	294.85	.01737	6.955	.1438	264.18	913.4	1177.6	.4300	1.2107	1.6407
63	295.91	.01738	6.850	.1460	265.27	912.7	1177.9	.4315	1.2080	1.6394
64	296.94	.01739	6.749	.1482	266.33	911.9	1178.2	.4329	1.2053	1.6382
65	297.97	.01740	6.652	.1503	267.39	911.1	1178.5	.4343	1.2026	1.6369
66	298.98	.01741	6.556	.1525	268.43	910.4	1178.8	.4356	1.2001	1.6357
67	299.99	.01742	6.464	.1547	269.47	909.6	1179.1	.4370	1.1975	1.6345
68	300.98	.01743	6.375	.1569	270.49	908.9	1179.4	.4384	1.1950	1.6333
69	301.96	.01744	6.288	.1590	271.50	908.2	1179.6	.4397	1.1924	1.6321
70	302.92	.01744	6.203	.1612	272.49	907.4	1179.9	.4410	1.1900	1.6310
71	303.88	.01746	6.121	.1634	273.48	906.7	1180.2	.4423	1.1876	1.6298
72	304.82	.01746	6.041	.1655	274.45	906.0	1180.5	.4435	1.1852	1.6287
73	305.76	.01747	5.963	.1677	275.42	905.3	1180.7	.4448	1.1828	1.6276
74	306.68	.01748	5.887	.1699	276.37	904.6	1181.0	.4460	1.1805	1.6265
75	307.60	.01749	5.813	.1720	277.32	903.9	1181.2	.4473	1.1782	1.6254
76	308.50	.01750	5.741	.1742	278.25	903.2	1181.5	.4485	1.1759	1.6244
77	309.39	.01751	5.671	.1763	279.17	902.6	1181.7	.4497	1.1736	1.6233
78	310.28	.01752	5.602	.1785	280.09	901.9	1182.0	.4509	1.1714	1.6223
79	311.16	.01753	5.535	.1807	281.00	901.2	1182.2	.4520	1.1692	1.6212
80	312.03	.01754	5.470	.1828	281.90	900.5	1182.4	.4532	1.1670	1.6202
81	312.88	.01755	5.406	.1850	282.79	899.9	1182.7	.4544	1.1649	1.6192
82	313.74	.01756	5.343	.1872	283.67	899.2	1182.9	.4555	1.1627	1.6182
83	314.58	.01756	5.282	.1893	284.55	898.6	1183.1	.4566	1.1606	1.6173
84	315.42	.01757	5.222	.1915	285.42	897.9	1183.4	.4578	1.1586	1.6163
85	316.25	.01758	5.164	.1936	286.28	897.3	1183.6	.4589	1.1565	1.6154
86	317.06	.01759	5.107	.1958	287.13	896.7	1183.8	.4599	1.1545	1.6144
87	317.88	.01760	5.051	.1979	287.97	896.0	1184.0	.4610	1.1524	1.6135
88	318.68	.01761	4.997	.2001	288.80	895.4	1184.2	.4621	1.1505	1.6126
89	319.48	.01762	4.944	.2023	289.63	894.8	1184.4	.4632	1.1485	1.6116
90	320.27	.01763	4.892	.2044	290.45	894.2	1184.6	.4642	1.1465	1.6107
91	321.05	.01764	4.841	.2066	291.26	893.6	1184.8	.4652	1.1446	1.6099
92	321.83	.01764	4.791	.2087	292.07	893.0	1185.0	.4663	1.1427	1.6090
93	322.60	.01765	4.742	.2109	292.87	892.4	1185.2	.4673	1.1408	1.6081
94	323.37	.01766	4.694	.2130	293.67	891.8	1185.4	.4683	1.1389	1.6072
95	324.13	.01767	4.647	.2152	294.47	891.2	1185.6	.4693	1.1370	1.6064
96	324.88	.01768	4.602	.2173	295.25	890.6	1185.8	.4703	1.1352	1.6055
97	325.62	.01768	4.557	.2194	296.02	890.0	1186.0	.4713	1.1334	1.6047
98	326.37	.01769	4.512	.2216	296.80	889.4	1186.2	.4723	1.1316	1.6038
99	327.10	.01770	4.468	.2238	297.57	888.8	1186.4	.4732	1.1298	1.6030
100	327.83	.01771	4.426	.2259	298.33	888.2	1186.6	.4742	1.1280	1.6022
102	329.27	.01773	4.344	.2302	299.83	887.1	1186.9	.4761	1.1245	1.6006
104	330.68	.01774	4.265	.2345	301.30	886.0	1187.3	.4779	1.1211	1.5990
106	332.08	.01776	4.189	.2387	302.76	884.9	1187.6	.4798	1.1177	1.5974
108	333.44	.01777	4.115	.2430	304.19	883.8	1188.0	.4816	1.1144	1.5959
110	334.79	.01779	4.044	.2473	305.61	882.7	1188.3	.4834	1.1111	1.5944
112	336.12	.01780	3.976	.2515	307.00	881.6	1188.6	.4851	1.1079	1.5930
114	337.43	.01782	3.910	.2558	308.36	880.6	1188.9	.4868	1.1048	1.5915
116	338.72	.01783	3.846	.2600	309.71	879.5	1189.2	.4885	1.1017	1.5901
118	340.01	.01785	3.784	.2643	311.05	878.5	1189.5	.4901	1.0986	1.5887
120	341.26	.01786	3.725	.2685	312.37	877.4	1189.8	.4918	1.0956	1.5874
122	342.50	.01788	3.670	.2725	313.67	876.4	1190.1	.4934	1.0926	1.5860
124	343.73	.01789	3.615	.2766	314.96	875.4	1190.4	.4950	1.0897	1.5847
126	344.94	.01791	3.560	.2809	316.23	874.4	1190.6	.4965	1.0868	1.5834
128	346.14	.01792	3.505	.2853	317.49	873.4	1190.9	.4981	1.0840	1.5821
130	347.31	.01794	3.451	.2898	318.73	872.4	1191.2	.4996	1.0812	1.5808
132	348.48	.01795	3.401	.2940	319.95	871.5	1191.4	.5011	1.0784	1.5796
134	349.64	.01796	3.353	.2982	321.17	870.5	1191.7	.5026	1.0757	1.5783
136	350.78	.01798	3.306	.3025	322.37	869.6	1191.9	.5041	1.0730	1.5771
138	351.91	.01799	3.260	.3067	323.56	868.6	1192.2	.5056	1.0703	1.5759

(*Table continued on following page*)

Table 1.—Properties of Saturated Steam—*Continued*

Absolute Pressure, lb. per sq. in. p	Temp., deg. F. t	Specific Volume, cu. ft. per lb. Of Liquid v_f	Specific Volume, cu. ft. per lb. Of Steam v_g	Weight of Steam, lb. per cu. ft.	Enthalpy or Heat Content, B.t.u. per lb. In Liquid h_f	Enthalpy or Heat Content, B.t.u. per lb. Latent h_{fg}	Enthalpy or Heat Content, B.t.u. per lb. Total h_g	Entropy Of Liquid s_f	Entropy Of Vaporization, s_{fg}	Entropy Of Steam s_g
140	353.03	0.01801	3.216	0.3109	324.74	867.7	1192.4	0.5070	1.0677	1.5747
142	354.14	.01802	3.173	.3152	325.91	866.7	1192.6	.5084	1.0651	1.5735
144	355.22	.01804	3.130	.3195	327.06	865.8	1192.9	.5098	1.0625	1.5724
146	356.31	.01805	3.089	.3237	328.20	864.9	1193.1	.5112	1.0600	1.5712
148	357.37	.01806	3.049	.3280	329.32	864.0	1193.3	.5126	1.0575	1.5701
150	358.43	.01808	3.010	.3322	330.44	863.1	1193.5	.5140	1.0550	1.5690
152	359.47	.01809	2.972	.3365	331.54	862.2	1193.7	.5153	1.0526	1.5679
154	360.51	.01810	2.935	.3407	332.64	861.3	1193.9	.5166	1.0502	1.5668
156	361.53	.01812	2.900	.3448	333.72	860.4	1194.1	.5180	1.0478	1.5658
158	362.54	.01813	2.864	.3492	334.80	859.5	1194.3	.5193	1.0454	1.5647
160	363.55	.01814	2.830	.3534	335.86	858.7	1194.5	.5205	1.0431	1.5636
162	364.54	.01816	2.797	.3575	336.91	857.8	1194.7	.5218	1.0408	1.5626
164	365.52	.01817	2.764	.3618	337.95	857.0	1194.9	.5230	1.0385	1.5616
166	366.50	.01818	2.733	.3659	338.99	856.1	1195.1	.5243	1.0363	1.5606
168	367.46	.01819	2.701	.3702	340.01	855.2	1195.3	.5255	1.0340	1.5596
170	368.42	.01821	2.671	.3744	341.03	854.4	1195.4	.5268	1.0318	1.5586
172	369.37	.01822	2.641	.3786	342.04	853.6	1195.6	.5280	1.0296	1.5576
174	370.31	.01823	2.612	.3828	343.04	852.7	1195.8	.5292	1.0275	1.5566
176	371.24	.01825	2.584	.3870	344.03	851.9	1196.0	.5304	1.0253	1.5557
178	372.16	.01826	2.556	.3912	345.01	851.1	1196.1	.5315	1.0232	1.5548
180	373.08	.01827	2.529	.3954	345.99	850.3	1196.3	.5327	1.0211	1.5538
182	374.00	.01828	2.502	.3997	346.97	849.5	1196.4	.5339	1.0190	1.5529
184	374.90	.01829	2.476	.4039	347.94	848.6	1196.6	.5350	1.0169	1.5520
186	375.78	.01831	2.451	.4080	348.89	847.9	1196.8	.5362	1.0149	1.5511
188	376.67	.01832	2.425	.4124	349.83	847.1	1196.9	.5373	1.0129	1.5502
190	377.55	.01833	2.401	.4165	350.77	846.3	1197.0	.5384	1.0109	1.5493
192	378.42	.01834	2.377	.4207	351.70	845.5	1197.2	.5395	1.0089	1.5484
194	379.27	.01835	2.353	.4250	352.61	844.7	1197.3	.5406	1.0070	1.5475
196	380.13	.01837	2.330	.4292	353.53	844.0	1197.5	.5417	1.0050	1.5467
198	380.97	.01838	2.307	.4335	354.43	843.2	1197.6	.5427	1.0031	1.5458
200	381.82	.01839	2.285	.4376	355.33	842.4	1197.8	.5438	1.0012	1.5450
205	383.89	.01842	2.231	.4482	357.56	840.5	1198.1	.5465	0.9964	1.5429
210	385.93	.01844	2.180	.4587	359.76	838.6	1198.4	.5491	.9918	1.5409
215	387.93	.01847	2.131	.4693	361.91	836.8	1198.7	.5516	.9873	1.5389
220	389.89	.01850	2.084	.4798	364.02	835.0	1199.0	.5540	.9829	1.5369
225	391.81	.01853	2.039	.4904	366.10	833.2	1199.3	.5565	.9786	1.5350
230	393.70	.01856	1.996	.5009	368.14	831.4	1199.6	.5588	.9743	1.5332
235	395.56	.01859	1.955	.5114	370.15	829.7	1199.8	.5612	.9702	1.5313
240	397.40	.01861	1.916	.5220	372.13	827.9	1200.1	.5635	.9661	1.5295
245	399.20	.01864	1.878	.5326	374.09	826.2	1200.3	.5658	.9620	1.5278
250	400.97	.01867	1.841	.5432	376.02	824.5	1200.5	.5680	.9581	1.5261
255	402.71	.01869	1.806	.5537	377.91	822.9	1200.8	.5701	.9542	1.5244
260	404.43	.01872	1.772	.5642	379.78	821.2	1201.0	.5723	.9504	1.5227
265	406.12	.01874	1.740	.5748	381.62	819.6	1201.2	.5744	.9467	1.5210
270	407.79	.01877	1.708	.5854	383.44	818.0	1201.4	.5765	.9430	1.5194
275	409.44	.01880	1.678	.5959	385.24	816.3	1201.6	.5785	.9393	1.5178
280	411.06	.01882	1.649	.6064	387.02	814.7	1201.8	.5805	.9357	1.5163
285	412.66	.01884	1.621	.6171	388.77	813.2	1201.9	.5825	.9322	1.5147
290	414.24	.01887	1.593	.6276	390.50	811.6	1202.1	.5845	.9287	1.5132
295	415.80	.01889	1.567	.6382	392.21	810.0	1202.2	.5864	.9253	1.5117
300	417.33	.01892	1.541	.6488	393.90	808.5	1202.4	.5883	.9220	1.5102
310	420.35	.01896	1.493	.6699	397.23	805.5	1202.7	.5920	.9153	1.5074
320	423.29	.01901	1.447	.6911	400.47	802.5	1203.0	.5957	.9089	1.5046
330	426.16	.01905	1.4039	.7123	403.64	799.5	1203.2	.5992	.9026	1.5018
340	428.96	.01910	1.3630	.7337	406.75	796.6	1203.4	.6027	.8965	1.4992
350	431.71	.01914	1.3245	.7550	409.81	793.7	1203.6	.6061	.8905	1.4966
360	434.39	.01918	1.2881	.7763	412.80	790.9	1203.7	.6094	.8846	1.4940
370	437.01	.01923	1.2536	.7977	415.73	788.1	1203.8	.6126	.8789	1.4915
380	439.59	.01927	1.2208	.8191	418.61	785.3	1203.9	.6157	.8733	1.4891
390	442.11	.01931	1.1898	.8405	421.44	782.6	1204.0	.6188	.8678	1.4867

Table 1. —Properties of Saturated Steam—*Continued*

Absolute Pressure, lb. per sq. in. p	Temp., deg. F. t	Specific Volume, cu. ft. per lb.		Weight of Steam, lb. per cu. ft.	Enthalpy or Heat Content, B.t.u. per lb.			Entropy		
		Of Liquid v_f	Of Steam v_g		In Liquid h_f	Latent h_{fg}	Total h_g	Of Liquid s_f	Of Vaporization, s_{fg}	Of Steam s_g
400	444.58	0.0194	1.1601	0.8620	424.2	779.8	1204.1	0.6218	0.8625	1.4843
420	449.38	.0194	1.1047	.9052	429.6	774.5	1204.1	.6277	.8520	1.4798
440	454.01	.0195	1.0540	.9488	434.8	769.3	1204.1	.6334	.8420	1.4753
460	458.48	.0196	1.0077	.9924	439.9	764.1	1204.0	.6388	.8322	1.4711
480	462.80	.0197	0.9656	1.0359	444.9	759.0	1203.9	.6441	.8228	1.4670
500	466.99	.0198	.9261	1.0798	449.7	754.0	1203.7	.6493	.8137	1.4630
520	471.05	.0198	.8899	1.1237	454.4	749.0	1203.5	.6543	.8048	1.4591
540	474.99	.0199	.8562	1.1680	459.0	744.1	1203.2	.6592	.7962	1.4554
560	478.82	.0200	.8247	1.2126	463.6	739.3	1202.9	.6639	.7878	1.4517
580	482.55	.0201	.7952	1.2575	468.0	734.5	1202.5	.6686	.7796	1.4482
600	486.17	.0202	.7677	1.3026	472.3	729.8	1202.1	.6731	.7716	1.4447
620	489.71	.0202	.7419	1.3479	476.6	725.1	1201.7	.6775	.7638	1.4413
640	493.16	.0203	.7175	1.3937	480.8	720.5	1201.2	.6818	.7562	1.4380
660	496.53	.0204	.6948	1.4392	484.9	715.9	1200.8	.6861	.7487	1.4348
680	499.82	.0205	.6732	1.4854	488.9	711.3	1200.2	.6902	.7414	1.4316
700	503.04	.0206	.6527	1.5321	492.9	706.8	1199.7	.6943	.7342	1.4285
720	506.19	.0206	.6334	1.5788	496.8	702.4	1199.2	.6983	.7272	1.4255
740	509.28	.0207	.6151	1.6258	500.6	697.9	1198.6	.7022	.7203	1.4225
760	512.30	.0208	.5977	1.6731	504.4	693.5	1198.0	.7060	.7136	1.4196
780	515.27	.0209	.5811	1.7209	508.2	689.2	1197.4	.7098	.7069	1.4167
800	518.18	.0209	.5653	1.7690	511.8	684.9	1196.7	.7135	.7004	1.4139
820	521.03	.0210	.5503	1.8172	515.5	680.6	1196.0	.7171	.6940	1.4111
840	523.83	.0211	.5360	1.8657	519.0	676.4	1195.4	.7207	.6877	1.4084
860	526.58	.0212	.5225	1.9139	522.6	672.1	1194.7	.7242	.6815	1.4057
880	529.29	.0213	.5094	1.9631	526.0	667.9	1194.0	.7277	.6754	1.4031
900	531.95	.0213	.4969	2.0125	529.5	663.8	1193.3	.7311	.6694	1.4005
920	534.56	.0214	.4849	2.0623	532.9	659.7	1192.6	.7344	.6635	1.3980
940	537.13	.0215	.4735	2.1119	536.2	655.6	1191.8	.7377	.6577	1.3954
960	539.66	.0216	.4625	2.1622	539.6	651.5	1191.1	.7410	.6520	1.3930
980	542.14	.0217	.4520	2.2124	542.8	647.5	1190.3	.7442	.6464	1.3905
1000	544.58	.0217	.4419	2.2630	546.0	643.5	1189.6	.7473	.6408	1.3881
1100	556.28	.0222	.3960	2.5253	561.7	623.9	1185.6	.7624	.6141	1.3765
1200	567.14	.0226	.3582	2.7917	576.5	604.9	1181.4	.7764	.5891	1.3656
1300	577.32	.0230	.3259	3.0684	590.6	586.3	1177.0	.7897	.5654	1.3552
1400	586.96	.0235	.2983	3.3523	604.3	568.1	1172.4	.8024	.5428	1.3452
1500	596.08	.0239	.2741	3.6483	617.5	550.2	1167.6	.8146	.5212	1.3357
1600	604.74	.0244	.2528	3.9557	630.2	532.6	1162.7	.8262	.5003	1.3265
1700	612.98	.0249	.2338	4.2772	642.5	515.0	1157.5	.8373	.4801	1.3174
1800	620.86	.0254	.2167	4.6147	654.7	497.2	1151.8	.8482	.4601	1.3083
1900	628.39	.0260	.2014	4.9652	666.8	478.9	1145.7	.8589	.4402	1.2990
2000	635.6	.0265	.1875	5.3333	679.0	460.0	1139.0	.8696	.4200	1.2896
2100	642.6	.0271	.1744	5.7339	691.3	440.4	1131.7	.8804	.3996	1.2800
2200	649.2	.0277	.1623	6.1614	703.7	420.0	1123.8	.8912	.3788	1.2700
2300	655.7	.0284	.1510	6.6225	716.4	398.7	1115.2	.9021	.3575	1.2596
2400	661.9	.0292	.1404	7.1225	729.4	376.4	1105.8	.9133	.3356	1.2488
2500	668.0	.0301	.1303	7.6746	742.8	352.8	1095.6	.9247	.3129	1.2375
2600	673.8	.0310	.1205	8.2998	756.7	327.8	1084.5	.9364	.2892	1.2257
2700	679.5	.0321	.1111	9.0000	771.2	301.2	1072.4	.9487	.2644	1.2131
2800	684.9	.0333	.1021	9.7943	786.7	272.3	1058.9	.9618	.2379	1.1996
2900	690.2	.0348	.0933	10.718	803.6	240.0	1043.7	.9760	.2088	1.1847
3000	695.2	.0367	.0844	11.848	823.1	202.5	1025.6	.9922	.1754	1.1676
3100	700.2	.0395	.0743	13.459	847.2	155.0	1002.2	1.0126	.1336	1.1462
3200	704.9	.0459	.0601	16.639	887.0	75.9	962.9	1.0461	.0651	1.1112
3226	706.1	.0522	.0522	19.157	925.0	0.0	925.0	1.0785	.0000	1.0785

STEAM TABLE TOLERANCES AND CORRECTIONS.—The Third International Steam Table Conference has adopted tolerances for steam tables (*Mech. Engg.*, Nov., 1935) which have been made possible by greater refinements in research. Tables 4, 5, and 6 are the skeleton tables issued by the conference. Table 3 gives corrections which may be applied to Tables 1 and 2 to bring them into conformity with the International tolerances. Table 3 was compiled from curves developed by Ernest L. Robinson of the General

(*Continued on p. 5–13*)

Table 2.—Properties of Superheated Steam

v = specific volume, cu. ft. per lb.; h = enthalpy or total heat content, B.t.u. per lb.; s = entropy, B.t.u. per deg. F. per lb.

Absolute Pressure, lb. per sq. in.	Temperature, Sat., deg. F.		Superheated Steam, Total Temperature, deg. F.																
			300	320	340	360	380	400	420	440	460	480	500	540	600	700	800	900	1000
20	227.96	v	22.36	22.98	23.59	24.21	24.82	25.43	26.03	26.63	27.24	27.84	28.45	29.65	31.46	34.46	37.44	40.43	43.42
		h	1191.1	1200.6	1210.0	1219.5	1228.9	1238.3	1247.7	1257.1	1266.6	1276.0	1285.5	1304.6	1333.4	1382.1	1432.0	1482.8	1534.8
		s	1.7802	1.7926	1.8045	1.8162	1.8275	1.8386	1.8494	1.8600	1.8704	1.8806	1.8906	1.9096	1.9380	1.9819	2.0232	2.0620	2.0989
40	267.24	v	11.044	11.367	11.686	11.999	12.312	12.623	12.932	13.238	13.547	13.855	14.161	14.770	15.682	17.190	18.686	20.18	21.68
		h	1186.5	1196.5	1206.4	1216.2	1225.9	1235.6	1245.3	1254.9	1264.5	1274.1	1283.7	1303.0	1332.0	1381.2	1431.3	1482.3	1534.4
		s	1.6990	1.7122	1.7248	1.7369	1.7486	1.7599	1.7710	1.7818	1.7924	1.8027	1.8128	1.8325	1.8607	1.9050	1.9464	1.9854	2.0224
60	292.71	v	7.260	7.490	7.712	7.929	8.142	8.353	8.565	8.772	8.983	9.191	9.398	9.809	10.423	11.435	12.436	13.439	14.440
		h	1181.2	1192.1	1202.6	1212.8	1222.9	1232.8	1242.7	1252.5	1262.3	1272.1	1281.9	1301.3	1330.7	1380.2	1430.6	1481.8	1533.9
		s	1.6488	1.6631	1.6764	1.6890	1.7011	1.7128	1.7242	1.7352	1.7460	1.7564	1.7667	1.7866	1.8151	1.8597	1.9014	1.9406	1.9775
80	312.03	v	......	5.543	5.720	5.889	6.055	6.217	6.380	6.538	6.698	6.858	7.015	7.328	7.793	8.558	9.313	10.067	10.817
		h	......	1187.1	1198.4	1209.2	1219.6	1229.9	1240.1	1250.2	1260.1	1270.1	1280.0	1299.7	1329.3	1379.2	1429.9	1481.3	1533.5
		s	......	1.6262	1.6405	1.6538	1.6665	1.6785	1.6902	1.7015	1.7125	1.7232	1.7336	1.7537	1.7824	1.8274	1.8694	1.9086	1.9456
100	327.83	v	......	......	4.520	4.663	4.800	4.934	5.068	5.198	5.328	5.457	5.585	5.840	6.215	6.831	7.439	8.044	8.644
		h	......	......	1193.9	1205.3	1216.2	1226.9	1237.4	1247.7	1257.9	1268.0	1278.0	1298.0	1327.9	1378.2	1429.2	1480.7	1533.1
		s	......	......	1.6114	1.6254	1.6387	1.6512	1.6633	1.6748	1.6860	1.6969	1.7075	1.7279	1.7569	1.8023	1.8445	1.8838	1.9209
120	341.26	v	......	......	......	3.842	3.961	4.077	4.192	4.303	4.414	4.523	4.632	4.846	5.162	5.680	6.189	6.693	7.196
		h	......	......	......	1201.2	1212.7	1223.8	1234.6	1245.2	1255.6	1265.9	1276.1	1296.3	1326.5	1377.2	1428.5	1480.2	1532.7
		s	......	......	......	1.6013	1.6152	1.6283	1.6407	1.6526	1.6640	1.6751	1.6859	1.7065	1.7359	1.7816	1.8240	1.8635	1.9007
140	353.03	v	......	......	......	3.255	3.362	3.465	3.566	3.664	3.761	3.856	3.951	4.136	4.410	4.857	5.297	5.728	6.161
		h	......	......	......	1196.8	1208.9	1220.5	1231.7	1242.5	1253.2	1263.7	1274.1	1294.6	1325.1	1376.2	1427.8	1479.7	1532.2
		s	......	......	......	1.5801	1.5947	1.6084	1.6212	1.6334	1.6451	1.6564	1.6674	1.6883	1.7179	1.7640	1.8066	1.8462	1.8836
160	363.55	v	......	......	......	......	2.912	3.005	3.096	3.184	3.270	3.355	3.440	3.604	3.846	4.240	4.627	5.006	5.385
		h	......	......	......	......	1205.0	1217.1	1228.7	1239.9	1250.8	1261.5	1272.1	1292.8	1323.7	1375.2	1427.1	1479.1	1531.8
		s	......	......	......	......	1.5762	1.5906	1.6038	1.6164	1.6284	1.6399	1.6510	1.6722	1.7022	1.7487	1.7915	1.8313	1.8687
180	373.08	v	......	......	......	......	2.560	2.646	2.730	2.810	2.888	2.965	3.041	3.190	3.407	3.760	4.105	4.444	4.782
		h	......	......	......	......	1200.8	1213.5	1225.6	1237.1	1248.3	1259.2	1270.0	1291.1	1322.2	1374.2	1426.3	1478.5	1531.4
		s	......	......	......	......	1.5593	1.5742	1.5880	1.6010	1.6134	1.6251	1.6364	1.6580	1.6882	1.7351	1.7782	1.8180	1.8555
200	381.82	v	......	......	......	......	......	2.358	2.436	2.510	2.583	2.654	2.722	2.858	3.056	3.376	3.688	3.995	4.299
		h	......	......	......	......	......	1209.8	1222.3	1234.2	1245.7	1256.9	1267.9	1289.3	1320.8	1373.1	1425.6	1478.0	1531.0
		s	......	......	......	......	......	1.5592	1.5735	1.5869	1.5996	1.6116	1.6231	1.6450	1.6756	1.7228	1.7662	1.8062	1.8438
250	400.97	v	......	......	......	......	......	......	1.9053	1.9698	2.032	2.090	2.149	2.261	2.424	2.685	2.938	3.186	3.429
		h	......	......	......	......	......	......	1213.8	1226.7	1239.1	1250.9	1262.4	1284.7	1317.1	1370.5	1423.7	1476.6	1530.0
		s	......	......	......	......	......	......	1.5412	1.5558	1.5693	1.5854	1.5944	1.6170	1.6483	1.6966	1.7406	1.7810	1.8188
300	417.33	v	......	......	......	......	......	......	1.5493	1.6067	1.6618	1.7140	1.7648	1.8626	2.002	2.224	2.438	2.646	2.849
		h	......	......	......	......	......	......	1204.4	1218.6	1231.9	1244.6	1256.7	1280.0	1313.3	1367.8	1421.7	1475.1	1528.9
		s	......	......	......	......	......	......	1.5126	1.5284	1.5430	1.5567	1.5695	1.5932	1.6256	1.6747	1.7193	1.7601	1.7983
400	444.58	v	......	......	......	......	......	......	......	......	1.1958	1.2403	1.2828	1.3626	1.4740	1.6472	1.8119	1.9704	2.125
		h	......	......	......	......	......	......	......	......	1216.0	1230.6	1244.3	1269.9	1305.5	1362.3	1417.7	1472.1	1526.8
		s	......	......	......	......	......	......	......	......	1.4974	1.5132	1.5276	1.5538	1.5884	1.6396	1.6854	1.7270	1.7658

Table 2.—Properties of Superheated Steam—*Continued*

Absolute Pressure, lb. per sq. in.	Temperature, Sat., deg. F.		Superheated Steam, Total Temperature, deg. F. 520	540	560	580	600	620	640	660	680	700	720	740	760	780	800	900	1000
500	466.99	v	1.0267	1.0608	1.0937	1.1251	1.1558	1.1861	1.2153	1.2444	1.2727	1.3009	1.3285	1.3561	1.3831	1.4097	1.4365	1.5655	1.6903
		h	1245.2	1259.1	1272.3	1285.0	1297.3	1309.4	1321.4	1333.2	1344.9	1356.6	1368.1	1379.6	1391.0	1402.3	1413.6	1469.1	1524.8
		s	1.5067	1.5208	1.5338	1.5462	1.5579	1.5692	1.5802	1.5908	1.6012	1.6113	1.6212	1.6309	1.6403	1.6495	1.6586	1.7009	1.7404
600	486.17	v	0.8262	0.8577	0.8876	0.9159	0.9431	0.9695	0.9954	1.0206	1.0452	1.0694	1.0933	1.1169	1.1400	1.1628	1.1855	1.2953	1.4003
		h	1231.7	1247.2	1261.8	1275.5	1288.7	1301.5	1314.1	1326.4	1338.6	1350.6	1362.6	1374.4	1386.2	1397.8	1409.3	1466.1	1522.8
		s	1.4755	1.4913	1.5056	1.5191	1.5316	1.5436	1.5551	1.5662	1.5770	1.5874	1.5977	1.6076	1.6174	1.6268	1.6360	1.6794	1.7196
700	503.04	v	0.6801	0.7107	0.7389	0.7655	0.7905	0.8143	0.8376	0.8602	0.8822	0.9038	0.9250	0.9457	0.9661	0.9864	1.0063	1.1029	1.1929
		h	1216.3	1234.1	1250.4	1265.5	1279.7	1293.3	1306.5	1319.4	1332.0	1344.5	1356.9	1369.1	1381.1	1393.1	1404.9	1463.0	1520.8
		s	1.4457	1.4637	1.4797	1.4945	1.5080	1.5208	1.5329	1.5444	1.5557	1.5665	1.5771	1.5873	1.5973	1.6070	1.6165	1.6608	1.7018
800	518.18	v	0.5681	0.5988	0.6263	0.6515	0.6750	0.6974	0.7189	0.7395	0.7596	0.7791	0.7983	0.8172	0.8358	0.8541	0.8723	0.9577	1.0374
		h	1198.9	1219.3	1237.8	1254.6	1270.1	1284.6	1298.6	1312.1	1325.3	1338.2	1350.9	1363.5	1375.9	1388.2	1400.4	1459.9	1518.8
		s	1.4161	1.4368	1.4551	1.4714	1.4862	1.4998	1.5127	1.5248	1.5365	1.5477	1.5586	1.5692	1.5795	1.5894	1.5992	1.6446	1.6864
900	531.95	v	......	0.5094	0.5369	0.5616	0.5844	0.6057	0.6258	0.6452	0.6639	0.6821	0.6999	0.7171	0.7344	0.7511	0.7675	0.8451	0.9166
		h	......	1202.7	1223.8	1242.7	1259.8	1275.5	1290.3	1304.5	1318.3	1331.7	1344.8	1357.8	1370.6	1383.2	1395.8	1456.8	1516.8
		s	......	1.4100	1.4309	1.4493	1.4656	1.4803	1.4938	1.5067	1.5188	1.5304	1.5417	1.5526	1.5632	1.5734	1.5835	1.6301	1.6727
1000	544.58	v	......	......	0.4635	0.4885	0.5111	0.5317	0.5509	0.5692	0.5870	0.6040	0.6207	0.6369	0.6529	0.6684	0.6837	0.7547	0.8199
		h	......	......	1208.3	1229.7	1248.7	1265.8	1281.6	1296.6	1311.0	1324.9	1338.5	1351.9	1365.1	1378.1	1391.0	1453.6	1514.8
		s	......	......	1.4065	1.4274	1.4455	1.4615	1.4760	1.4895	1.5022	1.5144	1.5260	1.5373	1.5481	1.5588	1.5691	1.6169	1.6603
1100	556.28	v	......	......	0.4016	0.4274	0.4500	0.4705	0.4893	0.5071	0.5239	0.5401	0.5559	0.5712	0.5863	0.6009	0.6152	0.6810	0.7408
		h	......	......	1190.7	1215.2	1236.6	1255.4	1272.5	1288.3	1303.4	1317.9	1332.0	1345.8	1359.4	1372.8	1386.1	1450.4	1512.8
		s	......	......	1.3813	1.4054	1.4257	1.4433	1.4590	1.4733	1.4866	1.4993	1.5113	1.5229	1.5341	1.5450	1.5557	1.6048	1.6491
1200	567.14	v	......	......	......	0.3752	0.3985	0.4189	0.4373	0.4547	0.4710	0.4865	0.5015	0.5162	0.5304	0.5444	0.5578	0.6195	0.6750
		h	......	......	......	1199.1	1223.4	1244.2	1262.7	1279.6	1295.5	1310.6	1325.2	1339.5	1353.5	1367.4	1381.1	1447.2	1510.8
		s	......	......	......	1.3828	1.4058	1.4254	1.4423	1.4576	1.4716	1.4848	1.4972	1.5092	1.5208	1.5321	1.5431	1.5937	1.6388
1300	577.32	v	......	......	......	0.3297	0.3537	0.3746	0.3932	0.4100	0.4260	0.4412	0.4557	0.4699	0.4834	0.4966	0.5095	0.5675	0.6190
		h	......	......	......	1181.0	1208.7	1232.1	1252.4	1270.5	1287.2	1303.0	1318.2	1333.0	1347.5	1361.8	1376.0	1444.0	1508.8
		s	......	......	......	1.3591	1.3855	1.4074	1.4260	1.4423	1.4571	1.4709	1.4839	1.4962	1.5082	1.5199	1.5312	1.5832	1.6292
1400	586.96	v	......	......	......	......	0.3144	0.3361	0.3548	0.3717	0.3874	0.4021	0.4163	0.4299	0.4430	0.4556	0.4678	0.5229	0.5712
		h	......	......	......	......	1192.4	1218.9	1241.3	1260.8	1278.6	1295.2	1311.0	1326.3	1341.3	1356.1	1370.8	1440.8	1506.9
		s	......	......	......	......	1.3643	1.3891	1.4096	1.4272	1.4429	1.4574	1.4709	1.4837	1.4961	1.5082	1.5199	1.5736	1.6204
1500	596.08	v	......	......	......	......	0.2789	0.3017	0.3211	0.3381	0.3536	0.3682	0.3819	0.3951	0.4077	0.4200	0.4318	0.4840	0.5298
		h	......	......	......	......	1174.3	1204.3	1229.3	1250.5	1269.6	1287.1	1303.6	1319.4	1334.9	1350.2	1365.4	1437.6	1504.9
		s	......	......	......	......	1.3420	1.3701	1.3930	1.4122	1.4290	1.4442	1.4583	1.4716	1.4844	1.4969	1.5091	1.5642	1.6120
1600	604.74	v	......	......	......	......	......	0.2706	0.2908	0.3083	0.3239	0.3384	0.3518	0.3646	0.3769	0.3888	0.4002	0.4503	0.4935
		h	......	......	......	......	......	1188.2	1216.2	1239.6	1260.1	1278.6	1295.9	1312.3	1328.3	1344.2	1360.0	1434.3	1502.9
		s	......	......	......	......	......	1.3503	1.3760	1.3971	1.4152	1.4313	1.4461	1.4599	1.4731	1.4860	1.4986	1.5555	1.6042
1700	612.98	v	......	......	......	......	......	0.2422	0.2636	0.2817	0.2975	0.3119	0.3250	0.3376	0.3496	0.3612	0.3721	0.4204	0.4616
		h	......	......	......	......	......	1170.4	1202.0	1227.9	1250.1	1269.8	1288.0	1305.1	1321.7	1338.0	1354.4	1431.1	1500.8
		s	......	......	......	......	......	1.3294	1.3584	1.3818	1.4014	1.4185	1.4340	1.4484	1.4621	1.4755	1.4885	1.5472	1.5968
1800	620.86	v	......	......	......	......	......	......	0.2384	0.2574	0.2736	0.2880	0.3012	0.3135	0.3253	0.3365	0.3472	0.3938	0.4330
		h	......	......	......	......	......	......	1186.3	1215.3	1239.4	1260.6	1279.8	1297.6	1314.8	1331.8	1348.7	1427.8	1498.8
		s	......	......	......	......	......	......	1.3399	1.3661	1.3874	1.4058	1.4222	1.4372	1.4514	1.4652	1.4787	1.5392	1.5897

Table 3.—Steam Table Corrections.

The values ΔH (enthalpy) and ΔN (entropy) are to be *deducted* from the values given in Tables 2 and 3, to make them conform to the International Steam Table Conference tolerances.

Pressure, lb. per sq. in.	Steam Temperature, deg. F.									
	800		850		900		950		1000	
	ΔH	ΔN	ΔH	ΔN	ΔH	ΔN	ΔH	ΔN	ΔH	ΔN
300	2.2	0.0018	2.8	0.0022	3.2	0.0025	3.5	0.0027	4.2	0.0032
600	3.3	.0027	4.0	.0032	4.7	.0037	5.6	.0044	6.7	.0052
900	3.8	.0032	5.0	.0041	6.2	.0050	7.8	.0061	9.4	.0074
1200	4.1	.0037	6.1	.0051	8.2	.0067	10.4	.0082	12.5	.0098
1500	4.1	.0041	7.0	.0062	10.2	.0085	13.1	.0105	15.7	.0124
1800	4.0	.0044	7.9	.0073	12.1	.0105	15.8	.0130	18.7	.0150

Table 4.—Properties of Saturated Liquid and Saturated Vapor.

(Third International Steam Table Conference, 1934)

Temp., deg. C.	Pressure, kg. per sq.cm.	Toler-ance, ±	Specific Volume, cu. cm. per gram				Enthalpy or Total Heat, Int. Calories per gram			
			Liquid	Toler-ance, ±	Vapor	Toler-ance, ±	Liquid	Toler-ance, ±	Vapor	Toler-ance, ±
0	0.006228	0.000006	1.00021	0.00005	206,310	210	0*	0	597.3	0.7
10	0.012513	0.000010	1.00035	0.00010	106,410	110	10.04	0.01	601.6	0.7
20	0.023829	0.000020	1.00184	0.00010	57,824	58	20.03	0.02	605.9	0.6
30	0.043254	0.000030	1.00442	0.00010	32,922	33	30.00	0.02	610.2	0.5
40	0.075204	0.000038	1.00789	0.00010	19,543	19	39.98	0.02	614.5	0.5
50	0.12578	0.00006	1.0121	0.0002	12,045	12	49.95	0.03	618.9	0.5
60	0.20312	0.00010	1.0171	0.0002	7,678.3	7.7	59.94	0.03	623.1	0.5
70	0.31775	0.00016	1.0228	0.0002	5,046.3	5.0	69.93	0.03	627.3	0.5
80	0.48292	0.00024	1.0290	0.0002	3,409.2	3.4	79.95	0.04	631.4	0.5
90	0.71491	0.00036	1.0359	0.0002	2,361.5	2.4	89.98	0.05	635.3	0.5
100	1.03323	Nil	1.0435	0.0002	1,673.2	1.7	100.04	0.05	639.1	0.5
110	1.4609	0.0010	1.0515	0.0004	1,210.1	1.2	110.12	0.06	642.7	0.5
120	2.0245	0.0013	1.0603	0.0004	891.65	0.89	120.25	0.06	646.2	0.5
130	2.7544	0.0016	1.0697	0.0004	668.21	0.67	130.42	0.07	649.6	0.5
140	3.6848	0.0021	1.0798	0.0004	508.53	0.51	140.64	0.07	652.7	0.6
150	4.8535	0.0032	1.0906	0.0004	392.46	0.39	150.92	0.08	655.7	0.7
160	6.3023	0.0042	1.1021	0.0004	306.76	0.31	161.26	0.08	658.5	0.8
170	8.0764	0.0053	1.1144	0.0004	242.55	0.24	171.68	0.09	661.0	0.8
180	10.225	0.007	1.1275	0.0004	193.80	0.19	182.18	0.09	663.3	0.9
190	12.800	0.008	1.1415	0.0004	156.32	0.16	192.78	0.10	665.2	0.9
200	15.857	0.008	1.1565	0.0004	127.18	0.13	203.49	0.10	666.8	0.9
210	19.456	0.008	1.1726	0.0004	104.24	0.10	214.32	0.11	668.0	0.9
220	23.659	0.009	1.1900	0.0004	86.070	0.086	225.29	0.11	669.0	0.9
230	28.531	0.010	1.2087	0.0004	71.483	0.071	236.41	0.12	669.4	0.9
240	34.140	0.012	1.2291	0.0004	59.684	0.060	247.72	0.12	669.4	0.9
250	40.560	0.013	1.2512	0.0004	50.061	0.050	259.23	0.13	668.9	0.9
260	47.866	0.015	1.2755	0.0004	42.149	0.042	270.97	0.18	667.8	0.9
270	56.137	0.017	1.3023	0.0004	35.593	0.036	282.98	0.19	666.0	0.9
280	65.457	0.020	1.3321	0.0004	30.122	0.030	295.30	0.20	663.6	0.9
290	75.917	0.022	1.3655	0.0005	25.522	0.030	307.99	0.20	660.4	0.9
300	87.611	0.024	1.4036	0.0005	21.625	0.035	320.98	0.30	656.1	1.0
310	100.64	0.03	1.4475	0.0005	18.300	0.035	334.63	0.40	650.8	1.2
320	115.12	0.03	1.4992	0.0005	15.438	0.035	349.00	0.50	644.2	1.4
330	131.18	0.04	1.5619	0.0005	12.952	0.035	364.23	0.60	636.0	1.6
340	148.96	0.04	1.6408	0.0005	10.764	0.035	380.69	0.70	625.6	1.8
350	168.63	0.04	1.7468	0.0006	8.802	0.035	398.9	0.8	611.9	2.0
360	190.42	0.05	1.9066	0.0040	6.963	0.040	420.8	0.8	592.9	2.0
370	214.68	0.05	2.231	0.021	4.997	0.100	452.3	1.5	559.3	3.0
371	217.26	0.10	2.297	0.026	4.761	0.100	457.2	1.5	553.8	3.5
372	219.88	0.11	2.381	0.034	4.498	0.110	462.9	2.2	547.1	4.0
373	222.53	0.11	2.502	0.053	4.182	0.120	471.0	3.5	538.9	4.5
374	225.22	0.11	2.79	0.15	3.648	0.120	488.0	5.0	523.3	5.0

Observed values of critical temperature: Mass. Inst. of Tech., 374.11° C.; Reichsanstalt, 374.2±0.1°C.

* By definition.

Table 5.—Specific Volume of Compressed Liquid Water and Superheated Steam.

(Third International Steam Table Conference, 1934)

Values to left and below the heavy line are for compressed liquid water; values to the right and above it are for superheated steam. Of each pair of figures the upper represents the accepted value and the lower the tolerance (±)

Pressure, kg. per sq. cm.	Temperature, deg. C. 0	50	100	150	200	250	300	350	400	450	500	550
	Specific Volume, cu. cm. per gram											
1	1.00016 .00005	1.01210 .00020	1730. 1.	1975. 2.	2216. 2.	2454. 2.	2691. 3.	2928. 3.	3164. 3.	3400. 3.	3636. 4.	3872. 4.
5	0.9999 .0002	1.0119 .0002	1.0432 .0002	1.0906 .0002	433.8 .4	484.1 .5	533.2 .5	581.6 .6	629.6 .6	677.4 .7	725.0 .7	772.5 .8
10	.9997 .0002	1.0117 .0002	1.0431 .0002	1.0902 .0002	210.4 .2	237.6 .2	263.3 .3	288.2 .3	312.7 .3	337.0 .3	361.1 .4	385.1 .4
25	.9989 .0002	1.0110 .0002	1.0422 .0002	1.0893 .0002	1.1556 .0003	89.0 .1	101.1 .1	112.1 .1	122.6 .1	132.7 .1	142.7 .1	152.6 .2
50	.9977 .0002	1.0099 .0002	1.0409 .0002	1.0877 .0002	1.1532 .0003	1.2495 .0004	46.41 .07	53.12 .08	59.05 .09	64.60 .09	69.92 .10	75.10 .12
75	.9965 .0002	1.0088 .0002	1.0397 .0002	1.0861 .0002	1.1508 .0003	1.2452 .0004	27.48 .05	33.22 .07	37.78 .08	41.83 .08	45.62 .09	49.25 .10
100	.9952 .0002	1.0077 .0002	1.0385 .0002	1.0845 .0002	1.1485 .0003	1.2410 .0004	1.3979 .0007	23.03 .05	27.05 .05	30.41 .06	33.45 .07	36.32 .07
125	.9940 .0002	1.0067 .0002	1.0372 .0002	1.0829 .0002	1.1462 .0003	1.2369 .0004	1.3877 .0007	16.66 .03	20.53 .04	23.52 .05	26.14 .05	28.55 .06
150	.9929 .0002	1.0056 .0002	1.0360 .0002	1.0814 .0002	1.1439 .0003	1.2330 .0004	1.3782 .0007	11.98 .02	16.10 .03	18.90 .04	21.25 .04	23.36 .05
200	.9905 .0002	1.0035 .0002	1.0337 .0002	1.0784 .0002	1.1395 .0003	1.2255 .0004	1.3612 .0007	1.671 .002	10.31 .02	13.05 .03	15.11 .03	16.87 .03
250	.9882 .0002	1.0015 .0002	1.0314 .0002	1.0755 .0002	1.1353 .0003	1.2184 .0004	1.3462 .0007	1.604 .002	6.366 .013	9.46 .02	11.39 .02	12.96 .03
300	.9859 .0002	0.9995 .0002	1.0291 .0002	1.0726 .0002	1.1312 .0003	1.2117 .0004	1.3327 .0007	1.557 .002	3.02 .01	6.98 .02	8.90 .02	10.35 .02
350	.9837 .0002	.9975 .0002	1.0269 .0002	1.0698 .0002	1.1272 .0003	1.2054 .0004	1.3207 .0007	1.521 .002				
400	.9814 .0002	.9956 .0002	1.0247 .0002	1.0670 .0002	1.1234 .0003	1.1994 .0004	1.3097 .0007					

The specific volume of the liquid at 4° C, at a pressure of one atmosphere is 1.000027

Table 6.—Enthalpy or Total Heat of Compressed Liquid Water and Superheated Steam.

(Third International Steam Table Conference, 1934)

Values to the left of and below the heavy line are for compressed liquid water; values to the right and above it are for superheated steam. Of each pair of figures the upper represents the accepted value and the lower the tolerance (±)

Pressure, kg. per sq. cm.	Temperature, deg. C.											
	0	50	100	150	200	250	300	350	400	450	500	550
	Enthalpy or Total Heat, Int. Calories per gram											
1	.023 .005	49.97 .03	639.2 .5	663.2 .5	686.5 .6	710.1 1.6	734.0 1.2	758.0 1.2	782.4 1.2	807.2 1.2	832.3 1.2	857.8 2.0
5	.120 .005	50.05 .03	100.11 .05	150.92 .08	681.9 1.0	706.7 1.0	731.5 1.2	756.1 1.2	780.8 1.2	805.9 1.2	831.3 1.2	856.9 2.0
10	.240 .005	50.15 .03	100.20 .05	151.00 .08	675.1 1.0	702.1 1.1	728.0 1.2	753.5 1.2	778.9 1.2	804.5 1.2	830.1 1.2	855.9 2.0
25	.599 .005	50.45 .03	100.46 .05	151.21 .08	203.6 .1	687.8 1.1	718.0 1.2	746.3 1.2	773.3 1.2	800.0 1.2	826.5 1.2	852.6 2.0
50	1.20 .01	50.96 .03	100.90 .05	151.58 .08	203.8 .1	259.2 .1	698.4 1.2	732.9 1.2	763.1 1.2	791.6 1.2	819.9 1.2	847.3 2.0
75	1.79 .01	51.46 .03	101.34 .05	151.95 .08	204.1 .1	259.2 .1	672.6 1.2	717.6 1.2	752.1 1.2	783.2 1.2	813.1 1.2	841.8 2.0
100	2.39 .01	51.96 .03	101.78 .05	152.32 .08	204.3 .1	259.2 .1	320.5 .3	699.5 1.2	740.0 1.3	774.5 1.3	806.0 1.3	836.1 2.0
125	2.98 .01	52.46 .03	102.22 .05	152.69 .08	204.6 .1	259.3 .1	319.9 .3	676.7 1.2	726.9 1.3	765.2 1.3	799.1 1.3	830.3 2.0
150	3.57 .01	52.96 .03	102.65 .05	153.06 .08	204.8 .1	259.3 .1	319.3 .3	646.8 1.5	712.1 1.3	755.3 1.3	791.8 1.3	824.4 2.0
200	4.74 .01	53.96 .03	103.57 .05	153.82 .08	205.2 .1	259.4 .1	318.4 .3	393.1 .8	676.5 2.0	733.4 2.0	776.0 2.0	812.0 2.5
250	5.90 .01	54.96 .03	104.46 .05	154.57 .08	205.8 .2	259.5 .2	317.6 .3	387.6 .8	622.5 2.5	707.5 2.5	758.8 2.5	798.9 3.0
300	7.05 .01	55.96 .03	105.35 .05	155.33 .08	206.2 .3	259.7 .3	317.0 .3	384.0 .8	524.5 3.0	677.5 2.5	739.7 2.5	

The following conversion factors were used in Tables 4, 5, and 6:

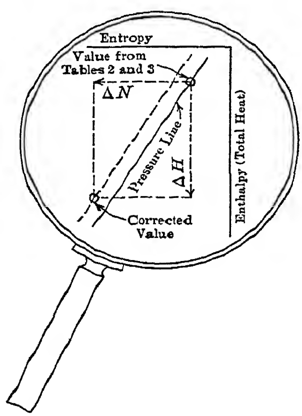
1000 Int. calories = 3.9683 B.t.u.
10 B.t.u. = 2519.96 Int. cal.
1 cu. ft. = 28,316.8 cu. cm.
1 cu. ft. per lb. = 0.062428 cu. m. per kg.
1 cu. m. per kg = 16.0185 cu. ft. per lb.

Pressure Conversion Factors

Units	Atm.	Kg. per sq. cm.	Lb. per sq. in.	Bar	Mm. Hg
1 atmosphere =	1	1.033228	14.6959	1.013250	760
1 kg. per sq. cm. =	0.967841	1	14.2233	0.980665	735.559
10 lb. per sq. in. =	0.68046	0.70307	10	0.689476	517.149
1 bar =	0.986923	1.019716	14.5038	1	750.062
1 meter Hg =	1.31579	1.35951	19.3368	1.333224	1000

Electric Co., who based them on values of enthalpy (total heat) and entropy of saturated vapor furnished by N. S. Osborne of the U. S. Bureau of Standards, which conform to the International tolerances, and on values of enthalpy of superheated steam contained in Table 6.

Some of the values of enthalpy and entropy as given in Tables 1 and 2 are higher than are permitted by the conference tolerances. The corrections ΔH (enthalpy) and ΔN (entropy) as given in Table 3 should be *deducted* from the values in Tables 1 and 2, to bring them into conformity with the International tolerances. Fig. 1 shows the application of the corrections to the Mollier diagram.



. 1. Application of Correction to Mollier Diagram

HEAT ADDED BY THE BOILER-FEED PUMP.—Feedwater at a pressure and temperature much below that in the boiler frequently is delivered to the boiler-feed pump and raised to boiler pressure. The work per lb. of water done by the feed pump, $W_f = v_s(P_b - P_s)/778.6$ B.t.u., where P_b = boiler pressure, lb. per sq. ft.; P_s = absolute pump suction pressure in lb. per sq. ft.; v_s = volume, cu. ft., of 1 lb. water at the temperature of suction. This formula assumes that water is incompressible, which is approximately correct for usual pressure ranges. The work of the boiler-feed pump is of a small order at pressures below 350 lb. per sq. in., but is significant at high boiler pressures. For example, let water enter the feed pump at 220° F. and 20 lb. per sq. in., abs.; the work of the feed pump lifting 1 lb. of water under these conditions to boiler pressure is given in Table 7.

TEMPERATURE-ENTROPY DIAGRAM OF WATER AND STEAM.—Changes taking place in steam expansions or compressions may conveniently be represented on a temperature-entropy diagram. The line OA, Fig. 2, is the origin, *i.e.*, 32° F., from which entropy is measured on horizontal lines, and the line OS is the line of zero temperature, absolute. The diagram represents the changes in the state of 1 lb. of water due to the addition or subtraction of heat or to changes in temperature. Any point on the diagram is called a *state-point*. A is the state of 1 lb. of water at 32° F. or 491.6° abs., B the state at 212° F., and C at 392° F., corresponding to about 226 lb. per sq. in. absolute pressure. K is the state point at the critical temperature 706.1° F. At 212° F. the area $OABb$ is the heat added to the water, and Ob is the increase of entropy. At 392° F., $bBCc$ is the further addition of heat to the water, and the entropy at C, measured from OA, is Oc. The two quantities added are nearly the same, but the second increase of entropy is the smaller, since the mean temperature at which it is added is higher. If Q = the quantity of heat added, and T_1 and T_2 are, respectively, the lower and the higher temperatures, the addition of entropy from 32° F. to 212° F., $s_f = \bar{c}_p \log_e (T_2/T_1) = 0.3119 = Ob$, where $\bar{c}_p$ is the mean specific heat of water over this temperature range. Accurate values of the entropy of water, taking into account the variation in specific heat, will be found in the Steam Tables.

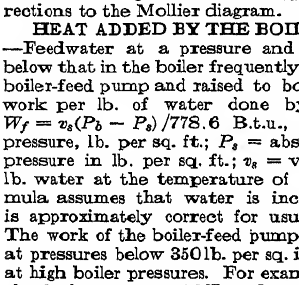


Fig. 2. Temperature-Entropy Diagram

Let the 1 lb. of water at the state B have heat added to it at the constant temperature of 212° F. until it is evaporated. The quantity of heat added will be the latent heat of

Table 7.—Work of Boiler-Feed Pump Lifting 1 Lb. of Water to Boiler Pressure.
Water enters pump at 220° F. and 20 lb. per sq. in., abs.

Absolute Boiler Pressure, lb. per sq. in.	Work, B.t.u. per lb.	Absolute Boiler Pressure, lb. per sq. in.	Work, B.t.u. per lb.	Absolute Boiler Pressure, lb. per sq. in.	Work, B.t.u. per lb.
40	0.061	750	2.25	3000	9.20
250	.710	1500	4.57	4000	12.29
450	1.328	2000	6.12	6000	18.47

evaporation at 212° (see Steam Table) or h_{fg} = 970.2 B.t.u., and it will be represented on the diagram by the rectangle *bBFf*. Dividing by T_1 = 671.6, the absolute temperature, gives s_{fg} = 1.4446 = *BF*. Adding s_f = 0.3119 gives s_g = 1.7564, the entropy of 1 lb. of steam at 212° F. measured from water at 32° F. = *Of*.

In like manner if we take h_{fg} = 833.0 for steam at 392° F., s_{fg} = 0.9782 = *CE*, and s_f = entropy of water at 392° F. = 0.5567, = *Oc*, the sum s_g = 1.5349 = *Oe*.

E is the state point of dry saturated steam at 392° F. and *F* is the state point at 212° F. The line *EFG* is the *line of saturated steam* and the line *ABC* the *water line*. The line *CE* represents the increase of entropy in the evaporation of water at 392° F. If entropy *CD* only is added, or *cCDd* of heat, then a part of the water will remain unevaporated, *viz.*, the fraction *DE/CE* of 1 lb. The state-point *D* thus represents wet steam having a dryness fraction of *CD/CE*.

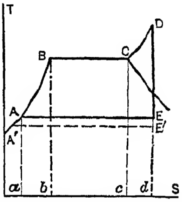


Fig. 3. Rankine Cycle

K is the *critical state for water* at a temperature of 706.1° F. and a pressure of 3226 lb. per sq. in., abs. To the left of *K* the substance is water; above *K* to the right, it becomes superheated steam. At pressures above 3226 lb. per sq. in., abs., the water has no latent heat but passes directly to superheated steam as at *K*. Line *AL* represents (but not to scale) the states for a pressure of 6000 lb. per sq. in., abs.

If steam having a state point *E* is expanded adiabatically at constant entropy to 212° F. its state-point is then e_1, having the same entropy as at *E*, a total heat less by the amount represented by the area $BCEe_1$, and a dryness fraction Be_1/BF. If it is expanded while remaining saturated, heat must be added equal to *eEFf*, and the entropy increases by *ef*.

If heat is added to the steam at *E*, the temperature and the entropy both increase, the line *EH* representing the *superheating*, and the area *EHhe* is the heat added. If from the state point *H* the steam is expanded adiabatically at constant entropy, the state-point follows the line *HJ* until it cuts the line *EFG*, when the steam is dry saturated, and if it crosses this line the steam becomes wet.

If the state-point follows a horizontal line to the left of line *EFG*, and to the water line *ABC* it respresents *condensation at a constant temperature*, the amount of heat rejected being shown by the area under the horizontal line down to line, *Og*. If heat is rejected at a decreasing temperature, corresponding with the decreasing pressure at release in a steam engine, or condensation in a cylinder at a decreasing pressure, the state-point follows a curved line to the left, as shown in the dotted curved line on the diagram.

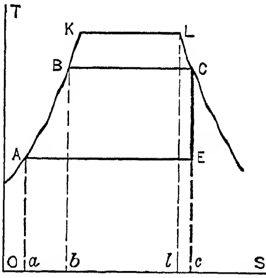


Fig. 4. Effect of Increased Pressure

In practical calculations with the entropy-temperature diagram it is necessary to have at hand tables or charts of entropy, enthalpy or total heat, etc., such as are given in Keenan's Steam Tables and Mollier Diagram, Goodenough's Properties of Steam and Ammonia, Marks and Davis's Steam Tables, and other works.

REVERSIBLE ADIABATIC EXPANSION on the temperature-entropy diagram is a vertical line at constant entropy. When it is necessary to make exact calculations of the conditions at the end of such expansion this can be done by means of the entropies. Let state 1 be the initial condition and state 2 the final condition after adiabatic expansion. With wet steam $s_{f_1} + x_1 s_{fg1} = s_{f2} + x_2 s_{fg2}$ from which x_2 can be found and the enthalpy or total heat at state 2 calculated from $h_2 = h_{f2} + x_2 h_{fg2}$. The heat drop, ($h_1$ —) then can be found. With superheated steam expanding into the wet region $s_1 = s_{f2} + x_2 s_{fg2}$. Then x_2 can be found and h_2 calculated as above.

STEAM CYCLES.—As stated on p. 3–78 the ideal cycle for a heat engine is the Carnot cycle, which is represented by a rectangle on the temperature-entropy diagram.

The Rankine or Clausius Cycle is shown in Fig. 3. The feedwater enters at temperature *A* and has the quantity of heat *ABba* added to it at constant pressure, increasing its temperature to the boiling point *B*. Then the latent heat *BCcb* may be added and finally the superheat *CDdc*. The steam expands adiabatically in an ideal engine along the line of constant entropy *DE* and exhausts to a condenser discharging the heat *EdaA* to the cooling water. The heat available for work is represented by the area *ABCDEA*

The efficiency of the cycle is the ratio of the heat available for work to the heat supplied, *i.e.*, area *ABCDEA/ABCDda*. This cycle is the usual standard of comparison for straight expansion steam engines and turbines.

The efficiency of the Rankine cycle can be increased by increasing the heat available for work in a greater ratio than the increase of heat supplied. This may be done by lowering the exhaust temperature. Obviously from Fig. 3, when the exhaust temperature is lowered to *A'E'*, less heat is rejected to the condenser and more heat is available for work. Hence when low absolute pressure or vacuum can be utilized it is desirable to lower the vacuum temperature as close as possible to that of the coldest available cooling water. If the pressure is increased without superheat, it would appear from Fig. 4 that efficiency increases as the pressure rises from *BC* to *KL*. Emswiler points out that the enthalpy or total heat of saturated steam increases up to 400 lb. per sq. in., abs., as shown in the steam tables and then decreases to a minimum at the critical pressure of 3226 lb. per sq. in., abs. The thermal efficiency of the Rankine cycle increases to a maximum with saturated steam at 2000 lb. per sq. in., abs., and then decreases slightly. But increased pressure leads to lower quality steam at the lower stages of expansion, and in turbines this causes internal losses which offset the gain by very high initial pressures.

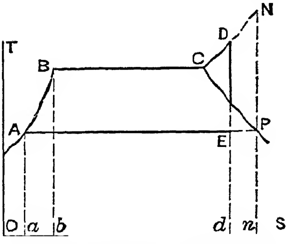


Fig. 5. Effect of Increased Steam Temperature

Increased steam temperature at moderate pressure as shown by Fig. 5 by the Rankine cycle *ABCNPA*, gives only a small increase in thermal efficiency. In turbines, such an increase in temperature leads to a substantial increase in efficiency, as less of the expansion takes place in the saturated section and this leads to a decrease in internal losses in the turbine. Obviously an increase in both pressure and superheat would tend to increase cycle efficiency.

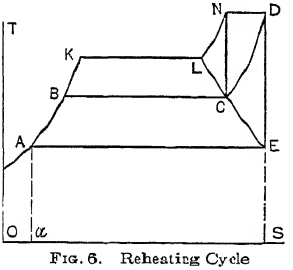


Fig. 6. Reheating Cycle

Reheating Cycle.—The use of high pressures with moderate superheat leads to high internal losses in a turbine. This is overcome by *resuperheating* as shown in Fig. 6 where pressure is increased from *BC* to *KL* but superheat temperature remains constant at *ND*. The steam after expansion from *N* to *C* is resuperheated again to the original temperature *D* and expanded to *E*. This is known as the *Reheating Cycle*. The theoretical advantages, as measured by cycle efficiencies, are small, but the practical gains due to decreased internal losses in steam turbines and engines are considerable.

Extraction or Regenerative Cycle.—Regenerative heating of the feedwater is another method of improving cycle efficiency. In Fig. 7 assume that at every infinitesimal step in expansion there is a small portion of steam withdrawn from the cycle to preheat the feed. The expansion line *CR* represents this condition. *CR* is parallel to *AB* for the heat added to the feed *CRrc* must equal *ABba*. But by geometry *ABCR* equals *BCEP*, which would be the energy available for work in a Carnot cycle. Also the heat supplied *ABCRra* equals *BCcb*. Hence theoretically with saturated steam the *regenerative cycle* can equal the maximum ideal efficiency of any engine, that is, of the Carnot cycle between the temperature limits. With superheat, the regenerative cycle is theoretically not so efficient, as it cannot utilize all heat at the highest temperature. This easily applied and efficient cycle should be used in every steam turbine power plant.

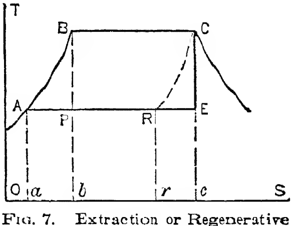


Fig. 7. Extraction or Regenerative Cycle

Many stations now operate on a combination of these last two cycles known as the *regenerative-reheating cycle*. Both extraction feed heating and interstage reheating are employed.

BINARY-VAPOR CYCLES.—The critical pressure of steam is 3226 lb. per sq. in., abs., with a temperature of 706.1° F. Several attempts have been made to use a fluid with higher range boiling points superimposed upon the regular steam cycle forming a *Binary-*

vapor cycle. The employment of the mercury vapor–steam cycle provides one of the most efficient means of generating power from the combustion of fuels.

Mercury-vapor-Steam Cycle.—Fig. 8 illustrates the mercury-vapor–steam cycle on a temperature-entropy diagram. The mercury vapor boils with 165 lb. per sq. in. gage pressure at 999.6° F., and is expanded isentropically to 28 in. vacuum. In practice this would be done in a mercury turbine and useful work developed. The mercury condenses at 28 in. vacuum at 456.8° F. in a condenser-steam boiler which transfers the exhaust heat from the mercury to the steam. The saturated steam is generated at 360 lb. per sq. in., gage, 438.5° F., by the condensation of the mercury vapor. For its efficient use in the steam turbine, the steam is superheated by flue gases from the mercury boiler to 700° F. and expanded to 29 in. vacuum, using regenerative extraction for feedwater heating. On account of the low heat values per pound of mercury, it is necessary to use about 10 lb. of mercury per 1 lb. of steam in the mercury vapor-steam cycle. With isentropic

Table 8.—Properties of Saturated Mercury Vapor*

For calculations involving superheated mercury vapor, the mean specific heat at constant pressure, c_p, may be taken as 0.0248.

Abs. Pressure, lb. per sq. in. p	Temp., deg. F. t	Specific Volume, cu. ft. per lb. v_g	Weight, lb. per cu. ft. $1/v_g$	Enthalpy or Total Heat: Sat. liquid h_f	Enthalpy or Total Heat: Evaporation h_{fg}	Enthalpy or Total Heat: Sat. vapor h_g	Entropy: Sat. liquid s_f	Entropy: Evaporation s_{fg}	Entropy: Sat. vapor s_g
0.49 (29 in. vac.)	414.0	94.63	0.01057	14.2	127.9	142.1	0.0214	0.1464	0.1678
0.735 (28.5 in. vac.)	438.4	64.64	.01547	15.2	127.3	142.5	.0225	.1418	.1643
0.98 (28 in. vac.)	456.8	49.37	.02025	15.9	126.9	142.8	.0232	.1385	.1617
1.2	470.0	40.83	.02449	16.3	126.7	143.0	.0238	.1362	.1600
1.4	480.5	35.37	.02827	16.7	126.4	143.1	.0242	.1345	.1587
1.6	489.6	31.21	.03204	17.1	126.2	143.3	.0245	.1330	.1575
1.8	497.7	27.94	.03579	17.4	126.0	143.4	.0248	.1317	.1565
2.0	505.2	25.32	.03949	17.6	125.9	143.5	.0251	.1305	.1556
3.0	535.4	17.34	.05767	18.8	125.2	144.0	.0263	.1258	.1521
4.0	558.0	13.26	.07540	19.6	124.7	144.3	.0271	.1226	.1497
5.0	576.2	10.77	.09281	20.3	124.3	144.6	.0278	.1200	.1478
6.0	591.4	9.096	.10993	20.9	124.0	144.9	.0283	.1179	.1462
7.0	605.0	7.882	.12686	21.4	123.7	145.1	.0288	.1162	.1450
8.0	616.8	6.963	.14361	21.8	123.4	145.2	.0292	.1147	.1439
9.0	627.5	6.245	.16013	22.2	123.2	145.4	.0296	.1133	.1429
10.0	637.3	5.661	.17664	22.6	123.0	145.6	.0299	.1121	.1420
15.0	676.5	3.892	.25691	24.0	122.1	146.1	.0312	.1075	.1387
20.0	706.2	2.983	.33523	25.1	121.5	146.6	.0322	.1042	.1364
25.0	730.4	2.429	.41169	26.1	120.9	147.0	.0330	.1016	.1346
30.0	750.9	2.053	.48709	26.8	120.5	147.3	.0336	.0995	.1331
35.0	769.0	1.782	.56132	27.5	120.1	147.6	.0342	.0977	.1319
40.0	784.8	1.576	.63440	28.1	119.7	147.8	.0346	.0962	.1308
45.0	799.3	1.415	.70686	28.6	119.4	148.0	.0351	.0948	.1299
50.0	812.5	1.284	.77881	29.1	119.1	148.2	.0355	.0936	.1291
55.0	824.6	1.177	.84983	29.6	118.8	148.4	.0358	.0925	.1283
60.0	836.1	1.087	.92038	30.0	118.6	148.6	.0361	.0915	.1276
65.0	847.0	1.010	.99009	30.4	118.4	148.8	.0365	.0906	.1271
70.0	856.6	0.9436	1.0597	30.8	118.1	148.9	.0367	.0898	.1265
75.0	866.0	0.8859	1.1287	31.1	117.9	149.0	.0370	.0890	.1260
80.0	874.8	0.8349	1.1977	31.4	117.8	149.2	.0373	.0882	.1255
85.0	883.7	0.7901	1.2656	31.8	117.5	149.3	.0375	.0875	.1250
90.0	891.6	0.7497	1.3338	32.0	117.4	149.4	.0377	.0869	.1246
100.0	906.9	0.6811	1.4682	32.6	117.0	149.6	.0381	.0857	.1238
110.0	921.1	0.6242	1.6020	33.2	116.7	149.9	.0385	.0845	.1230
120.0	934.4	0.5767	1.7340	33.7	116.4	150.1	.0389	.0835	.1224
130.0	946.7	0.5360	1.8656	34.1	116.2	150.3	.0392	.0826	.1218
140.0	958.3	0.5012	1.9952	34.6	115.9	150.5	.0395	.0818	.1213
150.0	969.4	0.4706	2.1249	34.9	115.7	150.6	.0398	.0809	.1207
160.0	979.9	0.4438	2.2531	35.4	115.4	150.8	.0401	.0802	.1203
170.0	989.9	0.4200	2.3807	35.7	115.2	150.9	.0403	.0795	.1198
180.0	999.6	0.3990	2.5062	36.1	115.0	151.1	.0406	.0788	.1194

* Abstracted from Properties of Mercury Vapor, by Lucian A. Sheldon, General Electric Company, 1927.

expansion of both mercury vapor and steam, and with extraction feed heating in the cycles shown in Fig. 8, an efficiency of 58.8% may be obtained. The maximum possible Carnot cycle efficiency between the temperature limits of 999.6° F. and 79° F. is 63.4%. Higher efficiencies can be developed on this combined cycle than on any of the preceding cycles.

The properties of mercury vapor are given in Table 8.

Diphenyl and Diphenyl-oxide Cycles.— Certain organic substances such as diphenyl and diphenyl-oxide have been proposed for binary cycles besides mercury. Diphenyl-oxide with additions of naphthalene to regulate the melting point is sold commercially as Dowtherm by the Dow Chemical Co., Midland, Mich. The naphthalene appears to have little effect on the thermal properties of the mixture beyond altering the point of solidification. Table 9 gives certain of the thermal properties of diphenyl-oxide. Above 750° F. diphenyl-oxide slowly decomposes to other organic compounds. Diphenyl has somewhat lower boiling temperature than diphenyl-oxide and also a slightly lower enthalpy or total heat per lb. for the same absolute pressures, although the critical temperature is somewhat higher.

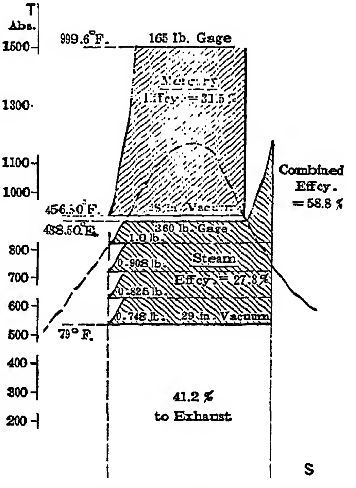


FIG. 8. Temperature-Entropy Diagram of a Mercury-vapor-Steam Cycle

SUPERSATURATED STEAM. (Ewing's Thermodynamics for Engineers.) — Callendar has shown that the adiabatic expansion of superheated steam follows closely the equation $P(V - b)^{1.3} = constant$, where P = pressure, lb. per sq. ft., abs.; V = specific volume, cu. ft. per lb. at pressure P; b = volume of 1 lb. of water at 32° F., *i.e.*, 0.017 cu. ft. Ordinarily b may be neglected without serious error, except at high pressures.

The usual theory of heat assumes that isentropic expansion of saturated steam takes place in thermal equilibrium, *i.e.*, condensation proceeds in the wet region as indicated on a Mollier diagram when expansion takes place at constant entropy. But such condensation takes considerable time, for droplets must form and grow in the mass of the steam. Consequently, sudden expansion of a saturated vapor may produce a temporary condition in which the mass continues to expand as superheated steam without any condensation taking place. This condition is called *supersaturation*. The density of the vapor in this state is abnormal, and higher than the density of saturated vapor at the actual pressure. The temperature at the end of expansion is lower than the temperature of saturation at that pressure and the vapor is said to be under-cooled. The supersaturated condition is not stable and disappears through condensation of part of the vapor. The temperature of the remaining mass is raised by the latent heat given off

Table 9.—Properties of Saturated Diphenyl-oxide Vapors

Temp., deg. Fahr.	Pressure, lb. per sq. in., abs.	Enthalpy or Total Heat Content, B.t.u. per lb.			Specific Heat of Liquid	Density, lb. per cu. ft.	
		Liquid	Latent	Total		Liquid	Vapor
81	....	0	146	146	0.39	......	......
230	0.14	63	141	204	0.46	62.2	0.003
280	0.47	87	138	225	0.49	60.7	0.008
320	1.1	108	136	244	0.52	59.4	0.019
400	4.0	151	129	280	0.57	57.2	0.068
450	8.0	180	124	304	0.60	55.9	0.14
496	14.7	208	120	328	0.62	54.8	0.26
500	15.3	211	119	330	0.63	54.7	0.28
550	25	243	114	357	0.65	52.8	0.48
600	42	277	108	385	0.66	50.9	0.88
650	68	310	102	412	0.67	49.1	1.29
700	100	343	95	438	0.68	47.3	1.7
750	146	377	86	464	0.69	44.8	2.5
950	Critical temperature						

during condensation until thermal equilibrium is restored. Supersaturation, therefore, may be assumed to occur when steam is expanded through a nozzle or orifice and the saturation line is passed during expansion. Such expansion can be represented by the equation $P(V - b)^{1.3} = constant$ as in the case of superheated steam.

Effect of Supersaturation.—The primary effect of supersaturation in nozzles is to increase the discharge of saturated steam by about 5% for a given throat area as compared with that usually calculated for saturated steam (see Callendar's Steady Flow of Steam through a Nozzle or Throttle, Inst. M. E., 1915). The secondary effect is to cause an increase of entropy and volume after passing the throat when the steam in the nozzle becomes wet. Applying the supersaturation theory, the critical pressure in the throat of a nozzle becomes $0.5457P_1$, and the flow of steam through any nozzle becomes $W = 0.3155A\sqrt{P_1/V_1}$, where W = lb. of steam per sec.; A = throat area, sq. in.; P_1 = initial pressure lb. per sq. in., abs., and V_1 = specific volume, cu. ft. per lb., at initial condition at pressure P_1. Since condensation requires time, it is doubtful if any takes place up to the nozzle throat, or if equilibrium is fully restored by the time the steam leaves the nozzle. Martin (A New Theory of the Steam Turbine, *Engg.*, vol. cvi, 1918) holds to the view that the steam is never in thermal equilibrium in a steam turbine until the condenser is reached, and he has worked out a new analysis of steam turbine performance on this basis. He also presents a new steam chart with lines representing supersaturated conditions. It also can be shown that there is about 6% less heat available to do work from that portion of the expansion below the saturation line when expansion is assumed to continue in the adiabatic supersaturated condition at constant entropy than if it took place in thermal equilibrium. This decrease in available energy usually is not considered in steam turbine design, but may be a factor in causing the lowered stage efficiencies experienced with saturated steam. For further information, see Callendar's Properties of Steam, Ewing's Thermodynamics for Engineers, Stodola's Steam Turbines, Goudie's Steam Turbines, and Martin's paper referred to above. See also Martin's later paper, The Proportioning of Steam Turbine Blading, *Engg.*, Jan. 6, 13, 20, 27, and Feb. 3, 1922; The Supersaturation Limit, *Engg.*, May 18, 1923; Powell, Supersaturation of Steam and its Influence on Some Problems of Steam Engineering, *Engg.*, June 7, 21, 1929; Yellot, Supersaturated Steam, *Trans.* A.S.M.E., FSP–56–7, 1934.

2. THE MOLLIER DIAGRAM (Keenan's)

CONSTRUCTION.—(See Fig. 9.) The enthalpy or total heat of 1 pound of steam above 32° F. is plotted as ordinates, and entropies above 32° F. form abscissae. Lines of constant absolute pressure in lb. per sq. in. slope up from left to right. In the low-pressure region at the right-hand side, dotted lines represent absolute pressures in inches of mercury, and are found convenient for exhaust steam calculations. Below the saturation line, curves of constant moisture content, in percent, slope down from left to right. Above the saturation line are shown lines of constant temperature and lines of constant superheat, both in deg. F. Heat quantities may be found directly from the diagram. For instance, the enthalpy or total heat at 650 lb. per sq. in. absolute, 860° F. from the chart = 1442 B.t.u. at entropy 1.6536. Again the total heat at 1 in. of mercury absolute and 89% quality (11% moisture) = 979.5 B.t.u.

NOTE.—A Mollier diagram about 20″ × 32″ accompanies Keenan's Steam Tables. Extra copies can be procured from the American Society of Mechanical Engineers.

THROTTLING EFFECTS.—In the phenomena of throttling or wire-drawing the enthalpy or total heat per lb. of steam is the same in the final as in the initial state, providing no energy has gone to increase velocity and no heat is lost by radiation. A line of constant total heat represents throttling on the Mollier diagram. Thus, if steam at 650 lb. per sq. in., abs., 860° F., is throttled by the turbine governor to 400 lb. per sq. in., abs., its condition on the constant total heat line for 1442 B.t.u. at 400 lb. per sq. in., abs., is 845° F.

The *quality* of slightly wet steam can be readily determined on the Mollier diagram from throttling calorimeter readings. Given: steam line pressure, 125 lb. per sq. in., gage (140 lb. per sq. in., absolute); atmospheric pressure, 14.696 lb. per sq. in., abs., in the calorimeter; temperature in the calorimeter, 280° F. From the chart, enthalpy or total heat in calorimeter at 14.696 lb. per sq. in., abs., and 280° F. = 1182.7 B.t.u., which total heat line representing throttling intersects the 140-lb. absolute pressure line at 1.1% moisture, giving 98.9% quality.

The heat available on the *Rankine cycle* or *heat drop* results from isentropic expansion, which follows a line of constant entropy. Thus, a turbine receives steam at 450 lb. per sq. in., abs., 750° F., and exhausts at 29 in. vacuum. Total heat at 450 lb., abs., 750° F., h_1 = 1387.8 B.t.u.; s = 1.6488. At 1 in. abs., s = 1.6488, h_2 = 885.5 B.t.u. Heat avail-

able on the Rankine cycle or the heat drop, $(h_1 - h_2) = 1387.8 - 885.5 = 502.3$ B.t.u. per lb. of steam.

Heat Added as Reheat in a reheating turbine also is readily found. Thus steam leaves the high-pressure section of the unit at 120 lb. per sq. in., abs., 370° F., and is reheated with a 10 lb. per sq. in. pressure drop in the reheater to 700° F. Total heat at

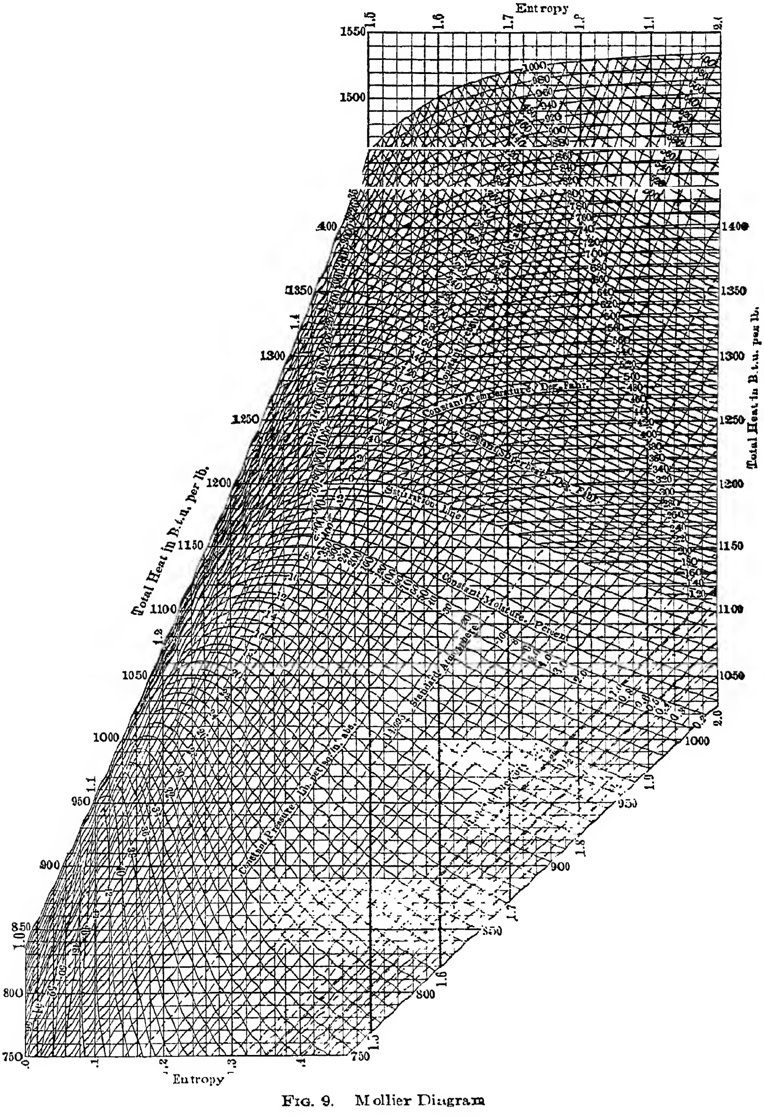


FIG. 9. Mollier Diagram

120 lb., abs., 370° F., = 1207.0 B.t.u. Total heat at 110 lb., abs., 700° F., = 1377.7 B.t.u. Heat added by reheater = 1377.7 − 1207.0 = 170.7 B.t.u. per lb.

The Condition of the Exhaust Steam can be found quickly on a Mollier diagram. In the case of a steam engine, the heat per pound to exhaust

$$h_e = (1 - L)h_1 - (\text{I.Hp.} \times 2543/W),$$

where h_1 = total heat per pound at throttle; L = external heat losses, as radiation, piston- and valve-rod leakage and conduction, etc., expressed as a decimal of the initial total heat; I.Hp. = indicated horsepower; W = total steam per hour, lb. L may be assumed as 0.01 for reciprocating engines. When h_e is determined, the quality can be found by locating the intersection of the total heat line for h_e and the absolute exhaust pressure line.

3. FLOW OF STEAM IN PIPES *

Formulas in general use, expressing the pressure drop along a pipe in which a fluid is flowing, are mainly empirical. They refer to a particular fluid, and do not take into account the surface condition of the pipe interior. Examples of such empirical formulas (see earlier editions of this book) are those of Babcock, Fritzsche, Unwin, Broido and others for steam; Hazen and Williams for water; and Unwin, Taylor and others for air.

The general formula for pressure drop in isothermal turbulent flow is

$$\Delta p = f(L/D)(\rho v^2/2g), \qquad [1]$$

where Δp = pressure drop; f = friction factor; L = length of pipe; D = diameter of pipe; ρ = density of the fluid; v = velocity of the fluid; g = acceleration due to gravity, all expressed in a homogeneous system of units.

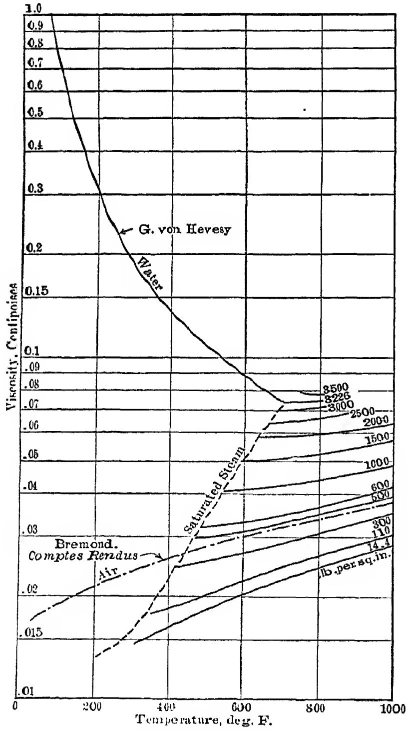


Fig. 10. Viscosity of Steam and Water

In some of the empirical formulas, as, for example, Babcock's for steam and Unwin's for air, it is recognized that f varies with the diameter of the pipe, both containing the term $\{1 + (3.6/d)\}$, in which d is the diameter of the pipe in inches. Still others recognize the effect of velocity, as Fritzsche's formula for friction factor, which contains the term $w^{-1/7}$, in which w denotes the weight flowing per second. The difficulty in using empirical formulas lies in the limitations of application, any change of fluid requiring a modification of the friction factor.

A rational method based on dimensional analysis (see Eshbach, Handbook of Engineering Fundamentals, Section 3, Vol. 1 of this series) has shown the friction factor to be a function of the density, velocity and viscosity of the fluid, as well as the diameter of the pipe. Therefore, a plot of friction factor against Reynolds number $\rho v d/\mu$ is applicable to any fluid, since all the physical properties affecting flow have been taken into account. μ = viscosity. However, the condition of the interior surface of the flow channel also has an effect on the friction factor as does the type of flow, whether viscous or turbulent.

A comprehensive study and correlation of all the available test data were undertaken by Mr. R. J. S. Pigott (*Mech. Engg.*, Aug., 1933). The fluids were brine, water, oil, air, steam and gas, and the pipe materials were glass, lead, tin, brass, steel (plain and riveted), and cast iron (plain and tarred). The results of the study are given on p. 4–59, and are

* Contributed by Robert K. Behr.

recommended for general use in the solution of flow problems. The type of flow may be viscous or turbulent, and the nature of the fluid flowing is immaterial, since its characteristics are taken into consideration in the solution of the problem.

The procedure is as follows: From Table 16, p. 4–60, select the curve to be followed by means of the pipe material and the internal diameter in inches. Solve for the Reynolds number. See p. 4–57. Enter Fig. 4, p. 4–60, or Fig. 11, below with Reynolds number; proceed vertically to the curve selected, and the ordinate is the friction factor.

The Reynolds number $\rho VD/\mu$ is a pure number, dimensionless, and, therefore, can be obtained in any system of consistent units. If the ft.-lb.-sec. system is used, the formulas for pressure drop in turbulent flow are:

Circular pipes
$$\left\{\begin{array}{l}\Delta p = 0.000108\, f(L/D)\rho v^2 \text{ lb. per sq. in.} \qquad [2]\\ \Delta p = 0.001295\, f(L/d)\rho v^2 \text{ lb. per per sq. in.} \qquad [3]\end{array}\right.$$

Any shape
$$\Delta p = 0.000027\, f(L/M)\rho v^2 \text{ lb. per sq. in.} \qquad [4]$$

where Δp = pressure drop, lb. per sq. in.; f = friction factor, a number; L = length of pipe, ft.; ρ = density of fluid, lb. per cu. ft.; v = mean velocity, ft. per sec.; D = internal diameter, ft.; d = internal diameter, in.; M = mean hydraulic radius, ft.; μ = absolute viscosity, lb. per ft. sec.

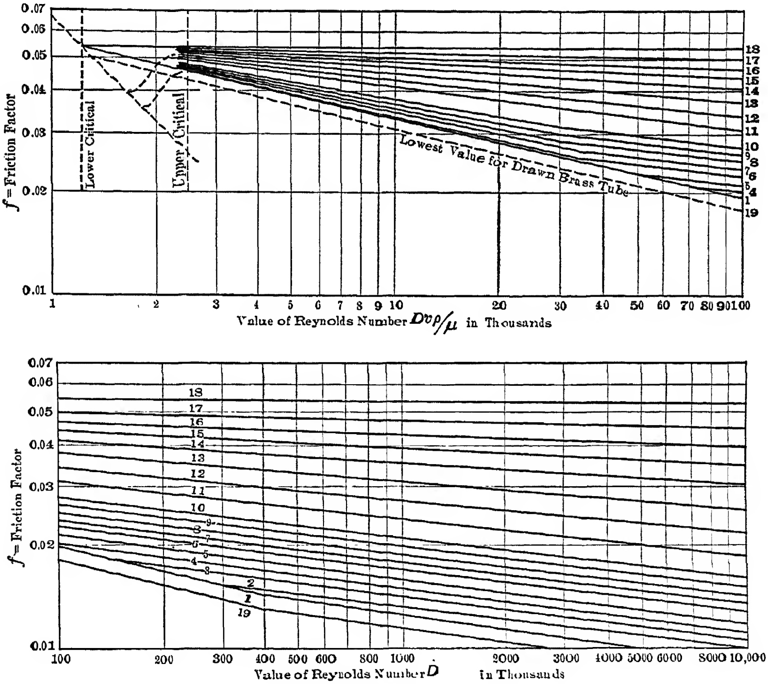


FIG. 11. Friction Factors and Reynolds Numbers

Usual velocities of steam in pipe lines are (S. Crocker, *Power*, June 6, 1926): Saturated steam, 0 to 15 lb. per sq. in. gage pressure, for heating service, 4000–6000 ft. per min.; saturated steam, 50 lb. per sq. in. gage pressure, and over, for miscellaneous services, 6000-8000 ft. per min.; superheated steam, 200 lb. per sq. in. gage pressure and over, for large turbines and boiler lines, 10,000–15,000 ft. per min.

Although the above formulas are of the turbulent form, they are applicable to either

type of flow, so long as the friction factor is selected from the chart according to the Reynolds number.

For viscosity of steam and water, see Fig. 10 (Viscosity of Water and Superheated Steam, Hawkins, Solberg and Potter, *Trans.* A.S.M.E. FSP–57–11, 1935), to which has been added the viscosity of air according to Bremond, *Comptes Rendus*. To convert viscosity in centipoises to ft.-lb.-sec. units, multiply by 0.000672.

The density of water and steam can be obtained from the steam tables, pp. 5–04 to 5–09, and the density of air from the base value, 0.0807 lb. per cu. ft. at 32 deg. F. and 29.92 in. of mercury, corrected for temperature and pressure by the perfect gas law (see p. 3–74).

RESISTANCE TO FLOW BY BENDS, VALVES, ETC.—Unfortunately no data on the resistance of specials have been assembled on the same basis as given above for straight pipe. One method in vogue is known as the velocity head method, the loss for each type of fitting being given as a number of velocity heads. The method of expressing the loss due to pipe fittings and valves, as the equivalent resistance of so many feet of straight pipe is convenient and time-saving in that one calculation will suffice, the length of straight pipe being adjusted to include the bends, valves, etc. Dean E. Foster (*Trans.* A.S.M.E. xlii, p. 648, 1920) gives formulas for the length of straight pipe equivalent in resistance to screwed fittings, as follows:

For steam, air and gases, $L_e = 43.7\, r_f\, D^{1.2}$ . . . . [5]

For water, $L_e = 53.75\, r_f\, D^{1.25}$ . . . . [6]

where L_e = length, ft., of straight pipe of resistance equivalent to the fitting; D = inside diameter of pipe, ft.; r_f = resistance factor of the fitting, which is dependent upon the type of fitting. Table 10 has been computed from formulas [5] and [6] for standard pipe.

Table 10.—Resistance of Screwed Pipe Fittings *

Nominal Pipe Size, in.	Actual Inside Diameter, in.	Gate Valve	Long Radius Ell or On-run of Standard Tee	Medium Radius Ell or On-run of Tee Reduced 25% in Size	Standard Ell or On-run of Tee Reduced 50% in Size	Angle Valve	Close Return Bend	Tee Through Side Outlet	Globe Valve
		Resistance Factor for Type of Fitting							
		0.25	0.33	0.42	0.67	0.90	1.00	1.33	2.00
Steam, Air, Gas. Equivalent Length of Straight Pipe, ft.									
1/2	0.622	0.31	0.41	0.52	0.84	1.1	1.3	1.7	2.5
3/4	0.824	0.44	0.57	0.73	1.2	1.6	1.8	2.3	3.5
1	1.049	0.57	0.77	0.98	1.6	2.1	2.3	3.1	4.7
1 1/4	1.380	0.82	1.1	1.4	2.2	2.9	3.3	4.4	6.5
1 1/2	1.610	0.98	1.3	1.6	2.6	3.5	3.9	5.2	7.8
2	2.067	1.3	1.7	2.2	3.6	4.8	5.3	7.1	10.6
2 1/2	2.469	1.6	2.2	2.8	4.4	5.9	6.6	8.7	13.1
3	3.068	2.1	2.8	3.6	5.7	7.7	8.5	11.4	17.1
4	4.026	3.0	3.9	5.0	7.9	10.7	11.8	15.8	23.7
5	5.047	3.9	5.1	6.5	10.4	13.9	15.5	20.6	31.0
6	6.065	4.8	6.4	8.1	12.9	17.4	19.3	25.6	38.5
8	7.981	6.7	8.9	11.2	17.9	24.1	26.8	35.6	53.6
10	10.020	8.8	11.5	14.7	23.4	31.5	35.0	46.6	70.0
12	12.090	10.9	14.4	18.4	29.3	39.3	43.7	58.1	87.4
Water and Non-viscous Liquids. Equivalent Length of Straight Pipe, ft.									
1/2	0.622	0.33	0.44	0.56	0.89	1.2	1.4	1.8	2.7
3/4	0.824	0.48	0.63	0.79	1.3	1.7	1.9	2.5	3.8
1	1.049	0.64	0.84	1.1	1.7	2.3	2.6	3.4	5.1
1 1/4	1.380	0.90	1.2	1.5	2.4	3.2	3.6	4.8	7.2
1 1/2	1.610	1.1	1.4	1.8	2.9	3.9	4.4	5.8	8.7
2	2.067	1.5	1.9	2.5	4.0	5.4	6.0	7.9	11.9
2 1/2	2.469	1.9	2.5	3.1	5.0	6.7	7.5	9.9	14.9
3	3.068	2.5	3.3	4.1	6.7	8.9	9.9	13.1	19.7
4	4.026	3.4	4.5	5.8	9.2	12.4	13.7	18.3	27.5
5	5.047	4.6	6.0	7.7	12.2	16.5	18.3	24.3	36.6
6	6.065	5.7	7.6	9.6	15.3	20.6	22.9	30.4	45.8
8	7.981	8.1	10.7	13.6	21.7	29.2	32.4	43.1	64.8
10	10.020	10.7	14.1	18.0	28.7	38.5	42.8	56.9	85.6
12	12.090	12.5	17.8	22.7	36.2	48.6	54.0	71.8	108.0

* For flanged fittings use 0.75 × resistance of screw fittings.

The weight of steam flowing through a pipe can be directly related to the value of Δp by rearranging equation [3] in terms of weight and area. The weight of steam flowing in lb. per sec. $= w = Av\rho$, where A = area of pipe, sq. ft., and v = velocity, ft. per sec. $A = \pi D^2/4 = \pi(d/12)^2/4$, whence $v = w/\{\rho\pi(d/12)^2/4\}$. By substituting this value of v in equation [3], we derive the equation

$$w = \sqrt{\Delta p\rho d^5/43.53 fL} \quad [7]$$

EXAMPLE.—Required the size of pipe to deliver 375,000 lb. of steam per hour, through a main 150 ft. long, having 2 gate valves, 1 angle valve and 3 long-radius ells, with fusion welded flanges. Average steam pressure in the main, 500 lb. per sq. in., abs.; steam temperature 800° F.; pressure drop, Δp, not to exceed 10 lb. per sq. in. *Solution.*—Assume a velocity of 12,500 ft. per min. = 208.34 ft. per sec. From Table 2, p. 5–09, density of steam at 500 lb. per sq. in., abs. pressure, and 800° F. temperature $= \rho = 1/v = 0.69614$.

$$\text{Area of pipe required} = (375{,}000 \times 144)/(60 \times 0.69614 \times 12{,}500) = 103.43 \text{ sq. in.}$$

From Table 13, the allowable fiber stress in electric fusion-welded Grade C steel pipe is 7200 lb. per sq. in., giving a value of schedule number for Table 14 of $1000 \times 500/7200 = 69.4$. Interpolating in Table 14, select a 12-in. O.D. pipe with wall thickness of 0.625 in. Then $d = 11.5$ and $D = 0.958$. From Fig. 10, find $\mu = 0.0345$; converting to the ft.-lb. sec. system,

$$\mu = 0.0345 \times 0.000672 = 0.0000232.$$

Reynolds number $= \rho vD/\mu = 0.69614 \times 208.34 \times 0.958/0.0000232 = 5{,}989{,}000$. Table 16, p. 4–60, indicates that f is determined by curve 4, Fig. 11, whence $f = 0.0119$.

By equation [5] and a value of f from Table 10, the equivalent length of straight pipe for 2 gate valves with flanged joints will be: $2 \times 0.75 \times 43.7 \times 0.25 \times 0.958 = 15.55$ ft. Similarly, the equivalent lengths of straight pipe for 1 angle valve and 3 long-radius ells will be, respectively, 29.50 ft. and 30.81 ft. Total equivalent length of straight pipe then will be $150 + 15.55 + 29.50 + 30.81 = 225.86$ ft. By equation [3] $\Delta p = 0.001295 \times 0.0119 \times (225.86/11.5) \times 0.69614 \times 208.34^2 = 9.15$ lb. per sq. in. Should the calculation of Δp result in too large a value, the calculation should be repeated with a larger diameter of pipe.

4. STEAM PIPING *

The construction of steam and other pressure piping is covered by the Code for Pressure Piping, American Tentative Standard, 1933. The code covers steam, gas, air, oil, and district heating piping systems, and fabrication details of hangers, supports, pipe joints, and expansion of pipes. The following notes are abstracted from the code.

Pipe

Steam piping systems are classified in the code according to service, gage pressures and temperatures as follows: 1. Pressures 250–1500 lb. per sq. in.; temperatures 450–750° F. 2. Pressures 125–250 lb. per sq. in.; maximum temperature, 450° F. 3. Pressures 25–125 lb. per sq. in.; maximum temperature 450° F. 4. Pressures under 25 lb. per sq. in.; maximum temperature, 450° F.

MATERIAL that may be used for pipe for these various services is given in Table 11. Chemical and physical properties of the various materials are summarized in Table 12.

THICKNESS OF PIPING.—For inspection purposes, minimum thickness of pipe wall is

$$t_m = (PD/2S) + C \quad [8]$$

where t_m = minimum pipe wall thickness, in.; P = maximum internal service pressure, lb. per sq. in., gage; D = actual outside diameter of pipe, in.; S = allowable stress in material, lb. per sq. in., plus water hammer allowance in case of cast-iron pipe; C = allowance for threading, mechanical strength and corrosion, in.; Values of S for various temperatures and materials are given in Table 13. Values of C are: Cast-iron pipe, cast in horizontal molds or centrifugally, 0.14; cast-iron pipe, pit-cast, 0.18; threaded steel, $0.8/n$, where n = no. of threads per in.; grooved steel pipe, depth of groove, in.; plain end steel, wrought-iron or non-ferrous pipe or tube, 1 in. size or smaller, 0.05; over 1 in. size, 0.065. Plain end pipe includes pipe joined by flared compression couplings, by lap (Van Stone) joints, and by welding.

Tables 14 and 15 (A.S.A.B–36) show approximately the thickness of steel and wrought-iron pipe to be used for various pressure-stress relations, but exact thickness should be calculated according to the Code rules.

Water hammer allowance for cast-iron pipe to be added to P in formula [8] is

Pipe size, in.	4–10	12–18	20	24–30	36–48	54–84
Water hammer allowance lb. per sq. in.	120	110	100	95	90	85

Valves and Fittings

Fittings shall conform to the requirements of the several classes of service as listed below. Welded fittings may be used. Wrought or forged fittings for welding, and built-up manifolds, shall conform to the same thickness requirements and be constructed like the pipe with which they are to be used.

Valves shall be manufacturers' standard for the respective pressure and temperature, and conform to the requirements for the several classes of service as listed below.

* Staff contribution.

Steam Pressures 250–1500 lb. per sq. in., gage. Temperature, 450–1500° F.—Flanged openings or welding ends are required on all valves and fittings above the following pipe sizes in the given pressure ranges:

Pipe size, in....................	3	2	1 1/2
Pressure range, lb. per sq. in......	250–400	400–600	600–1500

Flanges of flanged valves and fittings shall conform to American Standard B16e (see Table 48).

Fittings shall be cast or forged steel, but non-ferrous fittings may be used for temperatures under 500° F. Forged or cast-steel screw fittings may be used in sizes up to 3 in. for pressures above 250 lb. per sq. in. but not over 400 lb. per sq. in.; 1 1/2 in. above 600 lb. but not over 1500 lb. if their design is suitable for the pressure and temperature. Malleable-iron screwed fittings in accordance with the 300-lb. M.S.S.* Standard Practice SP–31 may be used for maximum pressures of 300 lb. and maximum temperatures of 500° F.

Valves shall be of cast or forged steel, or of forged or cast non-ferrous material if temperature is under 500° F. Malleable-iron valves may be used for service pressures not over 300 lb. per sq. in., and temperatures under 500° F. Stem threads may be internal or external with reference to the valve bonnet, which may be joined to the body either by screwed or flanged connections in the following sizes: 3 in. and smaller for pressures of 250 to 400 lb. per sq. in; 2 in. and smaller for pressures of 400 to 600 lb. per sq. in.; 1 1/2 in. and smaller for pressures of 600 to 1500 lb. per sq. in. Stem threads on valves larger than the above sizes shall be external to the valve body, and used in connection with a yoke and flanged bonnet. Steam valves 8 in. and larger shall have a 3/4-in. (minimum) by-pass of seamless steel of quality equal to A.S.T.M. specification A106 and of minimum weight equal to Schedule 80 of American Standard B36. (Table 14).

Pressures 125–250 lb. per sq. in., gage. Maximum Temperature, 450° F.—Steel flanged fittings shall conform to the 300-lb. American Standard B16e (Tables 44–48); cast-iron fittings shall conform to the 250 lb. American Standard B16b for flanged, and B16d for screwed fittings (Tables 34–38). Malleable-iron screw fittings shall conform to the 300-lb. M.S.S. Standard Practice SP–31 (Table 36), except that the 150-lb. American Standard B16c (Table 38) may be used in accordance with its table of adjusted temperature-pressure ratings. Bodies shall be of cast iron, malleable iron, steel, brass, bronze or Monel metal. Flanges of brass or bronze flanged fittings shall conform to M.S.S. SP–2 for 250 lb. Cast-steel fittings with welding ends shall conform, in material and thick-

* M.S.S. = Manufacturers Standardization Society of the Valve and Fittings Industry.

Table 11.—Type of Pipe for Various Classes of Service

Type of Pipe or Tubing	Class of Service 1[1]						2[1]		3[2]	4[3]
	450–1500 lb.			250–400 lb.			125–250 lb.		25–125 lb.	Under 25 lb.
		H.T.[4] H.P.	C, B[5]		H.T.[4] H.P.	C, B[5]		C, B[5]		
	A.S.T.M. Specification No. (See Table 12)									
Brass, seamless...........							B43[6]		B43[6]	B43[6]
Copper, seamless.........							B42[6]		B42[6]	B42[6]
Copper tubing...........							B75[6]		B75[6]	B75[6]
Steel, seamless, low carbon grade A...............			A155			A155				
Steel, seamless...........			A106		A53	A106				
Steel, seamless, low carbon.			A53			A53		A53		
Steel, seamless, high-temp. high-pressure service....	A106			A106						
Steel, lap-welded, high-temp. high-press. service				A106						
Steel, welded.............							A53		A120[7]	A120[7]
Steel, electric-fusion-welded, 18 in. and over for high temp. high-press. service.		A155			A155					
Steel, electric-fusion-welded, ordinary................							A139			
Steel, electric-fusion-welded, ordinary, large size.....							A134			
Steel, electric-resistance-welded................					A135					
Steel, forge welded..							A136			
Wrought iron, welded.....										

ness requirements, to American Standard B16e. The 150-lb. series may be used in accordance with its table of adjusted pressure-temperature ratings.

Valve bodies, bonnets and yokes shall be of cast iron, malleable iron, steel, brass, bronze or Monel metal. Flanges of brass or bronze valves shall conform to M.S.S. SP–2. (Table 42) Drilling and facing of flanged valves is recommended in accordance with American Standard B16b

Table 12.—Chemical and Physical Specifications for Pipe Material

	Chemical								Physical			
	C	Mn	P, max.	S, max.	Cu	Pb, max.	Sn	Fe max.	Tensile Strength, 1000 lb. per sq. in.	Yield Point, 1000 lb. per sq. in.	Elongation in 8 in., percent	Elongation in 2 in., percent
B42 Copper pipe, less than 2 in. O.D.					99.90				40		10[1]	
B42 Copper pipe, over 2 in. O.D.					99.90				30		25[1]	
B43 Muntz metal[2]					59–63	0.50	0.15[3]	0.07				
B43 High brass[2]					65–68	0.80	0.15[3]	0.07				
B43 Admiralty metal[2]					70–73	0.07	0.9–1.2	0.07				
B43 Red brass[2]					84–87	0.07	0.15[3]	0.07				
A53. Welded and Seamless Steel Pipe												
Lap-welded Bessemer			0.06						50	30	18	
" " open-hearth			0.06						45	25	20	30
Seamless Bessemer			0.045						48	26		40
" open-hearth			0.045						62	35		25
A72. Welded Wrought-iron Pipe									40	24	12	
B75. Copper Tubing, Seamless					99.90				4			
A106. Welded and Seamless Pipe												
Welded, open-hearth		0.30–0.60	0.045	0.06					45	25	22	
Seamless, Grade A		0.30–0.60	0.04	0.06					48	30		35
" Grade B	0.40[5]	0.35–1.50	0.04	0.06					60	35		25
" Grade C	[5]	0.35–1.50	0.04	0.06					75	45		20
A120. Black and Galvanized Welded and Seamless Steel Pipe for ordinary service	No chemical or physical requirements specified. Hydrostatic test required.											
A134												
Grade A Plate 3/4 in. or less	0.15[3]	0.35–0.60	0.06	0.05					45	0.5 T.S.[6]	28	
Grade A Plate over 3/4 in.	0.17[3]	0.35–0.60	0.06	0.05					45	0.5 T.S.[6]	28	
Grade B Plate 3/4 in. or less	0.20[3]	0.35–0.60	0.06	0.05					50	0.5 T.S.[7]	25	
Grade B Plate over 3/4 in.	0.22[3]	0.35–0.60	0.06	0.05					50	0.5 T.S.[7]	25	
A135. Electric Resistance-welded Steel Pipe												
Grade A			0.045	0.06					48	30		30
Grade B			0.045	0.06					60	35		25
A136. Forge Welded Steel Pipe												
Grade A		0.35–0.60	0.06	0.05					45	0.5 T.S.[6]	28	
Grade B		0.35–0.60	0.06	0.05					50	0.5 T.S.[7]	25	
A139. Electric Fusion-welded Pipe												
Grade A									48	30		30
Grade B									60	35		25
A155. Electric Fusion-welded Pipe, 18 in. and larger												
Plate 3/4 in. or less												
Grade A	0.15	0.35–0.60	0.05[8] 0.04[9]	0.05					45	0.5 T.S.[6]		1,500,000 / T.S.
Grade B	0.20	0.35–0.60		0.05					50	0.5 T.S.[7]		
Grade C	0.25	0.30–0.60		0.05					55	0.05 T.S.		
Plate over 3/4 in.												
Grade A	0.17	0.35–0.60	0.05[8] 0.04[9]	0.05					45	0.05 T.S.[6]		1,500,000 / T.S.
Grade B	0.22	0.35–0.60		0.05					50	0.05 T.S.[7]		
Grade C	0.30	0.30–0.60		0.05					55	0.05 T.S.		

[1] Elongation in 4 in. [2] Zn = remainder. [3] Maximum. [4] Physical requirements not specified. Hydrostatic test required, sufficient to subject material to fiber stress of 6000 lb. per sq. in. as calculated by formula [8], C being taken as 0.

[5] If grade B or C pipe are to be joined by fusion welding, purchaser may wish to specify carbon content by special agreement. See A.S.T.M. specification A151 of steels suitable for fusion welding.

[6] 2400 lb. per sq. in. minimum. [7] 27,000 lb. per sq. in. minimum. [8] Acid. [9] Basic.

(Table 33). Gate, angle and globe valves smaller than 3 in. may have inside screw. Stop valves, 8 in. or larger, are to be by-passed. If pipe is used in by-pass, it is to be steel or wrought iron.

Steam Pressures 25–125 lb. per sq. in., gage. Maximum Temperature 450° F.—Fittings shall be of the 125-lb. American Standard, either screwed (A.S.A. B16d) or flanged (A.S.A. B16a) (Tables 36 and 38), or 150-lb. American Standard B16c, malleable iron (Table 38) or bronze (M.S.S. SP–10) (Tables 37–41). Flanges of bronze fittings shall be in accordance with M.S.S. SP–2 (Table 42). Cast-steel fittings shall conform, in material and thickness, to American Standard B16e for 150 lb. service pressure.

Valves shall have cast-iron, malleable-iron, steel or brass bodies, bonnets, discs, and yokes.

Table 13.—Allowable Stress Values (S) for Pipe in Pressure Piping Systems *

Material	Specification	Values of S, lb. per sq. in. for Temperatures not to exceed†					
		406° F.	450° F.	700° F.	750° F.	800° F.	850° F.
Seamless steel:							
Grade A..........	A.S.T.M. A106	.....	....	9,600	9,000	7,020	5800
Grade B..........	A.S.T.M. A106	.....	....	12,400	11,500	9,160	7520
Grade C..........	A.S.T.M. A106	.....	....	15,000	14,000	11,200	9300
Seamless steel:							
Low carbon.......	A.S.T.M. A53	.....	....	9,600	9,000	.....	.....
Medium carbon...	A.S.T.M. A53	.....	....	12,400	11,500	.....	.....
Seamless steel.......	A.S.T.M. A120	.....	9600	.....	......	.....	.....
Electric-fusion-welded steel (high press.-high temp. service):							
Grade A..........	A.S.T.M. A155	.....	....	8,100	7,400	5,900	4900
Grade B..........	A.S.T.M. A155	.....	....	9,000	8,200	6,600	5450
Grade C..........	A.S.T.M. A155	.....	....	9,900	9,000	7,200	6070
Electric-fusion-welded steel:							
Grade A ‡........	A.S.T.M. A139	.....	....	7,700	7,200	.....	.....
Grade B ‡........	A.S.T.M. A139	.....	....	9,600	9,000	.....	.....
Electric-fusion-welded steel:							
Large size........	A.S.T.M. A134	.....	....	0.16TS	0.15TS §	.....	.....
Electric-resistance-welded steel:							
Grade A..........	A.S.T.M. A135	.....	....	8,200	7,600	.....	.....
Grade B..........	A.S.T.M. A135	.....	....	10,200	9,500	.....	.....
Lap-welded steel....	A.S.T.M. A106	.....	....	7,600	7,000	.....	.....
Lap-welded steel....	A.S.T.M. A53	.....	....	7,600	7,000	.....	.....
Lap-welded steel....	A.S.T.M. A120	.....	7600	.....	......	.....	.....
Forge-welded steel:							
Grade A..........	A.S.T.M. A136	.....	....	7,200	6,700	.....	.....
Grade B..........	A.S.T.M. A136	.....	....	8,000	7,500	.....	.....
Butt-welded steel....	A.S.T.M. A53	.....	....	6,000	5,600	.....	.....
Butt-welded steel....	A.S.T.M. A120	.....	6000	.....	......	.....	.....
Steel or wrought iron, riveted..........	A.S.T.M. A138	.....	$(TS \times E)/5$¶	.....	......	.....	.....
Lap-welded wrought iron.............	A.S.T.M. A72	.....	....	5,700	5,300	.....	.....
Butt-welded wrought iron.............	A.S.T.M. A72	.....	....	4,900	4,500	.....	.....
Seamless brass pipe..	A.S.T.M. B43	4500	....	.....	......	.....	.....
Seamless copper pipe	A.S.T.M. B42	4000	....	.....	......	.....	.....
Seamless copper tubing.............	A.S.T.M. B75	4000	....	.....	......	.....	.....
Seamless copper tubing.............	A.S.T.M. B88	4000	....	.....	......	.....	.....
Cast iron, centrifugally-cast.........	F.S.B.WW-P-421	.....	6000	.....	......	.....	.....
Cast iron, pit-cast...	A.W.W.A.	.....	4000	.....	......	.....	.....

* To the minimum pipe wall thickness calculated from any of the above S values, the manufacturing tolerance, demanded for the pipe considered, must be added to obtain the nominal wall thickness. (See American Standard for Wrought Iron and Wrought Steel Pipe, B36).

† The several types and grades of pipe tabulated above shall not be used at temperatures in cess of the maximum temperatures for which the S values are indicated.

‡ If plate material having physical properties other than stated in Section 6 of A.S.T.M. Specification A139 is used in the manufacture of ordinary electric-fusion-welded steel pipe, the allowable stress shall be taken as 0.15 × tensile strength for temperatures above 700° but not over 750° F., and 0.16 × tensile strength for temperatures of 700° F and below.

§ TS = Ultimate tensile strength of the material.

¶ E = Efficiency of the joint.

Drilling and facing of flanges in accordance with American Standard B16a (Table 33) is recommended. Flanges of brass or bronze valves shall conform to M.S.S. SP–2 (Table 42) for 150 lb.

Steam Pressures Up to 25 lb. per sq. in. gage. Maximum Temperature 450° F.—Flanged fittings shall conform to 25-lb. American Standard B16b2 (Table 34). Screwed fittings shall conform to 125-lb. American Standard B16d (Table 38), or the 150-lb. American Standard B16d (Table 38) for cast iron, or the 150-lb. American Standard B16c (Table 38) for malleable iron, or M.S.S. Standard Practice SP–10 (Tables 37–41) for bronze. Bodies may be of cast iron, malleable iron, brass or bronze. Flanges of brass or bronze fittings shall be in accordance with M.S.S. SP–2 dimensions for

Table 14.—Dimensions of Lap Welded and Seamless Steel Pipe for High Temperature Service

Nominal Pipe Size, in.	Outside Diameter, in.	Schedule Number *									
		10	20	30	40	60	80	100	120	140	160
		Nominal Wall Thickness, in.									
1/8	0.405	....	....	....	0.068	....	0.095	....	....	....	....
1/4	.540	....	....	....	.088	....	.119	....	....	....	....
3/8	.675	....	....	....	.091	....	.126	....	....	....	....
1/2	.840	....	....	....	.109	....	.147	....	....	....	0.187
3/4	1.050	....	....	....	.113	....	.154	....	....	....	.218
1	1.315	....	....	....	.133	....	.179	....	....	....	.250
1 1/4	1.660	....	....	....	.140	....	.191	....	....	....	.250
1 1/2	1.900	....	....	....	.145	....	.200	....	....	....	.281
2	2.375	....	....	....	.154	....	.218	....	....	....	.343
2 1/2	2.875	....	....	....	.203	....	.276	....	....	....	.375
3	3.500	....	....	....	.216	....	.300	....	....	....	.437
3 1/2	4.000	....	....	....	.226	....	.318	....	....	....	....
4	4.500	....	....	....	.237	....	.337	....	0.437	....	.531
5	5.563	....	....	....	.258	....	.375	....	.500	....	.625
6	6.625	....	....	....	.280	....	.432	....	.562	....	.718
8	8.625	....	0.250	0.277	.322	0.406	.500	0.593	.718	0.812	.906
10	10.75	....	.250	.307	.365	.500	.593	.718	.843	1.000	1.125
12	12.75	....	.250	.330	.406	.562	.687	.843	1.000	1.125	1.312
14 O.D.	14.00	0.250	.312	.375	.437	.593	.750	.937	1.062	1.250	1.406
16 O.D.	16.00	.250	.312	.375	.500	.656	.843	1.031	1.218	1.437	1.562
18 O.D.	18.00	.250	.312	.437	.562	.718	.937	1.156	1.343	1.562	1.750
20 O.D.	20.00	.250	.375	.500	.593	.812	1.031	1.250	1.500	1.750	1.937
24 O.D.	24.00	.250	.375	.562	.687	.937	1.218	1.500	1.750	2.062	2.312
30 O.D.	30.00	.312	.500	.625	....	....	....	....	....	....	....

* Schedule No. = approxim
able fiber stress, lb. per sq. in.

Table 15.—Thickness of Welded Wrought-iron Pipe

Nominal Pipe Size, in.	Outside Diameter, in.	Schedule Number *					
			20	30	40	60	
		Nominal Wall Thickness, in.					
	0.405				0.070		0.098
	540				.090		.122
	675				.093		.129
	840				.111		.151
3/4	050				.115		.157
1	315				.136		.183
1 1/4	660				.143		.195
1 1/2	900				.148		.204
2	375				.158		.223
2 1/2	875				.208		.282
3	5				.221		.306
3 1/2	0				.231		.325
4	5				.242		.344
5	563				.263		.383
	625				.286		.441
8	625			0.283	.329		.510
10	75			.313	.372	0.510	.606
12	75			.336	.414	0.574	.702
14 O. D.	0	0.250	0.312	.375	.437	0.625	.750
16 O. D	0	.250	.312	.375	.500	0.687	
18 O. D.	0	.250	.312	.437	.562	0.750	
20 O. D.	0	......	.375	.500	.562		......

* Schedule Number indicates approximately values of $\{1000 \times (P/S)\}$. P = internal pressure, lb. per sq. in.; S = allowable fiber stress, lb. per sq. in.

150 lb. (Table 42). Cast-steel fittings with welding ends may be used if their material and wall thickness are suitable for this class of service.

Valves shall have cast-iron, malleable-iron or brass bodies, bonnets, discs, and yokes. Flanges shall conform to 25-lb. American Standard B16b2 (Table 33) for pipe sizes 4 in. and over, and to 125-lb. American Standard B16a (Table 33) for sizes under 4 in. Bronze or brass flanges shall be in accordance with M.S.S. SP–2 dimensions (Table 42) for 150-lb.

Pressure-Temperature Ratings of Fittings.—The A.S.M.E. Boiler Code requires that all valves and fittings shall at least equal the requirements of the American Standard for 125-lb. per sq. in. except where a higher pressure or steel construction is specifically required. The scheduled working pressure for steel fittings may be adjusted to actual maximum allowable working pressure by means of Table 16.

Table 16.—Pressure-Temperature Ratings for Steel Flanged Fittings and Companion Flanges

A.S.M.E. Boiler Code, 1934 Revision

Service Tempera-tures, deg. F.	Pressures Given in American Standards for 750 deg. F.						Service Tempera-tures, deg. F.	Pressures Given in American Standards for 750 deg. F.					
	100	300	400	600	900	1500		100	300	400	600	900	1500
	Service Pressures, lb. per sq. in., gage, at Temperatures from 100 to 850 deg. F.							Service Pressures, lb. per sq. in., gage, at Temperatures from 100 to 850 deg. F.					
100	230	500	670	1000	1500	2500	500	150	375	500	750	1125	1875
150	220	480	640	960	1440	2400	550	140	360	480	720	1060	1800
200	210	465	620	930	1395	2325	600	130	345	460	690	1035	1725
250	200	450	600	900	1350	2250	650	120	330	440	660	990	1650
300	190	435	580	870	1305	2175	700	110	315	420	630	945	1575
350	180	420	560	840	1260	2100	750	100	300	400	600	900	1500
400	170	405	540	810	1215	2025	800	85	250	325	500	750	1250
450	160	390	520	780	1170	1950	850	70	200	270	400	600	1000

Pipe Joints

Pipe joints may be threaded, flanged or welded. Special joints are permissible under certain conditions.

Threaded joints shall conform to American Standard B2 for Taper Pipe Threads. Flanged joints may have: Flanges cast or forged integral with the pipe, fittings or valves; screwed companion flanges, permitted in sizes and for maximum service ratings covered by American Standards (see Tables 33, 42, 44, 45); steel flanges grooved for rolling in the pipe with an expanding tool, permitted in sizes and maximum service ratings for screwed companion flanges, the grooves being designed to carry the entire load of pressure and piping strains; lapped (Van Stone) flanges, permitted in sizes and maximum service ratings covered by integral flanges.

Lapped (Van Stone) Joints.—All cast-iron flanges for lapped joints shall have a high hub and the joint shall have an outside diameter of lap substantially equal to but not greater than the diameter formed by the innermost face edge of the bolt holes. All flanges for lapped joints shall have their faces machined true to provide an even bearing against the backs of the lapped pipe ends. Pipe for use with the 300-lb. American Standard B16e (Table 45) companion flanges may be lapped without upsetting to obtain a minimum thickness through the lap equal to the thickness of the pipe. Where a female tongue-and-groove ring joint, or ground joint facing, is machined in the lap, minimum thickness through lap shall not be less than thickness of pipe recommended for that service. Total clearance between bore of a lapped flange and the outside diameter of pipe, except for service pressures below 25 lb. per sq. in., shall not exceed the following:

Pipe size, in.	6 and less	8–12	14 and over
Maximum total clearance, in.	$^1/_8$	$^3/_{16}$	$^1/_4$

Flange Facing.—Plain flanges shall be faced smooth. Flanges with $^1/_{16}$-in. or $^1/_4$-in. raised face, may be used (Table 47). The gasket surface may be faced smooth or finished with concentric or special grooves, $^1/_{32}$ in. deep, 16 and 32 grooves per in. for cast iron, and steel, respectively, being recommended. Flanges with large male and female, or wide tongue-and-groove facing, may be used as covered by American Standards. The gasket surface may be smooth or have concentric or spiral grooves. The small male and female, or narrow tongue-and-groove, may be used in steel only as covered by American Standard B16e (Table 47). The gasket surface shall be finished smooth. Metallic gaskets are recommended. Ring-type joints having flange faces grooved for a soft steel ring gasket, ground-type joints with flange faces machined with a concave or conical seat to make a joint with a loose ring, or with a convex face on one of the flanges, and seal-welded-type joints are permitted with steel flanges only.

Gaskets of paper or vegetable fiber shall not be used for temperatures over 250° F., and only where this type of material is required to resist the action of the fluid. Rubber inserted gaskets may be used with plain-face flanges for temperatures not over 250° F. Asbestos composition gaskets may be used with any flange facing, except small male and female or narrow tongue-and-groove. Jacketed asbestos or metallic gaskets, either plain or corrugated, are not limited as to pressure or temperature.

WELDED JOINTS made by the fusion process may be butt or fillet welds. See Fig. 12. The Code covers joints formed by pipe end to end, pipe branches, pipe to flanges, fittings and valves, and pipe, valve or fitting to other equipment.

Materials for pipe, fittings, valves and flanges must conform to the requirements of the various sections of the Code. They must be of good weldable quality, free from laminations, harmful ingredients or defects. Filler metal, electrodes, welding wire, and welding rods shall be suitable for use with the base metals to be welded.

Butt Welds are those whose throat, *i.e.* the minimum thickness of the weld, not including reinforcement, along a straight line passing through the root, lies in a plane at approximately 90 deg. from the surface of at least one of the parts joined. In double welded joints, filler metal is added to both sides; in single welded joints it is added to one side only. Parts shall be prepared for fusion welding approximately as in Fig. 12. Welds may be single- or double-V type, or U-bevel. Welding procedure shall insure complete penetration of deposited metal to the bottom of the joint, and thorough fusion of deposited and base metal. Ferrules or backing strips inside the pipe may be used if properly secured and thoroughly fused to the weld. Minimum throat shall be the thickness of the thinner part joined. Welds shall be reinforced in excess of net throat not less than $1/16$ in. for material up to and including $1/2$ in. thick, and $1/8$ in. for thicker material.

Fillet Welds are of approximately triangular cross-section, with the throat in a plane at approximately 45 deg. from the surface of the parts joined. See Fig. 12. They may be either single or double fillet. Welding procedure shall insure complete penetration and thorough fusion of deposited and base metal. Minimum throat shall be 0.75 × nominal size of weld, *i.e.*, the width of the shortest of its adjacent fused legs.

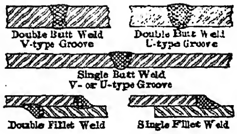


FIG. 12. Fusion Welds

Seal Welds are continuous arc or gas welds primarily intended to secure tightness. They may be either butt or fillet type, and should be made of as small cross-section as practicable. They shall not be considered as contributing to the strength of the joint. Seal welding must be so done as to avoid undue straining of the joint by temperature changes.

Cast Iron and Non-ferrous Materials shall be welded, when permitted, with bronze or other suitable filler metal. Copper pipe shall not be welded with copper welding rod, but may be welded with bronze rod; brass pipe also may be welded with bronze rod.

Flanges.—Fig. 13 shows designs of welds for steel flange connections. The slip-on flange is limited to service pressures not over 300 lb. per sq. in.

Welding Procedure.—Beveling preferably shall be by machine. Torch beveling may be used if the surfaces afterward are cleaned thoroughly from scale and oxidation. All surfaces must be free of paint, oil, rust and scale, except that a light coat of linseed oil to prevent rust is permissible. No part may be offset from an adjacent part by more than 20% of the pipe thickness. Length of tack welds shall be approximately twice the thickness of the thinner material joined. Tack welds must be kept below the outside surface, and melted out during welding. No globules of metal may project within the pipe so as to seriously restrict its area or cause danger of loosening and falling into the pipe. The thickness of reinforcement of butt welds shall increase gradually from edge to center, and no depressions below the surface of the pipe are permissible. Nozzles, tees and branches in materials of $3/4$ in. or more wall thickness shall be welded under shop conditions. Precautions must be taken to insure that welding operations do not cause distortion by heat that would prevent any valve, sliding fixture or other operating equipment from functioning properly.

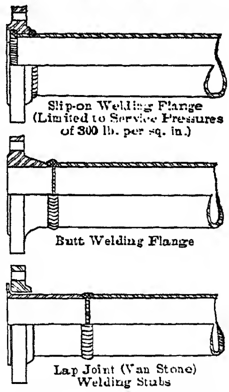


FIG. 13. Welds for Steel Flange Connections

Stress Relief shall be effected by uniformly heating joints to between 1100° and 1200° F., where the pipe wall is $3/4$ in. thick or more, and service temperature exceeds 250° F. The heated parts shall be brought slowly to the required temperature, held at this temperature for 1 hour per inch of wall thickness, and allowed to cool slowly in the atmosphere. The complete structure may be heated as a unit, or a complete section containing the weld may be heated before attachment to other sections of the work, or a circumferential band of minimum width of 6 × plate thickness on each side of the weld may be so heated that the entire band is brought to the required temperature and held there for the specified time.

The Code contains instructions for the testing of welded joints, and for determining the qualifications of welders.

SPECIFIC REQUIREMENTS FOR PIPE JOINTS for various services are abstracted below: Welded joints and joints other than welded to be in accordance with the Code (see above).

Steam Pressures Over 250 lb.–1500 lb. per sq. in., gage. Temperatures 450°–750° F.—Flanges and bolting to conform to American Standard B16e (Tables 44–46). Unions to be of forged steel and suitable for the pressure.

Steam Pressures 125–250 lb. per sq. in., gage. Maximum Temperature 450° F.—Flanges and bolting to conform to 300-lb. American Standard B16e (Tables 44–46), or 250-lb. American Standard B16b (Table 33) for steel and cast iron, respectively, and to M.S.S. Standard Practice SP-2b (Table 42) for bronze. The 150-lb. American Standard B16e may be used in accordance with its adjusted temperature pressure rating table. Unions shall be suitable for the pressure.

Steam Pressures, 25–125 lb. per sq. in., gage. Maximum Temperature 450° F.—Flanges and bolting to conform to 125-lb. American Standard B16a (Table 33) for cast iron, and to M.S.S. Stand-

ard Practice SP–2a (Table 42) for bronze. Unions to be suitable for temperature and pressure. Cast-iron pipe joints may be welded with bronze if service pressure is not over 125 lb. per sq. in., and temperature is not over 353° F.

Steam Pressure 25 lb. per sq. in., gage, and Less.—Maximum Temperature 450° F.—Companion flanges to conform to 25 lb. American Standard B16b2 (Table 33). Bronze flanges to conform to M.S.S. Standard Practice SP–2a (Table 42). Unions to be suitable for the pressure. Cast-iron pipe joints may be welded with bronze if temperature does not exceed 353° F.

EXPANSION of the more common piping materials shall be calculated on the basis of total expansion in inches per 100 ft. as follows:

Temp. °F.	32	100	150	200	250	300	350	400	450	500	550	600	650	700	750
	Expansion, in.														
Steel, Wrought iron	0	0.5	0.9	1.3	1.7	2.2	2.6	3.0	3.5	4.0	4.5	5.0	5.5	6.0	6.5
Cast iron	0	0.5	0.8	1.2	1.6	1.9	2.3	2.7	3.1	3.5	...	...	...	...	...
Copper	0	0.8	1.4	2.0	2.5	3.1	3.7	4.3	4.9	5.6	...	...	...	...	...
Brass, Bronze	0	0.8	1.4	2.1	2.7	3.4	4.1	4.8	5.5	6.2	...	...	...	...	...

FLEXIBILITY OF PIPING SYSTEMS shall be sufficient to prevent thermal expansion from causing unsafe stresses in piping material, excessive bending moments at joints or excessive thrusts on equipment or at anchorage points. Flexibility shall be provided by changes of direction, or by bends, loops, offsets or expansion joints of slip or corrugated types. Under certain conditions flexibility may be increased by corrugating or creasing the pipe wall. Where pipe lines join, provision must be made for expansion of both branches. Pipe may be cut short by an amount approximately one-half the calculated expansion and sprung into place, but no reduction in expansion to be cared for shall be made on account of such cold springing. However, if a line lies between anchors in one plane, the effective expansion between anchors may be taken as $2/3$ of the computed expansion. In calculating flexibility, allowance must be made for the flattening effect in curved pipes.

PIPING SUPPORTS shall be designed to allow free expansion and contraction of pipe without causing excessive strains in pipe, anchors or supports. Supports shall permit side movement of pipe caused by normal expansion. Anchors and guides shall be located to confine and guide expansion in a direction permitting proper use of the flexibility of the system.

5. STRESSES IN PIPE LINES *

In the design of a piping structure the limiting features usually are the thrust that can be imposed on the members to which the piping is to be connected and the stress allowable in the pipe at the temperature at which it is to be used.

The first step, therefore, is to determine the thrust and stress on the proposed layout. Once these are determined it usually is possible to modify the design to meet the imposed conditions. If the design is too stiff and highly stressed it will be necessary to add length to some of the members, to add short lengths at right angles to each other on the plan of the U-bend or to use some kind of expansion joint if the space is limited.

Stresses encountered come under several headings:

1. Stress due to thermal expansion which can be determined by the graphical method.
2. Bursting or hoop stress which is found from the formula

$$S = pd/2t \qquad [9]$$

where p = pressure, lb. per sq. inch; d = inside diam. of pipe, in.; t = wall thickness of pipe, in.

3. Longitudinal stress due to internal pressure, which is equal to $1/2 \times$ (hoop stress).
4. Longitudinal bending stress due to the dead weight of unsupported lengths of piping. Stresses can be determined from beam formulas, and hangers properly spaced to reduce the stresses.
5. Stresses due to direct thrust or direct tension. These are equal to the direct forces divided by the net area of the pipe and usually are small.
6. Torsion. If the piping between anchor points is in one plane no torsion will be present. However, if the piping lies in more than one plane the points where torsion will occur can be located by inspection and the torsional stress calculated by consideration of the forces applied.

In calculating the stresses due to thermal expansion the modulus of elasticity used should be selected for the temperature at which the pipe is to be used. The value of the modulus drops rapidly with rise in temperature. The proper value can be derived by the formula of G. A. Orrok (High Pressure Steam Boilers, *Trans.* A.S.M.E., FSP–50–28, 1928),

$$E_t = E_{32}\left[1 - \{(t - 32)/1700\}^2\right] \qquad [10]$$

where E_{32} = modulus of elasticity of the material at 32° F.; t = temperature at which the modulus is required; E_t = modulus of elasticity at temperature t.

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* Contributed by A. S. McCormick.

If the pipe is to operate at a temperature below the creep range, that is below 750° F., the stress may be reduced as much as 50% by cold setting or springing. If the working temperature is above 750° F. it will be found best to set up the pipe without any cold strains, provided that the stress during the first heating-up cycle does not exceed the elastic limit of the material for the temperature attained. Creep action will reduce the strains to a minimum for the temperature cycle and the remaining stress will appear in the opposite direction when the pipe is cooled. The use of pipe made of material which has good creep strength should be considered in high-temperature installations.

A GRAPHIC METHOD FOR DETERMINING STRESSES IN PIPING is given by C. T. Mitchell in *Trans.* A.S.M.E., FSP-52-25, 1930. Being graphic, it is independent of tables and complicated formulas. It considers the pipe structure as a whole, and is applicable to any independent pipe structure in one or more planes.

Two assumptions are made in every problem: 1. The ends or anchor points are assumed to be 100% fixed. 2. The moment of inertia of the pipe is assumed to be constant, whether the section be taken at a bend or elsewhere. No attempt is made to determine the horizontal or vertical reactions at anchor points. Also, all bends are eliminated from the structure during the analysis, and square corners substituted in their place. A correction subsequently is made for the influence of bends.

Notation.—A, B, C, F, G, H, J = moment areas, sq. ft.; a, b, c, f, g, h, j, l, m, n, o = distances on sketches, ft.; D = outside diam. of pipe, in.; d = inside diam. of pipe in.; Δ = expansion of pipe between anchors, in.; E = modulus of elasticity; I = moment of inertia, in.^{4}; $M = \Sigma$(moment areas $\times$ distance of center of gravity of each to neutral axis), *i.e.*, $M = \Sigma(Aa + Bb + Cc + \ldots)$; P = thrust between anchors, lb.; P_c = thrust corrected for bends in pipe, lb.; R = shortest distance from neutral axis to point of greatest bending moment, ft.; S = stress in outside fiber of pipe, lb. per sq. in.; T = total length of pipe with square corner bends, ft.; t = total length of pipe with full radius bends, ft.

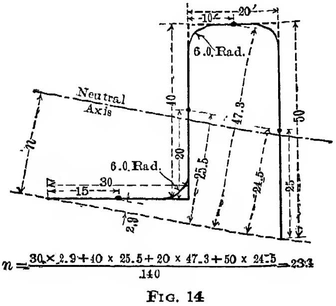
The application of the method will be illustrated by a problem wherein the pipe is in a single plane, and one wherein it is in two planes.

Typical Single-plane Problem.—Arrangement of piping is shown in Fig. 14. Conditions: 12-in. pipe, wall thickness, 1.219 in.; D = 12.75 in.; I = 740.0; steam temperature, 750° F.; room temperature, 70° F.; expansion of steel pipe through 680° F., 7.20 in. per 100 ft.

The procedure is as follows: 1. Redraw the arrangement to a convenient scale, with square corners instead of bends (dotted lines, Fig. 14). 2. Locate neutral axis as shown in Fig. 14. 3. Draw moment areas A, B, C, F, G, H as in Fig. 15. 4. Compute the square feet in these areas, as below. The sum of the areas on each side of the neutral axis should be equal.

$$\begin{aligned}
A &= 1/2\{(23.4 + 17.6) \times 30\} &&= 615\\
B &= 1/2\,(17.6 \times 17.9) &&= 158\\
C &= 1/2\,(21.9 \times 22.1) &&= 242\\
F &= 1/2\,\{(21.9 + 25.6) \times 20\} &&= 475\\
G &= 1/2\,(25.6 \times 26.2) &&= 335\\
H &= 1/2\,(23.4 \times 23.8) &&= 279
\end{aligned}$$

5. Compute M. $M = Aa + Bb + Cc + Ff + Gg + Hh$ = (615 × 20.8) + (158 × 11.7) + (242 × 14.4) + (475 × 24.0) + (335 × 16.8) + (279 × 15.7) = 39,545. 6. Compute P. Dis-



$$n = \frac{30 \times 2.9 + 40 \times 25.5 + 20 \times 47.3 + 50 \times 24.5}{140} = 23.4$$

FIG. 14

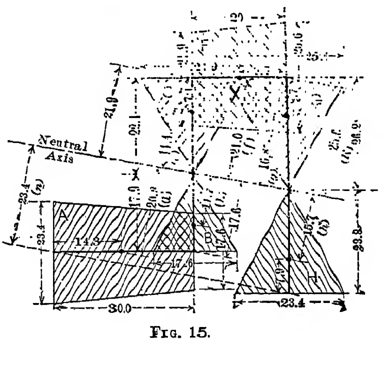


FIG. 15.

tance between anchors = 51.0 ft. Consequently, $\Delta = (7.20 \times 51)/100 = 3.68$ in., and $P = \Delta EI/1728\ M = (3.68 \times 29{,}000{,}000 \times 740.0)/(1728 \times 39{,}545) = 1160$ lb. $P_c = PT/t = 1160 \times 140/132.2 = 1230$ lb. 7. Compute S. The greatest stress in the pipe line is located at a point at the greatest distance from the neutral axis. By scaling Fig. 15, this distance is found to be $R = 25.6$ ft. Then by substitution in the formula

$$S = 6\, P_c RD/I \qquad [11]$$

we find $S = (6 \times 1230 \times 25.6 \times 12.750)/740 = 3250$ lb. per sq. in. The stress may be calculated at any point along the pipe by scaling the perpendicular distance from this point to the neutral axis, and substituting the value so found for the value of R in formula [11].

Typical Two-plane Problem.—Arrangement of the piping is shown in perspective in Fig. 16. Conditions: 6-in. pipe, wall thickness, 0.5 in.; $D = 6.625$ in.; $I = 45.4$; steam temperature, 750° F.; room temperature, 70° F.; expansion of steel pipe through 680° F., 7.20 in. per 100 ft.

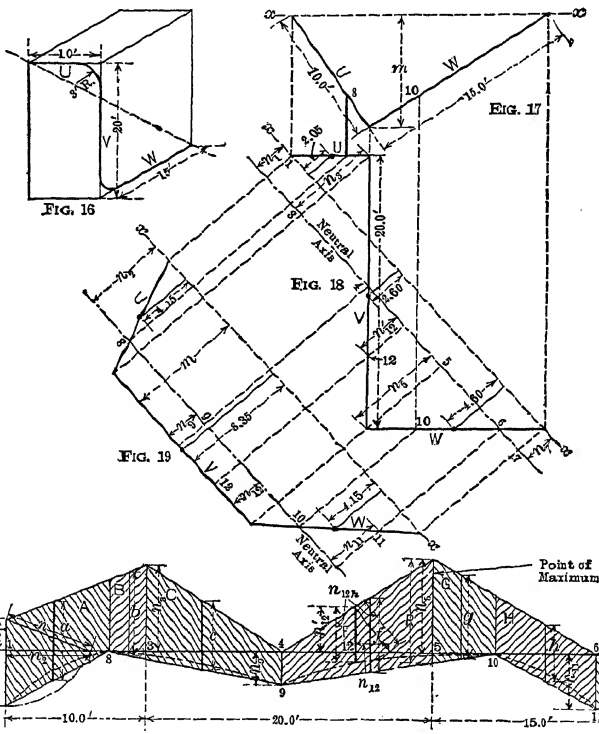


FIG. 16 FIG. 17 FIG. 18 FIG. 19

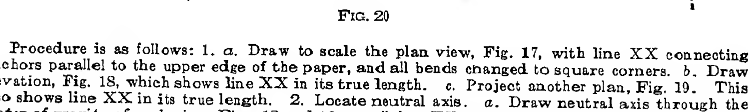


FIG. 20

Procedure is as follows: 1. *a*. Draw to scale the plan view, Fig. 17, with line XX connecting anchors parallel to the upper edge of the paper, and all bends changed to square corners. *b*. Draw elevation, Fig. 18, which shows line XX in its true length. *c*. Project another plan, Fig. 19. This also shows line XX in its true length. 2. Locate neutral axis. *a*. Draw neutral axis through the center of gravity of each view, Figs. 18 and 19, parallel to XX, taking care to use the true length of each pipe member in the calculations. Let U, V, W be the true length of the several pipe members, and u, v, w, respectively, the perpendicular distance from the mid-point of each member to the neutral axis. Let n = distance of neutral axis from the center of gravity of the system. Then

$$n_1 = (Uu_1 + Vv_1 + Ww_1)/(U + V + W).$$

and

$$n_2 = (Uu_2 + Vv_2 + Ww_2)/(U + VW)$$

If the mid-points of the several sections lie on the same side of the line XX, the products Uu, Vv,

Ww will be of like algebraic sign. If they lie on opposite sides of XX, the products will be of unlike sign.

$$n_1 = \{(10 \times 2.05) - (20 \times 2.60) - (15 \times 4.60)\}/45 = -\ 2.23 \text{ ft. (Fig. 18)}$$

$$n_2 = \{(10 \times 4.15) + (20 \times 8.35) + (15 \times 4.15)\}/45 = +\ 6.02 \text{ ft. (Fig. 19)}$$

3. Draw moment areas A, B, C, F, G, H, J, as shown in Fig. 20. *a.* To determine moment areas, it is necessary to know the perpendicular distance from the neutral axis to all parts of the piping arrangement. The distance for any point is the hypotenuse of a right triangle, the legs of which are the ordinates to the neutral axis from the point in question, which may be scaled from the drawings, Figs. 18 and 19. *b.* Draw the moment areas about the line YY, Fig. 20. This line represents the total length of pipe, with each member drawn to scale. Above YY, and perpendicular to it, draw the ordinates for the end of each member, as scaled from Fig. 18. These are drawn *above* YY, irrespective of their position in Fig. 18. Connect the ends of the ordinates by straight lines. The irregular dotted line below YY is so drawn that at any point the vertical distance between it and the line above YY is the bending moment at that point. This is done by laying out on YY a right triangle, whose base along YY is the ordinate to the neutral axis as scaled from Fig. 18, and whose altitude is the ordinate to the neutral axis as scaled from Fig. 19; the hypotenuse is the bending moment. The length of the hypotenuse, measured on a line perpendicular to YY through the given point, from the upper line of the moment diagram gives a point on the lower line of the diagram. See point n_{12} on Fig. 20, and the corresponding points on Figs. 18 and 19. Where the pipe intersects the neutral axis in Fig. 18, the ordinate at that point is, of course, zero, and the position of the point in the lower line of the moment diagram is determined by the length of the ordinate in Fig. 19. See points 9 and 11 in Figs. 18, 19 and 20. It is unnecessary to plot a large number of bending moments. The dotted line, forming the true boundary of the moment diagram, is a refinement that is not worth while. Using the trapezoidal areas A, B, C, etc., instead of the true moment area will give results that are sufficiently accurate. In locating points along YY in Fig. 20, the true distance along the pipe member under consideration should be taken, as determined by projecting the point to Fig. 17 or 18 as may be necessary. See point 8, which is projected from Fig. 19 to Fig. 17, to find its correct position in Fig. 20.

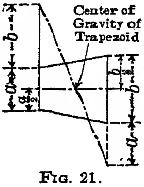


Fig. 21.

4. The next step is to divide the total moment area into a convenient number of trapezoids as A, B, C, F, G, H, and J, and to compute the square feet in each area. Scaling the bases and altitudes of the several trapezoids in Fig. 20, we find the areas to be

$$A = 1/2\,(6.4 + 5.2) \times 7.2 = 41.7$$
$$B = 1/2\,(5.2 + 6.9) \times 2.8 = 17.0$$
$$C = 1/2\,(6.9 + 2.3) \times 9.5 = 43.7$$
$$F = 1/2\,(2.3 + 7.7) \times 10.5 = 52.5$$
$$G = 1/2\,(7.7 + 4.5) \times 4.2 = 25.6$$
$$H = 1/2\,(4.5 + 4.0) \times 7.1 = 30.2$$
$$J = 1/2\,(4.0 + 6.4) \times 3.7 = 19.3$$

5. Next compute M. The lengths of lines a, b, c, f, g, h, j are determined by locating the center of gravity of each trapezoid, as shown in Fig. 21, drawing a perpendicular to YY through each center of gravity, and scaling the intercept between the sides of the trapezoid. Then

$$M = Aa + Bb + Cc + Ff + Gg + Hh + Jj = (41.7 \times 5.8) + (17.0 \times 6.1) + (43.7 \times 5.0)$$
$$+ (52.5 \times 5.4) + (25.6 \times 6.2) + (30.2 \times 4.1) + (19.3 \times 5.2) = 1237.$$

6. Compute P. Distance between anchors = 26.9 ft., and $\Delta = (7.20 \times 26.9)/100 = 1.94$ in. Then $P = \Delta EI/1728\,M = (1.94 \times 29{,}000{,}000 \times 45.4)/(1728 \times 1237) = 1195$ lb.

$$P_c = PT/t = (1195 \times 45)/42.6 = 1260 \text{ lb.}$$

7. Compute S. The maximum strain occurs at the point of greatest bending moment. This point is shown in Fig. 20, and the distance R is found to be 7.7 ft. Substituting in formula [11].

$$S = (6 \times 1260 \times 7.7 \times 6.625)/45.4 = 8500 \text{ lb. per sq. in.}$$

The stress may be calculated at any point along the pipe line by scaling the ordinate at this point in Fig. 20, and substituting the value so found for R in formula [11].

Correction for Square Corners can be made by multiplying the values obtained for Δ in the above analysis by the factor

(Total length of actual structure) ÷ (Total length of square-corner structure).

Results will be 90 to 100% accurate, which is all that reasonably can be expected in problems of this character, in view of the uncertainties always present. If allowance is made for flattened cross-section of the pipe at bends, it should be remembered that the factor of safety of the structure therby is lowered.

In two-plane problems the stress may be due to a combination of torsion and bending. Mr. Mitchell explains in the paper why the stress may be treated as a simple bending stress, and also explains the development of the correction factor for square corners. The theory underlying the graphic method is demonstrated.

Table 17.—Dimensions of Standard Welded Steel Pipe
(National Tube Company, Pittsburgh)

Size	Diameter, in. External	Diameter, in. Internal	Thickness, in.	Threads per Inch	Weight per Foot, lb. Plain Ends	Weight per Foot, lb. Threads and Coupling	Circumference, in. External	Circumference, in. Internal	Transverse Area, sq. in. External	Transverse Area, sq. in. Internal	Transverse Area, sq. in. Metal	Length of Pipe, ft. per sq. ft. External Surface	Length of Pipe, ft. per sq. ft. Internal Surface	Length, ft., to Contain 1 cu. ft.	U. S. Gallons in 1 Lineal Foot of Pipe	Water in 1 Lineal Foot of Pipe, lb.
1/8	0.405	0.269	0.068	27	0.244	0.245	1.272	0.845	0.129	0.057	0.072	9.431	14.199	2533.775	0.003	0.025
1/4	.540	.364	.088	18	.424	.425	1.696	1.144	.229	.104	.125	7.073	10.493	1383.789	.005	.045
3/8	.675	.493	.091	18	.567	.568	2.121	1.549	.358	.191	.167	5.658	7.747	754.360	.010	.083
1/2	.840	.622	.109	14	.850	.852	2.639	1.954	.554	.304	.250	4.547	6.141	473.906	.016	.132
3/4	1.050	.824	.113	14	1.130	1.134	3.299	2.589	.866	.533	.333	3.637	4.635	270.034	.028	.231
1	1.315	1.049	.133	11 1/2	1.678	1.684	4.131	3.296	1.358	.864	.494	2.904	3.641	166.618	.045	.375
1 1/4	1.660	1.380	.140	11 1/2	2.272	2.281	5.215	4.335	2.164	1.495	.669	2.301	2.767	96.275	.078	.648
1 1/2	1.900	1.610	.145	11 1/2	2.717	2.731	5.969	5.058	2.835	2.036	.799	2.010	2.372	70.733	.106	.883
2	2.375	2.067	.154	11 1/2	3.652	3.678	7.461	6.494	4.430	3.355	1.075	1.608	1.847	42.913	.174	1.455
2 1/2	2.875	2.469	.203	8	5.793	5.819	9.032	7.757	6.492	4.788	1.704	1.328	1.547	30.077	.249	2.076
3	3.500	3.068	.216	8	7.575	7.616	10.996	9.638	9.621	7.393	2.228	1.091	1.245	19.479	.384	3.205
3 1/2	4.000	3.548	.226	8	9.109	9.202	12.566	11.146	12.566	9.886	2.680	0.954	1.076	14.565	.514	4.286
4	4.500	4.026	.237	8	10.790	10.889	14.137	12.648	15.904	12.730	3.174	.848	0.948	11.312	.661	5.519
4 1/2	5.000	4.506	.247	8	12.538	12.642	15.708	14.156	19.635	15.947	3.688	.763	.847	9.030	.828	6.913
5	5.563	5.047	.258	8	14.617	14.810	17.477	15.856	24.306	20.006	4.300	.686	.756	7.198	1.039	8.673
6	6.625	6.065	.280	8	18.974	19.185	20.813	19.054	34.472	28.891	5.581	.576	.629	4.984	1.501	12.524
7	7.625	7.023	.301	8	23.544	23.769	23.955	22.063	45.664	38.738	6.926	.500	.543	3.717	2.012	16.793
8	8.625	8.071	.277	8	24.696	25.000	27.096	25.356	58.426	51.161	7.265	.442	.473	2.815	2.658	22.179
8	8.625	7.981	.322	8	28.554	28.809	27.096	25.073	58.426	50.027	8.399	.442	.478	2.878	2.599	21.687
9	9.625	8.941	.342	8	33.907	34.188	30.238	28.089	72.760	62.786	9.974	.396	.427	2.294	3.262	27.218
10	10.750	10.192	.279	8	31.201	32.000	33.772	32.019	90.763	81.585	9.178	.355	.374	1.765	4.238	35.368
10	10.750	10.136	.307	8	34.240	35.000	33.772	31.843	90.763	80.691	10.072	.355	.376	1.785	4.192	34.980
10	10.750	10.020	.365	8	40.483	41.132	33.772	31.479	90.763	78.855	11.908	.355	.381	1.826	4.096	34.184
11	11.750	11.000	.375	8	45.557	46.247	36.914	34.558	108.434	95.033	13.401	.325	.347	1.515	4.937	41.198
12	12.750	12.090	.330	8	43.773	45.000	40.055	37.982	127.676	114.800	12.876	.299	.315	1.254	5.964	49.767
12	12.750	12.000	.375	8	49.562	50.706	40.055	37.699	127.676	113.097	14.579	.299	.318	1.273	5.875	49.028
14 O. D.	14.000	13.250	.375	8	54.568	55.824	43.982	41.626	153.938	137.886	16.052	.272	.288	1.044	7.163	59.775
15 O. D.	15.000	14.250	.375	8	58.573	60.375	47.124	44.768	176.715	159.485	17.230	.254	.268	0.903	8.285	69.138
16 O. D.	16.000	15.250	.375	8	62.579	64.500	50.265	47.909	201.062	182.654	18.408	.238	.250	.788	9.489	79.182
17 O. D.	17.000	16.214	.393	8	69.704	72.602	53.407	50.938	226.980	206.476	20.504	.224	.235	.697	10.726	89.509
18 O. D.	18.000	17.182	.409	8	76.840	80.482	56.549	53.979	254.469	231.866	22.603	.212	.222	.621	12.045	100.516
20 O. D.	20.000	19.182	.409	8	85.577	89.617	62.832	60.262	314.159	288.986	25.173	.190	.199	.498	15.012	125.278

6. COMMERCIAL PIPE AND TUBING

Tables 17 to 32 give data on steel, wrought-iron and non-ferrous pipe commercially available for steam service, and also on seamless tubes and boiler tubes.

STEEL WELDED PIPE is furnished with threads and couplings and in random lengths unless otherwise ordered. Permissible variation in weight is ±5% of the weights given in Tables 17–19. Weights are based on 1 cu. in. of steel = 0.2833 lb., and on 20-ft.

Table 18.—Extra Strong Welded Steel Pipe

(National Tube Co., Pittsburgh)

Size	Diameter, in.				Circumference, in.		Transverse Area, sq. in.			Length, ft., of Pipe per sq. ft. External urface	iternal Surfac	
1/8	0.405	0.215	0 095	0.314	.272	0.675	0.129	0.036	0.093	9.431	17.766	3966.393
1/4	.540	.302	.119	535	.696	.949	.229	072	.157	7.073	12.648	2010.290
3/8	.675	.423	.126	738	.121	1.329	.358	141	.217	5.658	9.030	1024.689
1/2	.840	.546	.147	087	.639	1.715	.554	234	.320	4 547	6.995	615.017
3/4	1.050	.742	.154	473	.299	2.331	.866	433	.433	3.637	5.147	333.016
1	1.315	.957	.179	171	.131	3.007	1.358	719	.639	2.904	3.991	200.193
1 1/4	1.660	1.278	.191	996	.215	4.015	2.164	283	.881	2.301	2.988	112.256
1 1/2	1.900	1.500	.200	631	.969	4.712	2.835	767	1.068	2.010	2.546	81.487
2	2.375	1.939	.218	022	.461	6.092	4.430	953	1.477	1.608	1.969	48.766
2 1/2	2.875	2.323	.276	661	.032	7.298	6.492	238	2.254	1.328	1.644	33.976
3	500	2.900	.300	10.252	.996	9.111	9.621	605	3.016	1.091	1.317	21.801
3 1/2	000	3.364	.318	12.505	12.566	10.568	12.566	888	3.	.954	1.135	16.202
4	500	3.826	.337	14.983	14.137	12.020	15.904	497	4.407	.848	0.998	12.525
4 1/2	000	4.290	.355	17.611	15.708	13.477	19.635	455	5.180	.763	.890	9.962
5	5.563	4.813	.375	20.778	17.477	15.120	24.306	194	6.112	.686	.793	7.915
6	6.625	5.761	.432	28.573	20.813	18.099	34.472	067	8.405	.576	.663	5.524
7	7.625	6.62	.500	38.048	23.955	20.813	45.664	472	11.192	.500	.576	4.177
8	8.625	7.625	.500	43.388	27.096	23.955	58.426	663	12.763	.442	.500	3.154
9	9.625	8.625	.500	48.728	30.238	27.096	72.760	426	14.334	.396	.442	2.465
10	10.750	9.750	.500	54.735	33.772	30.631	90.763	662	16.101	.355	.391	1.929
11	11.750	10.750	.500	60.075	36.914	33.772	108.434	763	17.671	.325	.355	1.587
12	12.750	11.750	.500	65.415	40.055	36.914	127.676	.434	19.242	.299	.325	1.328

Table 19.—Double Extra Strong Welded Steel Pipe

(National Tube Co., Pittsburgh)

Size	Diameter, in. External	Diameter, in. Internal	Thickness, in.	Weight per ft., lb. Plain Ends	Circumference, in. External	Circumference, in. Internal	Transverse Area, sq. in. External	Transverse Area, sq. in. Internal	Transverse Area, sq. in. Metal	Length, ft., of Pipe per sq. ft. External Surface	Length, ft., of Pipe per sq. ft. Internal Surface	Length, ft., of Pipe containing 1 cu. ft.
1/2	0.840	0.252	0.294	1.714	2.639	0.792	0.554	0.050	0.504	4.547	15.157	2887.165
3/4	1.050	.434	.308	2.440	3.299	1.363	.866	.148	.718	3.637	8.801	973.404
1	1.315	.599	.358	3.659	4.131	1.882	1.358	.282	1.076	2.904	6.376	510.998
1 1/4	1.660	.896	.382	5.214	5.215	2.815	2.164	.630	1.534	2.301	4.263	228.379
1 1/2	1.900	1.100	.400	6.408	5.969	3.456	2.835	.950	1.885	2.010	3.472	151.526
2	2.375	1.503	.436	9.029	7.461	4.722	4.430	1.774	2.656	1.608	2.541	81.162
2 1/2	2.875	1.771	.552	13.695	9.032	5.564	6.492	2.464	4.028	1.328	2.156	58.457
3	3.500	2.300	.600	18.583	10.996	7.226	9.621	4.155	5.466	1.091	1.660	34.659
3 1/2	4.000	2.728	.636	22.850	12.566	8.570	12.566	5.845	6.721	0.954	1.400	24.637
4	4.500	3.152	.674	27.541	14.137	9.902	15.904	7.803	8.101	.848	1.211	18.454
4 1/2	5.000	3.580	.710	32.530	15.708	11.247	19.635	10.066	9.569	.763	1.066	14.306
5	5.563	4.063	.750	38.552	17.477	12.764	24.306	12.966	11.340	.686	0.940	11.107
6	6.625	4.897	.864	53.160	20.813	15.384	34.472	18.835	15.637	.576	.780	7.646
7	7.625	5.875	.875	63.079	23.955	18.457	45.664	27.109	18.555	.500	.650	5.312
8	8.625	6.875	.875	72.424	27.096	21.598	58.426	37.122	21.304	.442	.555	3.879

lengths of pipe. Weight of water in 1 ft. length of pipe is based on 1 cu. ft. = 62.425 lb.

Average analysis and physical properties of steel used for lap-welded pipe is

	C	Mn	S	P	Elastic Limit	Tensile Strength	Elong. in 8 in.
Bessemer........	0.07	0.35	0.050	0.100	36,000	58,000	22%
Open-hearth.....	0.09	0.40	0.035	0.020	33,000	52,000	25%

Table 20.—Dimensions and Weights of Standard Weight Wrought-iron Pipe
(A. M. Byers Co., Pittsburgh)

Size, in.	Diameter, in.		Weight per ft., lb. Plain Ends	Circumference, in.		Transverse Area, sq. in.		Length, ft., of Pipe per sq. ft.		Length to Contain 1 cu. ft. (Feet)	Content of 1 Lineal Foot, U. S. Gallons
	External	Internal		External	Internal	External	Internal	External Surface	Internal Surface		
1/8	0.405	0.266	0.24	1.27	0.84	0.13	0.06	9.43	14.35	2588.76	0.003
1/4	.540	.360	.42	1.70	1.13	.23	.10	7.07	10.61	1414.23	.005
3/8	.675	.489	.57	2.12	1.54	.36	.19	5.66	7.81	766.99	.010
1/2	.840	.617	.85	2.64	1.94	.55	.30	4.55	6.19	481.32	.016
3/4	1.050	.819	1.13	3.30	2.57	.87	.53	3.64	4.66	273.30	.027
1	1.315	1.043	1.68	4.13	3.28	1.36	.85	2.90	3.66	168.47	.044
1 1/4	1.660	1.374	2.27	5.22	4.32	2.16	1.48	2.30	2.78	97.13	.077
1 1/2	1.900	1.604	2.72	5.97	5.04	2.84	2.02	2.01	2.38	71.28	.105
2	2.375	2.060	3.65	7.46	6.47	4.43	3.33	1.61	1.85	43.18	.173
2 1/2	2.875	2.450	5.79	9.03	7.76	6.49	4.75	1.33	1.55	30.29	.247
3	3.500	3.059	7.58	11.00	9.61	9.62	7.35	1.09	1.25	19.60	.382
3 1/2	4.000	3.538	9.11	12.57	11.12	12.57	9.83	0.95	1.08	14.64	.511
4	4.500	4.016	10.79	14.14	12.62	15.90	12.67	.85	0.95	11.37	.658
5	5.563	5.036	14.62	17.48	15.82	24.31	19.92	.69	.76	7.23	1.035
6	6.625	6.053	18.97	20.81	19.02	34.47	28.78	.58	.63	5.00	1.495
8	8.625	8.059	24.70	27.10	25.32	58.43	51.02	.44	.47	2.82	2.650
8	8.625	7.967	28.55	27.10	25.03	58.43	49.86	.44	.48	2.89	2.590
10	10.750	10.181	31.20	33.77	31.98	90.76	81.40	.36	.38	1.77	4.229
10	10.750	10.124	34.24	33.77	31.80	90.76	80.49	.36	.38	1.79	4.181
10	10.750	10.005	40.48	33.77	31.43	90.76	78.62	.36	.38	1.83	4.084
12	12.750	12.077	43.77	40.06	37.94	127.68	114.54	.30	.32	1.26	5.950
12	12.750	11.985	49.56	40.06	37.65	127.68	112.81	.30	.32	1.28	5.860

Table 21.—Dimensions and Weights of Extra Heavy Wrought-iron Pipe
(A. M. Byers Co., Pittsburgh)

Size, in.	Diameter, in.		Weight per ft., lb. Plain Ends	Circumference, in.		Transverse Area, sq. in.		Length, ft., of Pipe per sq. ft.		Length to Contain 1 cu. ft. (Feet)	Content of 1 Lineal Foot, U. S. Gallons.
	External	Internal		External	Internal	External	Internal	External Surface	Internal Surface		
1/8	0.405	0.207	0.31	1.27	0.65	0.13	0.03	9.43	18.19	4158.85	0.002
1/4	.540	.295	.54	1.70	.93	.23	.07	7.07	12.93	2101.52	.004
3/8	.675	.417	.74	2.12	1.31	.36	.14	5.66	9.16	1055.36	.007
1/2	.840	.539	1.09	2.64	1.69	.55	.23	4.55	7.09	631.37	.012
3/4	1.050	.735	1.47	3.30	2.31	.87	.42	3.64	5.20	339.62	.022
1	1.315	.949	2.17	4.13	2.97	1.36	.71	2.90	4.04	204.25	.037
1 1/4	1.660	1.269	3.00	5.22	3.99	2.16	1.27	2.30	3.01	113.80	.066
1 1/2	1.900	1.491	3.63	5.97	4.68	2.84	1.75	2.01	2.56	82.48	.091
2	2.375	1.929	5.02	7.46	6.06	4.43	2.92	1.61	1.98	49.26	.152
2 1/2	2.875	2.311	7.66	9.03	7.26	6.49	4.19	1.33	1.65	34.34	.218
3	3.500	2.887	10.25	11.00	9.07	9.62	6.55	1.09	1.32	22.00	.340
3 1/2	4.000	3.350	12.51	12.57	10.53	12.57	8.82	0.95	1.14	16.34	.458
4	4.500	3.811	14.98	14.14	11.97	15.90	11.41	.85	1.00	12.62	.593
5	5.563	4.797	20.78	17.48	15.07	24.31	18.07	.69	.80	7.97	.939
6	6.625	5.743	28.57	20.81	18.04	34.47	25.90	.58	.67	5.56	1.345
8	8.625	7.604	43.39	27.10	23.89	58.43	45.41	.44	.50	3.17	2.359
10	10.750	9.729	54.74	33.77	30.57	90.76	74.34	.36	.39	1.94	3.862
12	12.750	11.729	65.42	40.06	36.85	127.68	108.05	.30	.33	1.33	5.613

WROUGHT-IRON WELDED PIPE has a thicker wall and smaller internal diameter than welded steel pipe. It is furnished with threads and couplings, and in random lengths. Permissible variation in weight is +5% and −2 1/2% from the weights given in Tables 20 and 21.

SEAMLESS STEEL TUBES are both hot and cold drawn. Composition of the various grades of steel used are given in Table 22. Physical properties of cold-drawn tubing are given in Table 23. Weights are given in Tables 24 and 25. Table 26 gives the dimensions of standard seamless and lap-welded steel boiler tubes, and Table 27 of seamless steel locomotive boiler tubes. Weights are based on 1 cu. in. of steel = 0.2833 lb. These tables were compiled from data supplied by the National Tube Co., Pittsburgh.

Table 22.—Grades of Steel Used in Seamless Steel Tubing

Grade or Chemical Designation	Percentage of						
	C	Mn	P, max.	S, max.	Ni	Cr	Mo
Boiler Tube	0.08–0.18	0.30–0.60	0.040	0.045	......	....	....
0.10– .20C	.10– .20	.30– .60	.040	.050	......	....	....
.20– .30C	.20– .30	.40– .65	.040	.050	......	....	....
.30– .40C	.30– .40	.40– .65	.040	.050	......	....	....
3.5Ni 0.25– .35C	.25– .35	.50– .80	.040	.045	3.25 3.75	....	....
Cr–Mo	.25– .35	.40– .60	.040	.045	......	0.80–1.10	0.15–0.25

Table 23.—Physical Properties of Cold-drawn Seamless Tubing

Grade (See Table 22)	Anneal, deg. C.	Yield Point, lb. per sq. in.	Ultimate Strength, lb. per sq. in.	Elongation, Percent		Reduction in Area, Percent
				In 2 in.	In 8 in.	
Boiler Tube	700	27,000	47,000	50	27	50
0.10–0.20C	Unannealed	58,000	60,000	16	6	24
.10– .20C	500	46,000	55,000	27	12	30
.10– .20C	600	29,000	48,000	42	20	40
.10– .20C	700	27,000	47,000	50	27	45
.20– .30C	Unannealed	65,000	70,000	10	3	15
.20– .30C	500	53,000	67,000	20	8	22
.20– .30C	600	36,000	55,000	35	15	32
.30– .40C	Unannealed	70,000	82,000	5	..	8
.30– .40C	500	62,000	80,000	15	6	16
.30– .40C	600	46,000	65,000	28	12	27
3.5Ni 0.25– .35C	Unannealed	80,000	90,000	5	..	12
	500	65,000	85,000	14	6	22
	600	52,000	75,000	25	10	30
Cr–Mo	Normalized	60,000	95,000	10	..	...

Table 24.—Weight per Foot of Hot Drawn Seamless Steel Tubing

Outside Diam., in.	Thickness, in.											
	1/4 0.250	5/16 0.313	3/8 0.375	1/2 0.500	5/8 0.625	3/4 0.750	7/8 0.875	1 1.00	1 1/8 1.125	1 1/4 1.250	1 3/8 1.375	1 1/2 1.500
	Pounds per Foot											
6 1/2	16.69	20.65	24.53	32.04	39.22	46.06	52.57	58.74	64.58	70.09	75.26	80.10
7	18.02	22.32	26.53	34.71	42.55	50.06	57.24	64.08	70.59	76.76	82.60	88.11
7 1/2	19.36	23.99	28.54	37.38	45.89	54.07	61.91	69.42	76.60	83.44	89.95	96.12
8	20.69	25.66	30.54	40.05	49.23	58.07	66.58	74.76	82.60	90.11	97.29	104.13
8 1/2	22.03	27.33	32.54	42.72	52.57	62.08	71.26	80.10	88.61	96.79	104.63	112.14
9	23.36	28.99	34.54	45.39	55.90	66.08	75.93	85.44	94.62	103.46	111.97	120.15
9 1/2	24.70	30.66	36.55	48.06	59.24	70.09	80.60	90.78	100.63	110.14	119.32	128.16
10	26.03	32.33	38.55	50.73	62.58	74.09	85.27	96.12	106.63	116.81	126.66	136.17
10 1/2	27.37	34.00	40.55	53.40	65.92	78.10	89.95	101.46	112.64	123.49	134.00	144.18
11	28.70	35.67	42.55	56.07	69.25	82.10	94.62	106.80	118.65	130.16	141.35	152.19
11 1/2	30.04	37.34	44.56	58.74	72.59	86.11	99.29	112.14	124.66	136.84	148.69	160.20
12	31.37	39.01	46.56	61.41	75.93	90.11	103.96	117.48	130.67	143.51	156.03	168.21
13	34.04	42.35	50.56	66.75	82.60	98.12	113.31	128.16	142.68	156.86	170.72	184.23
14	36.71	45.68	54.57	72.09	89.28	106.13	122.65	138.84	154.70	170.22	185.40	200.25
15	39.38	49.02	58.57	77.43	95.95	114.14	132.00	149.52	166.71	183.57	200.09	216.27
16	42.05	52.36	62.58	82.77	102.63	122.15	141.35	160.20	178.73	196.92	214.77	232.29
17	44.72	55.70	66.58	88.11	109.30	130.16	150.69	170.88	190.74	210.27	229.46	248.31
18	47.39	59.03	70.59	93.45	115.98	138.17	160.04	181.56	202.76	223.62	244.14	264.33
19	50.06	62.37	74.59	98.79	122.65	146.18	169.38	192.24	214.77	236.97	258.83	280.35
20	52.73	65.71	78.60	104.13	129.33	154.19	178.73	202.92	226.79	250.32	273.51	296.37

Table 25.—Weight per Foot of Seamless Cold Drawn Mechanical Tubing

(Condensed from table issued by National Tube Co., Pittsburgh)

Outside Diam., In.	Thickness—B.W.G. and Decimals of an Inch								Thickness—Fractions and Decimals of an Inch														
	20	18	16	14	13	12	11	10	5/32	3/16	7/32	1/4	9/32	5/16	11/32	3/8	7/16	1/2	9/16	5/8	3/4	7/8	1
	0.035	0.049	0.065	0.083	0.095	0.109	0.120	0.134	0.156	0.188	0.219	0.250	0.281	0.313	0.344	0.375	0.438	0.500	0.563	0.625	0.750	0.875	1.00
1/2	0.17	0.24	0.30	0.37	0.41	0.46	0.49	0.52	....	....	....	....	....	....	....	....	....	....	....	....	....	....	....
5/8	.22	.30	.39	.48	.54	.60	.65	.70	....	....	....	....	....	....	....	....	....	....	....	....	....	....	....
3/4	.27	.37	.48	.59	.66	.75	.81	.88	0.99	1.13	....	....	....	....	....	....	....	....	....	....	....	....	....
7/8	.31	.43	.56	.70	.79	.89	.97	1.06	1.20	1.38	1.53	....	....	....	....	....	....	....	....	....	....	....	....
1	.36	.50	.65	.81	.92	1.04	1.13	1.24	1.41	1.63	1.83	2.00	....	....	....	....	....	....	....	....	....	....	....
1 1/8	.41	.56	.74	.92	1.05	1.18	1.29	1.42	1.62	1.88	2.12	2.34	2.53	2.71	....	....	....	....	....	....	....	....	....
1 1/4	.45	.63	.82	1.03	1.17	1.33	1.45	1.60	1.83	2.13	2.41	2.67	2.91	3.13	3.33	3.50	....	....	....	....	....	....	....
1 3/8	.50	.69	.91	1.15	1.30	1.47	1.61	1.78	2.03	2.38	2.70	3.00	3.29	3.55	3.79	4.01	4.38	....	....	....	....	....	....
1 1/2	.55	.76	1.00	1.26	1.43	1.62	1.77	1.95	2.24	2.63	2.99	3.34	3.66	3.96	4.24	4.51	4.96	5.34	....	....	....	....	....
1 3/4	....	.89	1.17	1.48	1.68	1.91	2.09	2.31	2.66	3.13	3.58	4.01	4.41	4.80	5.16	5.51	6.13	6.68	7.13	....	....	....	....
2	....	1.02	1.34	1.70	1.93	2.20	2.41	2.67	3.08	3.63	4.16	4.67	5.16	5.63	6.08	6.51	7.30	8.01	8.64	9.18	....	....	....
2 1/4	....	1.15	1.52	1.92	2.19	2.49	2.73	3.03	3.49	4.13	4.75	5.34	5.91	6.47	7.00	7.51	8.47	9.35	10.14	10.85	....	....	....
2 1/2	....	1.28	1.69	2.14	2.44	2.78	3.05	3.39	3.91	4.63	5.33	6.01	6.66	7.30	7.92	8.51	9.64	10.68	11.64	12.52	14.02	....	....
2 3/4	....	....	1.86	2.36	2.69	3.07	3.37	3.74	4.33	5.13	5.91	6.68	7.42	8.14	8.83	9.51	10.81	12.02	13.14	14.18	16.02	....	....
3	....	....	....	2.59	2.95	3.37	3.69	4.10	4.75	5.63	6.50	7.34	8.17	8.97	9.75	10.51	11.97	13.35	14.64	15.85	18.02	19.86	21.36
3 1/4	....	....	....	2.81	3.20	3.66	4.01	4.46	5.16	6.13	7.08	8.01	8.92	9.80	10.67	11.51	13.14	14.69	16.15	17.52	20.03	22.19	24.03
3 1/2	....	....	....	3.03	3.45	3.95	4.33	4.82	5.58	6.63	7.67	8.68	9.67	10.64	11.59	12.52	14.31	16.02	17.65	19.19	22.03	24.53	26.70
3 3/4	....	....	....	3.25	3.71	4.24	4.65	5.18	6.00	7.13	8.25	9.35	10.42	11.47	12.51	13.52	15.48	17.36	19.15	20.86	24.03	26.87	29.37
4	....	....	....	....	3.96	4.53	4.97	5.53	6.41	7.63	8.83	10.01	11.17	12.31	13.42	14.52	16.65	18.69	20.65	22.53	26.03	29.20	32.04
4 1/4	....	....	....	....	....	4.82	5.29	5.89	6.83	8.14	9.42	10.68	11.92	13.14	14.34	15.52	17.81	20.03	22.15	24.20	28.04	31.54	34.71
4 1/2	....	....	....	....	....	5.11	5.61	6.25	7.25	8.64	10.00	11.35	12.67	13.98	15.26	16.52	18.98	21.36	23.65	25.87	30.04	33.88	37.38
4 3/4	....	....	....	....	....	5.40	5.93	6.61	7.67	9.14	10.59	12.02	13.42	14.81	16.18	17.52	20.15	22.70	25.16	27.53	32.04	36.21	40.05
5	....	....	....	....	....	5.69	6.25	6.96	8.08	9.64	11.17	12.68	14.17	15.64	17.09	18.52	21.32	24.03	26.66	29.20	34.04	38.55	42.72
5 1/4	....	....	....	....	....	....	6.57	7.32	8.50	10.14	11.75	13.35	14.93	16.48	18.01	19.52	22.49	25.37	28.16	30.87	36.05	40.88	45.39
5 1/2	....	....	....	....	....	....	6.90	7.68	8.92	10.64	12.34	14.02	15.68	17.31	18.93	20.53	23.65	26.70	29.66	32.54	38.05	43.22	48.06
5 3/4	....	....	....	....	....	....	....	8.04	9.33	11.14	12.92	14.69	16.43	18.15	19.85	21.53	24.82	28.04	31.16	34.21	40.05	45.56	50.73
6	....	....	....	....	....	....	....	8.40	9.75	11.64	13.51	15.35	17.18	18.98	20.77	22.53	25.99	29.37	32.67	35.88	42.05	47.89	53.40
6 1/4	....	....	....	....	....	....	....	....	10.17	12.14	14.09	16.02	17.93	19.82	21.68	23.53	27.16	30.71	34.17	37.55	44.06	50.23	56.07
6 1/2	....	....	....	....	....	....	....	....	10.59	12.64	14.67	16.69	18.68	20.65	22.60	24.53	28.33	32.04	35.67	39.22	46.06	52.57	58.74
6 3/4	....	....	....	....	....	....	....	....	....	13.14	15.26	17.36	19.43	21.49	23.52	25.53	29.50	33.38	37.17	40.88	48.06	54.90	61.41
7	....	....	....	....	....	....	....	....	....	13.64	15.84	18.02	20.18	22.32	24.44	26.53	30.66	34.71	38.67	42.55	50.06	57.24	64.08
7 1/4	....	....	....	....	....	....	....	....	....	14.14	16.43	18.69	20.93	23.15	25.35	27.53	31.83	36.05	40.18	44.22	52.07	59.58	66.75
7 1/2	....	....	....	....	....	....	....	....	....	....	17.01	19.36	21.68	23.99	26.27	28.54	33.00	37.38	41.68	45.89	54.07	61.91	69.42
7 3/4	....	....	....	....	....	....	....	....	....	....	17.60	20.03	22.43	24.82	27.19	29.54	34.17	38.72	43.18	47.56	56.07	64.25	72.09
8	....	....	....	....	....	....	....	....	....	....	18.18	20.69	23.19	25.66	28.11	30.54	35.34	40.05	44.68	49.23	58.07	66.58	74.76
8 1/2	....	....	....	....	....	....	....	....	....	....	....	22.03	24.69	27.33	29.94	32.54	37.67	42.72	47.69	52.57	62.08	71.26	80.10
9	....	....	....	....	....	....	....	....	....	....	....	23.36	26.19	28.99	31.78	34.54	40.01	45.39	50.69	55.90	66.08	75.93	85.44

Table 26.—Standard Boiler Tubes and Flues—Seamless and Lap-Welded *

(National Tube Co., Pittsburgh)

Diameter, in.				Circumference, in.		Transverse Area, sq. in.			Length, ft., of Tube per sq. ft.			gth f Con u.
3/4	1.560	0 095	1.679	5.498	4.901	2.405	1.911	0.494	2.182	2.448	2.315	75.340
2	1.810	095	1.932	6.283	5.686	3.142	2.573	.569	1.909	2.110	2.010	55.965
2 1/4	2.060	095	2.186	7.069	6.472	3.976	3.333	.643	1.697	.854	.775	43.205
2 1/2	2.282	109	2.783	7.854	7.169	4.909	4.090	.819	1.527	.673	.600	35.208
2 3/4	2.532	109	3.074	8.639	7.955	5.940	5.036	.904	1.388	.508	.448	28.599
3	2.782	109	3.365	9.425	8.740	7.069	6.079	.990	.273	.373	.323	23.690
3 1/4	3.010	120	4.011	10.210	9.456	8.296	7.116	.180	.175	.269	.222	20.237
3 1/2	3.260	120	4.331	10.996	10.242	9.621	8.347	.274	1.091	.171	.131	17.252
3 3/4	3.510	120	4.652	11.781	11.027	11.045	9.677	.368	1.018	.088	.053	14.882
4	3.732	.134	5.532	12.566	11.724	.2.566	10.939	.627	0.954	.023	0.989	13.164
4 1/2	4.232	.134	6.248	14.137	13.295	15.904	14.066	.838	848	0.902	.875	10.237
5	4.704	.148	7.669	15.708	14.778	19.635	17.379	2.256	.763	.812	.787	8.286
6	5.670	.165	10.282	18.850	17.813	28.274	25.249	3.025	.636	.673	.655	5.703
7	6.670	.165	12.044	21.991	20.954	38.485	34.942	3.543	.545	.572	.559	4.121
8	7.670	.165	13.807	25.133	24.096	50.265	46.204	4.061	.477	.498	.487	3.117
9	8.640	.180	16.955	28.274	27.143	63.617	58.629	4.988	.424	.442	.433	2.456
10	9.594	.203	21.240	31.416	30 140	78.540	72.292	6.248	.381	.398	.390	1.992
11	10.560	220	25.329	34.558	33.175	95.033	87.582	7.451	.347	.361	.354	1.644
12	11.542	.229	28.788	37.699	36.260	113.097	104.629	8.468	.318	.330	.324	1.376
13	12.524	.238	32.439	40.841	39.345	132.732	123.190	9.542	.293	.304	.299	1.169
14	13.504	248	36.424	43.982	42.424	153.938	143.224	10.714	.272	.282	.277	1.005
15	14.48	.259	40.775	47.124	45.497	176.715	164.721	11.994	.254	.263	.259	0.874
16	15.460	.270	45.359	50.265	48.569	201.062	187.719	13.343	.238	.247	.242	.767

* Sizes up to 4-in. seamless only; 4 1/2 to 6-in., seamless and lap-welded; over 6 in., lap-welded only.

Table 27.—Seamless Locomotive Boiler Tubes *

(National Tube Co., Pittsburgh)

Diameter, in.		Thickness		Wt. per ft., lb.	Surface			
					Length, ft., per sq. ft.		Sq. ft. per Lineal ft.	
External	Internal	In.	B.W.G.		External	Internal	External	Internal
1 1/2	1.310	0.095	13	1.425	2.546	2.915	0.392	0.342
1 1/2	1.282	.109	12	1.619	2.546	2.979	.392	.335
1 1/2	1.260	.120	11	1.768	2.546	3.031	.392	.329
1 1/2	1.232	.134	10	1.954	2.546	3.100	.392	.322
1 1/2	1.204	.148	9	2.137	2.546	3.172	.392	.315
1 3/4	1.560	.095	13	1.679	2.182	2.448	.458	.408
1 3/4	1.532	.109	12	1.910	2.182	2.493	.458	.401
1 3/4	1.510	.120	11	2.089	2.182	2.529	.458	.395
1 3/4	1.482	.134	10	2.312	2.182	2.577	.458	.387
1 3/4	1.454	.148	9	2.532	2.182	2.627	.458	.380
2	1.810	.095	13	1.932	1.909	2.110	.523	.473
2	1.782	.109	12	2.201	1.909	2.143	.523	.466
2	1.760	.120	11	2.409	1.909	2.170	.523	.460
2	1.732	.134	10	2.670	1.909	2.205	.523	.453
2 1/4	2.032	.109	12	2.492	1.697	1.879	.589	.531
2 1/4	2.010	.120	11	2.729	1.697	1.900	.589	.526
2 1/4	1.982	.134	10	3.028	1.697	1.927	.589	.518
2 1/2	2.282	.109	12	2.783	1.527	1.673	.654	.597
2 1/2	2.260	.120	11	3.050	1.527	1.690	.654	.591
2 1/2	2.232	.134	10	3.386	1.527	1.711	.654	.584
3	2.782	.109	12	3.365	1.273	1.373	.785	.728
3	2.760	.120	11	3.691	1.273	1.383	.785	.722
3	2.732	.134	10	4.101	1.273	1.398	.785	.715
3	2.704	.148	9	4.508	1.273	1.412	.785	.707

* Test pressure for all tubes in this table is 1000 lb. per sq. in.

SEAMLESS BRASS TUBES are made from 1/16 in. to 1/8 in. outside diameter, varying by 1/64 in., in all gages from No. 23 to 36 Stubs gage; from 1/8 in. to 5/16 in., varying by 1/32 in., gages from 18 to 36; from 3/8 to 5/8 in., varying by 1/16 in., gages 16 to 35; from 3/4 to 1 in., varying by 1/8 in., gages 8 to 30; from 1 1/4 in. to 7 3/4 in., varying by 1/4 in., gages 1 to 24; from 8 to 10 in., varying by 1/2 in., gages 3 to 16. To determine the weight

of a tube of a given inside diameter, add to the weight for the outside diameter and gage given in Table 28 the weight given below for the corresponding wall thickness.

Thickness, Stubs Gage..........				8	10	12	14	16	18	20	22	24	26
Pounds..........	1.87	1.31	0.954	0.630	0.416	0.275	0.159	0.098	0.056	0.028	0.018	0.011	0.008

Table 28.—Dimensions of Seamless Brass and Copper Tubes, Standard Pipe Sizes
(American Brass Co., Waterbury, Conn.)

Standard Pipe Size, in.	Outside Diam., in.	Regular			Extra Heavy			Standard Pipe Size, in.	Outside Diam., in.	Regular			Extra Heavy		
		Inside Diam., in.	Wt. per ft., lb.		Inside Diam., in.	Wt. per ft., lb.				Inside Diam., in.	Wt. per ft., lb.		Inside Diam., in.	Wt. per ft., lb.	
			Brass	Copper		67 Brass	Copper				67 Brass	Copper		67 Brass	Copper
1/8	0.405	0.281	0.246	0.259	0.205	0.353	0 .371	3 1/2	4.000	3.500	10.85	11.41	3.358	13.67	14.37
1/4	.540	.375	.437	.460	.294	.593	.624	4	4.500	4.000	12.29	12.94	3.818	16.41	17.25
3/8	.675	.494	.612	.643	.421	.805	.847	4 1/2	5.000	4.500	13.74	14.46	4.250	20.07	21.10
1/2	.840	.625	.911	.957	.542	1.19	1.25	5	5.563	5.062	15.40	16.21	4.813	22.52	23.67
3/4	1.050	.822	1.24	1.30	.736	1.62	1.71	6	6.625	6.125	18.44	19.41	5.751	31.32	32.93
1	1.315	1.062	1.74	1.83	.951	2.39	2.51	7	7.625	7.062	23.92	25.17	6.625	41.23	43.34
1 1/4	1.660	1.368	2.56	2.69	1.272	3.30	3.46	8	8.625	8.000	30.05	31.63	7.625	47.02	49.42
1 1/2	1.900	1.600	3.04	3.20	1.494	3.99	4.19	9	9.625	8.937	36.94	38.83	8.625	52.81	55.56
2	2.375	2.062	4.02	4.23	1.933	5.51	5.79	10	10.750	10.019	43.91	46.22	9.750	59.32	62.40
2 1/2	2.875	2.500	5.83	6.14	2.315	8.41	8.84	11	11.750	11.000	49.37	51.94	......	......	.....
3	3.500	3.062	8.31	8.75	2.892	11.24	11.82	12	12.750	12.000	53.71	56.51	......	......	.....

Table 29.—Weight and Dimensions of Everdur Brass Pipe
(American Brass Co., Waterbury, Conn.)

Standard Pipe Size, in.	Outside Diam., in.	Regular			Extra Heavy		
		Inside Diam., in.	Thickness, in.	Lb. per ft.	Inside Diam., in.	Thickness, in.	Lb. per ft.
1/8	0.405	0.281	0.0620	0.247	......	......	......
1/4	.540	.375	.0825	.438	0.294	0.123	0.595
3/8	.675	.494	.0905	.614	.421	.127	.808
1/2	.840	.625	.1075	.914	.542	.149	1.20
3/4	1.050	.822	.1140	1.24	.736	.157	1.63
1	1.315	1.062	.1265	1.75	.951	.182	2.39
1 1/4	1.660	1.368	.1460	2.57	1.272	.194	3.30
1 1/2	1.900	1.600	.1500	3.05	1.494	.203	4.00
2	2.375	2.062	.1565	4.03	1.933	.221	5.53
2 1/2	2.875	2.500	.1875	5.85	2.315	.280	8.44
3	3.500	3.062	.2190	6.34	2.892	.304	11.28
3 1/2	4.000	3.500	.2500	10.88	3.358	.321	13.71
4	4.500	4.000	.2500	12.34	3.818	.341	16.47

Table 30.—Length of Pipe Nipples in Inches

Nominal Pipe Size	Wrought Iron				Bronze			Nominal Pipe Size	Wrought Iron				Bronze		
	Close	Short	Long *		Close	Long *			Close	Short	Long *		Close	Long *	
			Min.	Max.		Min.	Max.				Min.	Max.		Min.	Max.
1/8	3/4	1 1/2	2	3 1/2	3/4	1 1/2	6	2 1/2	2 1/2	3	3 1/2	5	2 1/2	3	6
1/4	7/8	1 1/2	2	3 1/2	7/8	1 1/2	6	3	2 5/8	3	3 1/2	5	2 5/8	3	6
3/8	1	1 1/2	2	3 1/2	1	1 1/2	6	3 1/2	2 3/4	4	4 1/2	6	2 3/4	4	6
1/2	1 1/8	1 1/2	2	3 1/2	1 1/8	1 1/2	6	4	2 7/8	4	4 1/2	6	2 7/8	4	6
3/4	1 3/8	2	2 1/2	4	1 3/8	2	6	5	3	4 1/2	4 1/2	6	3	4 1/2	6
1	1 1/2	2	2 1/2	4	1 1/2	2	6	6	3 1/8	4 1/2	4 1/2	6	3 1/8	4 1/2	6
1 1/4	1 5/8	2 1/2	3	4 1/2	1 5/8	2 1/2	6	8	3 1/2	5	6	8	.....	.....	.....
1 1/2	1 3/4	2 1/2	3	4 1/2	1 3/4	2 1/2	6	10	3 7/8	5	6	8	.....	.....	.....
2	2	2 1/2	3	4 1/2	2	3	6	12	4 1/2	6	.....	8	.....	.....	.....

* Lengths advance by 1/2 in. up to and including 6 in.; 8 in. and 10 in. sizes advance by 1 in.

STANDARD PIPE THREADS.—(American Standards Assoc., 1919). The dimensions of standard pipe threads are given in Table 31. The formulas for these dimensions are:

Pitch diameters.— $A = D - (0.05\,D + 1.1)P$; $B = A + 0.625\,F$;
Length of thread.— $E = (0.80\,D + 6.8)P$,

where A = pitch diameter at end of pipe; B = pitch diameter at gaging notch: D = out-

side diameter of pipe; E = length of effective thread; F = normal engagement by hand between male and female threads; P = pitch of thread = distance axis will advance in one revolution, expressed in threads per inch. All dimensions are in inches. See Fig. 22.

Taper of Thread, 1 in 16, measured on the diameter.

Manufacturing Tolerance.—The maximum allowable variation in the commercial product is one turn plus or minus from the gaging notch when using working gages. Due to an allowance of 1/2 turn on the working gages, this is a maximum allowance of 1 1/2 turns from the basic dimensions.

Form of the Thread.—The included angle of the thread is 60°, measured in the axial plane. The thread is perpendicular to the axis of the pipe for both straight and taper threads. The crest and root are truncated an amount = 0.033 P, and the depth of thread is 0.80 P.

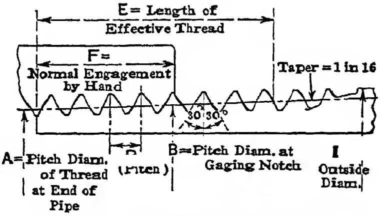


Fig. 22. American Standard Pipe Thread

Length of Thread, as given by the formula, is the effective length, and includes two threads that are imperfect on the crest. In this the formula differs from the original Briggs formula, which determined the number of perfect threads. The Briggs formulas

Table 31.—Dimensions of American Standard Pipe Threads

American Standards Association (A.S.A. B3, 1919)

Nominal Size, in.	Outside		Diam. at End 'ipe, Taper ead, in.		Pitch Diam. at Gaging Notch, Locknut Threads, in.		Length Threa Threa E	Normal Engagemen by Hand between Male and Female Taper Threads, in	
1/8	0.405	27	0.36351	0.37476	0.38402	0.38633	0.2638	0.180	0.02963
	.540	18	.47739	.48989	.50378	.50725	.4018	.200	.04444
/8	.675	18	.61201	.62701	.64090	.64437	.4078	.240	.04444
1/2	.840	14	.75843	.77843	.79628	.80075	.5337	.320	.05714
3/4	1.050	14	.96768	.98886	1.00672	1.01118	.5457	.339	.05714
1	1.315	11 1/2	1.21363	1.23863	1.26037	1.26580	.6828	.400	.06956
1 1/4	1.660	11 1/2	1.55713	1.58338	1.60512	1.61055	.7068	.420	.06956
1 1/2	1.900	11 1/2	1.79609	1.82234	1.84407	1.84951	.7235	.420	.06956
2	2.375	11 1/2	2.26902	2.29627	2.31801	2.32344	.7565	.436	.06956
2 1/2	2.875		2.71953	2.76216	2.79341	2.80122	1.1375	.682	.1000
3	3.500		3.34063	3.38850	3.41975	3.42756	1.2000	.766	.1000
3 1/2	4.00 .		3.83750	3.88881	3.92006	3.92787	1.2500	.821	.1000
4	4.500		4.33438	4.38713	4.41838	4.42619	1.3000	.844	.1000
4 1/2	5.000		4.83125	4.88594	4.91719	4.92500	1.3500	.875	.1000
5	5.563		5.39073	5.44929	5.48054	5.48836	1.4063	.937	.1000
6	6.625		6.44609	6.50597	6.53722	6.54503	1.5125	.958	.1000
7	7.625		7.43984	7.50234	7.53359	7.54141	1.6125	1.000	.1000
8	8.625		8.43359	8.50003	8.53128	8.53909	1.7125	1.063	.1000
9	9.625		9.42734	9.49797	9.52922	9.53703	1.8125	1.130	.1000
10	10.750		10.54531	10.62094	10.65219	10.66000	1.9250	1.210	.1000
11	11.750		11.53906	11.61938	11.65063	11.65844	2.0250	1.285	.1000
12	12.750		12.53281	12.61781	12.64906	12.65688	2.1250	1.360	.1000
14 O.D.	14.000		13.77500	13.87262			2.250	1.562	.1000
15 O.D.	15.000		14.76875	14.87419			2.350	1.687	.1000
16 O.D.	16.000		15.76250	15.87575			2.450	1.812	.1000
17 O.D.	17.00		16.75625	16.87500			2.550	1.900	.1000
18 O.D.	18.00		17.75000	17.87500			2.650	2.000	.1000
20 O.D.	20.00		19.73750	19.87031			2.850	2.125	.1000
22 O.D.	22.00		21.72500	21.86562			3.050	2.250	.1000
24 O.D.	24.00		23.71250	23.86094			3.250	2.375	.1000
26 O.D.	26.00		25.70000	25.85625			3.450	2.500	.1000
28 O.D.	28.00		27.68750	27.85156			3.650	2.625	.1000
30 O.D.	30.00		29.67500	29.84687			3.850	2.750	.1000

which are expressed in different terms than those of the American standard, give identical results with the latter. For a statement of the Briggs formulas, see earlier editions of this book.

Types of Threads.—Taper male and female threads are recommended for threaded joints for any service. Straight threaded female standard-weight couplings may be used with taper threaded pipe for ordinary pressures. For high pressure, only taper male and female threads should be used. Straight male threads are applicable only to special purposes, as long screws and tank nipples. Long screw joints are not satisfactory when subjected to temperature or pressure. In this application the coupling has a straight thread and must make a joint with an American taper pipe thread.

The straight thread of the largest diameter it is possible to cut on a pipe has been standardized under the name of Maximum Male and Minimum Female Locknut Threads. An American standard taper thread is cut on the end of the pipe after having cut the male locknut thread.

Dimensions of the various applications noted above are given in Table 31.

Table 32.—Tap Drills for Pipe Taps

Size of Tap, in.	Size of Drill, in.	Size of Tap, in.	Size of Drill, in.	Size of Tap, in.	Size of Drill, in.	Size of Tap, in.	Size of Drill, in.
1/8	21/64	3/4	15/16	2	2 3/16	4	4 3/16
1/4	29/64	1	1 3/16	2 1/2	2 11/16	4 1/2	4 11/16
3/8	19/32	1 1/4	1 15/32	3	3 5/16	5	5 1/4
1/2	23/32	1 1/2	1 23/32	3 1/2	3 13/16	6	6 5/16

7. PIPE FITTINGS

Pipe Fittings of steel, cast iron, malleable iron and bronze are authorized by the Code for Pressure Piping, with limitations imposed by the service in which they are to be used. Tables 33 to 48 give dimensions of various types and classes of fittings according to the standards of the American Standards Association and the Manufacturers Standardization Society of the Valve and Fitting Industry. The following notes are abstracted from the specifications of the American Standards Association.

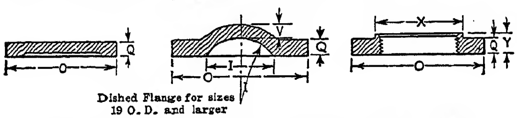


FIG. 23. Cast-iron Companion and Blind Flanges, 125-lb.

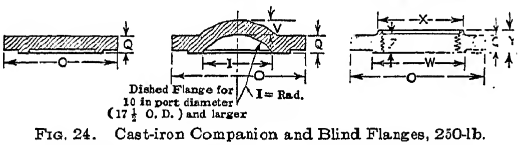


FIG. 24. Cast-iron Companion and Blind Flanges, 250-lb.

In designating the outlets of reducing fittings, the openings should be given in the order indicated by the sequence of the letters *abcd* in Fig. 27. In side outlet fittings, the side outlet is named last. In tees and crosses, run openings are given first and side openings next, the largest dimension being given first in each case.

Cast-iron Fittings.—Physical and chemical requirements for material for flanged fittings for 125 lb. and 250 lb. maximum pressure are: Maximum sulphur, 0.12%; minimum tensile strength, light castings, 20,000 lb. per sq. in.; medium castings, 21,000 lb. per sq. in.; heavy castings, 24,000 lb. per sq. in. For 25 lb. maximum pressure, maximum phosphorous, 0.75%, maximum sulphur, 0.12%; minimum tensile strength, 21,000 lb. per sq. in. Light castings are defined as those in which any section is less than 1/2 in. thick; heavy castings, no section less than 2 in. thick; medium castings, those not included in heavy or light classes. If the fittings are used in non-shock hydraulic work, the following maximum working pressures, lb. per sq. in. may be used: 250-lb. fittings, 10-in. and smaller, 325 lb. at 250° F.; 400 lb. at atmospheric temperatures; 125-lb. fittings, 12 in. and smaller, 175 lb. at atmospheric temperature; 25-lb. fittings, 36 in. and smaller, 43 lb. at atmospheric temperature. All 25-lb. and 125-lb. flanges are plain faced. All 250-lb. flanges have a 1/16-in. raised face; diameters are given in Table 33. Inspection limits of flanged fittings are: 25-lb. fittings, ±1/16 in. on center-to-face dimensions, and ±1/8 in. on face-to-face dimensions; 125-lb. and 250-lb. fittings, ±1/32 in. on center-to-face dimensions of 10-in. fittings and smaller, and ±1/16 in. on sizes over 10 in.; ±1/16 in. on

face-to-face dimensions of 10-in. fittings and smaller, and ±1/8 in. on sizes over 10 in. Inspection limits of center-to-end dimensions of screwed fittings are

Size, in.	1/4	3/8	1/2 to 3/4	1 to 1 1/4	1 1/2 to 2	2 1/2 to 3 1/2	4 to 5	6	8	10	12	14 O. D.	16 O. D.
Limits, ± in.	0.04	0.05	0.06	0.07	0.08	0.10	0.12	0.14	0.16	0.19	0.21	0.23	0.25

Table 33.—Dimensions of Cast-iron Companion and Blind Flanges (Am. Stds. Assoc.)
All dimensions in inches

Nominal Size of Pipe						Dia of X	Len thr Hr Y					Diam. of Bolt Hol	
Maximum Pressure, 25 lb. per sq. in. (A.S.A. B16b2, 1931)													
4	.....	9	3/4	......	......	......	......	......	7 1/2	8	5/8	3/4	2 1/4
5	.....	10	3/4	......	......	......	......	......	8 1/2	8	5/8	3/4	2 1/4
6	.....	11	3/4	......	......	......	......	......	9 1/2	8	5/8	3/4	2 1/4
8	.....	13 1/2	3/4	......	......	......	......	......	11 3/4	8	5/8	3/4	2 1/4
10	.....	16	7/8	......	......	......	......	......	14 1/4	12	5/8	3/4	2 1/2
12	.....	19	1	......	......	......	......	......	17	12	5/8	3/4	2 3/4
14	.....	21	1 1/8	......	......	......	......	......	18 3/4	12	3/4	7/8	3 1/4
16	.....	23 1/2	1 1/8	......	......	......	......	......	21 1/4	16	3/4	7/8	3 1/4
18	.....	25	1 1/4	......	......	......	......	......	22 3/4	16	3/4	7/8	3 1/2
20	.....	27 1/2	1 1/4	......	......	......	......	......	25	20	3/4	7/8	3 1/2
24	.....	32	1 3/8	......	......	......	......	......	29 1/2	20	3/4	7/8	3 3/4
30	.....	38 3/4	1 1/2	......	......	......	......	......	36	28	7/8	1	4 1/4
36	.....	46	1 5/8	......	......	......	......	......	42 3/4	32	7/8	1	5
42	.....	53	1 3/4	......	......	......	......	......	49 1/2	36	1	1 1/8	5 1/4
48	.....	59 1/2	2	......	......	......	......	......	56	44	1	1 1/8	5 1/2
54	.....	66 1/4	2 1/4	......	......	......	......	......	62 3/4	44	1	1 1/8	5 3/4
60	.....	73	2 1/4	......	......	......	......	......	69 1/4	52	1 1/8	1 1/4	6
72	.....	86 1/2	2 1/2	......	......	......	......	......	82 1/2	60	1 1/8	1 1/4	6 1/4
Maximum Pressure, 125 lb. per sq. in. (A.S.A. B16a, 1928). See Fig. 23.													
		4 1/4	7/16			1	0.68	0.68	3 1/8		1/2		1 1/2
1 1/4		4 5/8	1/2			[illegible] /16	.76	.76	3 1/2		1/2		1 1/2
1 1/2		5	9/1			2 9/16	.87	.87	3 7/8		1/2		1 3/4
2		6				3 1/16	1.00	1.00	4 3/4		5/8	3/4	2
2 1/2		7				3 9/16	1.14	1.14	5 1/2		5/	3/4	2 1/4
3		7 1/2				4 1/	1.20	1.20	6		5/8	3/4	2 1/4
3 1/2		8 1/2				4 13/1	1.25	1.25	7		5/8	3/	2 1/2
4		9				5 5/16	1.30	1.30	7 1/2		5/8	3/	2 3/4
5		10				6 7/16	1.41	1.41	8 1/2		3/4	7/8	2 3/4
6		11				7 9/16	1.51	1.51	9 1/2		3/4	7/8	3
8		13 1/2	1/8			9 11/	1.71	1.71	13/4		3/4	7/8	3 1/4
10		16	3/16			11 15/1	1.93	1.93	14 1/4		7/	1	3 1/2
12		19	1/4		13/16	14 1/16	2.13	2.13	17		7/	1	3 1/2
14 O.D.		21	3/8		7/	15 3/8	2.25	2.25	3/4			1 1/8	4
16 O.D.						17 1/2	2.45	2.45	1/4			1 1/8	4 1/4
18 O.D.					1 1/16	19 5/8	2.65	2.65	22 3/4		1 1/8	1 1/4	4 1/2
20 O.D.		2			1 1/8	21 3/4	2.85	2.85	25		1 1/8	1 1/4	4 3/4
24 O.D.		3			1 1/4	26	3.25	3.25			1 1/4	1 3/8	5 1/4
30 O.D.		38 3/4	2 1/8		1 7/16						1 1/4	1 3/8	5 3/4
36 O.D.		46	2 3/8		1 5/8				42 3/4		1 1/2	1 5	6 1/2
42 O.D.		53	2 5/8		1 13/				49 1/2		1 1/2	1	7 1/4
48 O.D.		59 1/2	2 3/4						56		1 1·2		7 1/2
Maximum Pressure, 250 lb. per sq. in. (A.S.A. B16b, 1928). See Fig. 24.													
1	1	4 7/8	11/16	2 11/16	......	2 1/16	7/8	0.68	3 1/2	4	5/8	3/4	2 1/4
1 1/4	1 1/4	5 1/4	3/4	3 1/16	......	2 1/2	1	.76	3 7/8	4	5/8	3/4	2 1/2
1 1/2	1 1/2	6 1/8	13/16	3 9/16	......	2 3/4	1 1/8	.87	4 1/2	4	3/4	7/8	2 1/2
2	2	6 1/2	7/8	4 3/16	......	3 5/16	1 1/4	1.00	5	8	5/8	3/4	2 1/2
2 1/2	2 1/2	7 1/2	1	4 15/16	......	3 15/16	1 7/16	1.14	5 7/8	8	3/4	7/8	3
3	3	8 1/4	1 1/8	5 11/16	......	4 5/8	9/16	1.20	6 5/8	8	3/4	7/8	3 1/4
3 1/2	3 1/2	9	3 3/16	6 5/16	......	5 1/4	1 5/8	1.25	7 1/4	8	3/4	7/8	3 1/4
4	4	10	1 1/4	6 15/16	......	5 3/4	1 3/4	1.30	7 7/8	8	3/4	7/8	3 1/2
5	5	11	1 3/8	8 5/16	......	7	1 7/8	1.41	9 1/4	8	3/4	7/8	3 3/4
6	6	12 1/2	1 7/16	9 11/16	......	8 1/8	1 15/16	1.51	10 5/8	12	3/4	7/8	3 3/4
8	8	15	1 5/8	11 15/16	......	10 1/4	2 3/16	1.71	13	12	7/8	1	4 1/4
10	10	17 1/2	1 7/8	14 1/16	15/16	12 5/8	2 3/8	1.92	15 1/4	16	1	1 1/8	5
12	12	20 1/2	2	16 7/16	1	14 3/4	2 9/16	2.12	17 3/4	16	1 1/8	1 1/4	5 1/2
14 O.D.	13 1/4	23	2 1/8	18 15/16	1 1/8	16 1/4	2 11/16	2.25	20 1/4	20	1 1/8	1 1/4	5 3/4
16 O.D.	15 1/4	25 1/2	2 1/4	21 1/16	1 1/4	18 3/8	2 7/8	2.45	22 1/2	20	1 1/4	1 3/8	6
18 O.D.	17	28	2 3/8	23 5/16	1 3/8	20 5/8	3 1/8	2.65	24 3/4	24	1 1/4	1 3/8	6 1/4
20 O.D.	19	30 1/2	2 1/2	25 9/16	1 1/2	22 3/4	3 5/16	2.85	27	24	1 1/4	1 3/8	6 1/2
24 O.D.	23	36	2 3/4	30 5/16	1 5/8	27 3/8	3 11/16	3.25	32	24	1 1/2	1 5/8	7 1/2
30 O.D.	29	43	3	37 3/16	2	......	......	......	39 1/4	28	1 3/4	2	8 1/4

Metal thickness of screwed fittings shall not be more than 10% below the minimum as given in the tables.

Drilling templates for flanges are made in multiples of four so that fittings may face in any quarter. Bolt holes straddle the center line and are drilled larger than the bolts,

Table 34.—Dimensions of Cast-iron Flanged Fittings (Am. Stds. Assoc.)

All dimensions in inches. See Fig. 25

Nominal Pipe Size	Minimum Inside Diameter of Fitting		Face to Face, Tees, *AA*	Center to Face, Long Radius Elbow *B*	Center to Face, 45-deg. Elbow	Face to Face, Lateral	Center to Face Lateral	Center to Face, Lateral, *Y*	Face to Face, Reducer *G*	Diameter of Flange	mum Thickness of Flange	Diameter of Raised Face	Thick- Body Metal, minimum
Maximum Pressure, 25 lb. per sq. in., gage (A.S.A. B16b2, 1931)													
4	......	6 1/2	13	9	4	......	......	......	......	9	3/4	......	0.42
5	......	7 1/2	15	10 1/4	4 1/2	......	......	......	......	10	3/4	......	.44
6	......	8	16	11 1/2	5	......	......	......	......	11	3/4	......	.44
8	......	9	18	14	5 1/2	......	......	......	......	13 1/2	3/4	......	.46
10	......	11	22	16 1/2	6 1/2	......	......	......	......	16	7/8	......	.50
12	......	12	24	19	7 1/2	......	......	......	......	19	1	......	.54
14	......	14	28	21 1/2	7 1/2	......	......	......	......	21	1 1/8	......	.57
16	......	15	30	24	8	......	......	......	......	23 1/2	1 1/8	......	.60
18	......	16 1/2	33	26 1/2	8 1/2	......	......	......	......	25	1 1/4	......	.64
20	......	18	36	29	9 1/2	......	......	......	......	27 1/2	1 1/4	......	.67
24	......	22	44	34	11	......	......	......	......	32	1 3/8	......	.76
30	......	25	50	41 1/2	15	......	......	......	......	38 3/4	1 1/2	......	.88
36	......	28	56	49	18	......	......	......	......	46	1 5/8	......	.99
42	......	31	62	56 1/2	21	......	......	......	......	53	1 3/4	......	1.10
48	......	34	68	64	24	......	......	......	......	59 1/2	2	......	1.26
54	......	39	78	71 1/2	27	......	......	......	......	66 1/4	2 1/4	......	1.35
60	......	44	88	79	30	......	......	......	......	73	2 1/4	......	1.39
72	......	53	106	94	36	......	......	......	......	86 1/2	2 1/2	......	1.62
Maximum Pressure, 125 lb. per sq. in., gage (A.S.A. B16a, 1939)													
1	......	3 1/2	7	5	1 3/4	7 1/2	5 3/4	1 3/4	......	4 1/4	7/16	......	5/16
1 1/4	......	3 3/4	7 1/2	5 1/2	2	8	6 1/4	1 3/4	......	4 5/8	1/2	......	5/16
1 1/2	......	4	8	6	2 1/4	9	7	2	......	5	9/16	......	5/16
2	......	4 1/2	9	6 1/2	2 1/2	10 1/2	8	2 1/2	5	6	5/8	......	5/16
2 1/2	......	5	10	7	3	12	9 1/2	2 1/2	5 1/2	7	11/16	......	5/16
3	......	5 1/2	11	7 3/4	3	13	10	3	6	7 1/2	3/4	......	3/8
3 1/2	......	6	12	8 1/2	3 1/2	14 1/2	11 1/2	3	6 1/2	8 1/2	13/16	......	7/16
4	......	6 1/2	13	9	4	15	12	3	7	9	15/16	......	1/2
5	......	7 1/2	15	10 1/4	4 1/2	17	13 1/2	3 1/2	8	10	15/16	......	1/2
6	......	8	16	11 1/2	5	18	14 1/2	3 1/2	9	11	1	......	9/16
8	......	9	18	14	5 1/2	22	17 1/2	4 1/2	11	13 1/2	1 1/8	......	5/8
10	......	11	22	16 1/2	6 1/2	25 1/2	20 1/2	5	12	16	1 3/16	......	3/4
12	......	12	24	19	7 1/2	30	24 1/2	5 1/2	14	19	1 1/4	......	13/16
14 O.D.	......	14	28	21 1/2	7 1/2	33	27	6	16	21	1 3/8	......	7/8
16 O.D.	......	15	30	24	8	36 1/2	30	6 1/2	18	23 1/2	1 7/16	......	1
18 O.D.	......	16 1/2	33	26 1/2	8 1/2	39	32	7	19	25	1 9/16	......	1 1/16
20 O.D.	......	18	36	29	9 1/2	43	35	8	20	27 1/2	1 11/16	......	1 1/8
24 O.D.	......	22	44	34	11	49 1/2	40 1/2	9	24	32	1 7/8	......	1 1/4
30 O.D.	......	25	50	41 1/2	15	59	49	10	30	38 3/4	2 1/8	......	1 7/16
36 O.D.	......	28†	56	49	18	......	......	......	36	46	2 3/8	......	1 5/8
42 O.D.	......	31†	62	56 1/2	21	......	......	......	42	53	2 5/8	......	1 13/16
48 O.D.	......	34†	68	64	24	......	......	......	48	59 1/2	2 3/4	......	2
Maximum Pressure, 250 lb. per sq. in., gage (A.S.A. B16b, 1928)													
1	1	4	8	5	2	8 1/2	6 1/2	2	......	4 7/8	11/16	2 11/16	1/2
1 1/4	1 1/4	4 1/4	8 1/2	5 1/2	2 1/2	9 1/2	7 1/4	2 1/4	......	5 1/4	3/4	3 1/16	1/2
1 1/2	1 1/2	4 1/2	9	6	2 3/4	11	8 1/2	2 1/2	......	6 1/8	13/16	3 9/16	1/2
2	2	5	10	6 1/2	3	11 1/2	9	2 1/2	5	6 1/2	7/8	4 3/16	1/2
2 1/2	2 1/2	5 1/2	11	7	3 1/2	13	10 1/2	2 1/2	5 1/2	7 1/2	1	4 15/16	9/16
3	3	6	12	7 3/4	3 1/2	14	11	3	6	8 1/4	1 1/8	5 11/16	9/16
3 1/2	3 1/2	6 1/2	13	8 1/2	4	15 1/2	12 1/2	3	6 1/2	9	1 3/16	6 5/16	9/16
4	4	7	14	9	4 1/2	16 1/2	13 1/2	3	7	10	1 1/4	6 15/16	5/8
5	5	8	16	10 1/4	5	18 1/2	15	3 1/2	8	11	1 3/8	8 5/16	11/16
6	6	8 1/2	17	11 1/2	5 1/2	21 1/2	17 1/2	4	9	12 1/2	1 7/16	9 11/16	3/4
8	8	10	20	14	6	25 1/2	20 1/2	5	11	15	1 5/8	11 15/16	13/16
10	10	11 1/2	23	16 1/2	7	29 1/2	24	5 1/2	12	17 1/2	1 7/8	14 1/16	15/16
12	12	13	26	19	8	33 1/2	27 1/2	6	14	20 1/2	2	16 7/16	1
14 O.D.	13 1/4	15	30	21 1/2	8 1/2	37 1/2	31	6 1/2	16	23	2 1/8	18 15/16	1 1/8
16 O.D.	15 1/4	16 1/2	33	24	9 1/2	42	34 1/2	7 1/2	18	25 1/2	2 1/4	21 1/16	1 1/4
18 O.D.	17	18	36	26 1/2	10	45 1/2	37 1/2	8	19	28	2 3/8	23 5/16	1 3/8
20 O.D.	19	19 1/2	39	29	10 1/2	49	40 1/2	8 1/2	20	30 1/2	2 1/2	25 9/16	1 1/2
24 O.D.	23	22 1/2	45	34	12	57 1/2	47 1/2	10	24	36	2 3/4	30 5/16	1 5/8
30 O.D.	29	27 1/2†	55	41 1/2	15	......	......	......	30	43	3	37 3/16	2

same
larger
size of

utlet. Sizes 18 in. and larger, reducing on outlet, are made in
outlet. See Table 35. † Applies only to elbows, tees and crosses.

as shown in Table 33. Bolt holes are not spot faced for ordinary service, but when required, flanges in sizes 36 in. and larger may be spot faced or back faced to minimum thickness of flange with a tolerance of $\pm ^1/_8$ in.

Screwed fittings are threaded according to American Standard Pipe Threads (A.S.A. B3–1919, or its latest revision), and variations in tapping and dieing are limited to one turn either way from the standard. See Table 31.

Malleable Fittings.—The same inspection limits apply to both malleable and cast-iron screwed fittings. The addition of lugs or ribs is permitted.

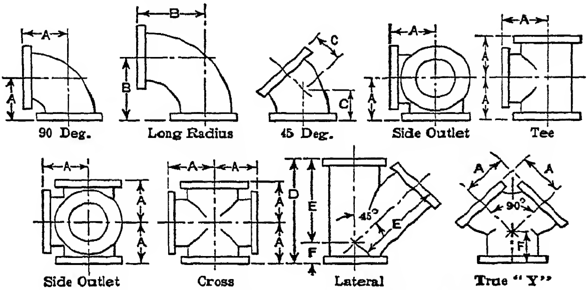


Fig. 25. Cast-iron Straight Size Fittings

Standard Bronze Flanges.—(Adopted May, 1930, by Manufacturers Standardization Society of Valve and Fittings Industry.) Dimensions of standard bronze flanges for valves and fittings are shown in Fig. 33 and Table 42. The material, known as steam-bronze, has the following percentage compositions:

	Cu (min.)	Sn + Zn (min.)	Pb (max.)	Tensile Strength lb. per sq. in. (min.)	Elong. in 2 in.,% (min.)
Grade A......	83.0	8.0	6.0	27,000	15
Grade B......	84.0	9.0	2.5	32,000	15

Carbon steel bolts should have tensile strength of 55,000 lb. per sq. in. (min.) and yield point of 28,000 lb. per sq. in. (min.). If non-corrosive bolts are required the material should have minimum physical properties equal to the required carbon steel bolts. The maximum temperatures that may accompany a given pressure and service condition are:

		Steam	Water	Gas or Liquid	Hydrostatic Tests: Seats	Hydrostatic Tests: Shells
150-lb. Valves	Pressure, lb. per sq. in....	150	200	225	225	375
	Temp., Grade A, deg. F...	366	300	150	125	125
	Temp., Grade B, deg. F...	500	350	150	125	125
250 lb. Valves	Pressure, lb. per sq. in....	250	300	375	375	625
	Temp., Grade A, deg. F...	406	325	150	125	125
	Temp., Grade B, deg. F...	500	350	150	125	125

The faces of flanges should be machined to a smooth finish over their entire face, and two concentric gasket-retaining rings not over $^1/_{16}$ in. deep, and $^1/_{16}$ in. wide, are recommended. Gaskets should be of same diameter as flanges and should be non-metallic.

Raised Faces.—The American Standards provide that all steel and cast-iron flanged valves and fittings shall have raised faces, as follows: 150- and 300-lb. standard, $^1/_{16}$ in. on all sizes, included in the minimum flange thickness; all other standards, $^1/_4$ in., on all sizes, added to the minimum flange thickness; outside diameters of raised face of steel flanges are given in Table 48 and of cast iron flanges in Table 33. Bottom or contact surfaces of groove and female facings are in the same plane as the edge of the flange. Outside diameters of groove and female are to be $^1/_{16}$ in. larger than corresponding outside diameters of tongue and male. A tolerance of ± 0.016 is allowed on inside and outside diameters of all facings. Gaskets for male-female and tongue-groove joints should cover the bottom of joint with minimum clearances. Drilling templates are in multiples of four, and bolt holes straddle the center lines. Bolt holes are drilled $^1/_8$ in. larger diameter than nominal size of bolts. Bolts, or bolt studs threaded at both ends, may be used, with cold-punched or cold pressed semi-finished nuts, American

Standard rough dimensions, chamfered and trimmed. From 1/2 in. through 1 in. diameter, American (National) standard coarse screw thread is used. Over 1 in. diameter, special threads of American form are used, with a pitch of 8 threads per inch. Bolt studs, with a nut on each end are recommended for high temperature work. All flanges are spot faced or back faced parallel to the flange face, except on forged steel flanges where the back

Table 35.—Dimensions of Cast-iron Flanged Reducing Fittings (Short Body Pattern)

American Standards Association

All dimensions in inches. See Figs. 26 and 27.

	Tees and Crosses				Laterals					Base Elbows and Base Tees			
Nominal Size of Pipe	Maximum Size of Outlet	Center to Face, Run *H*	Face to Face, Run *HH*	Center to Face, Outlet *J*	Maximum Size of Branch	Face to Face, Run *L*	Center to Face, Run *M*	Center to Face, Run *N*	Center to Face, Branch *P*	Center to Base *R* *	Diam. of Round Base *S* *	Thickness of Base *T* *	Thickness of Ribs *U* *
25 lb. Maximum Pressure (A.S.A. B16b2, 1931)													
18				15 1/2									
20				17									
24				19									
30				23									
36				26									
42				30									
48				34									
54				37									
60				41									
72				48									
125 lb. Maximum Pressure (A.S.A. B16a, 1928)													
1	1	3 1/2	7	3 1/2	1	7 1/2	5 3/4	1 3/4	5 3/4	3 1/2	3 1/2	7/16	3/8
1 1/4	1 1/4	3 3/4	7 1/2	3 3/4	1 1/4	8	6 1/4	1 3/4	6 1/4	3 5/8	3 1/2	7/16	3/8
1 1/2	1 1/2	4	8	4	1 1/2	9	7	2	7	3 3/4	4 1/4	7/16	1/2
2	2	4 1/2	9	4 1/2	2	10 1/2	8	2 1/2	8	4 1/8	4 5/8	1/2	1/2
2 1/2	2 1/2	5	10	5	2 1/2	12	9 1/2	2 1/2	9 1/2	4 1/2	4 5/8	1/2	1/2
3	3	5 1/2	11	5 1/2	3	13	10	3	10	4 7/8	5	9/16	1/2
3 1/2	3 1/2	6	12	6	3 1/2	14 1/2	11 1/2	3	11 1/2	5 1/4	5	9/16	1/2
4	4	6 1/2	13	6 1/2	4	15	12	3	12	5 1/2	6	5/8	1/2
5	5	7 1/2	15	7 1/2	5	17	13 1/2	3 1/2	13 1/2	6 1/4	7	11/16	5/8
6	6	8	16	8	6	18	14 1/2	3 1/2	14 1/2	7	7	11/16	5/8
8	8	9	18	9	8	22	17 1/2	4 1/2	17 1/2	8 3/8	9	15/16	7/8
10	10	11	22	11	10	25 1/2	20 1/2	5	20 1/2	9 3/4	9	15/16	7/8
12	12	12	24	12	12	30	24 1/2	5 1/2	24 1/2	11 1/4	11	1	1
14 O.D.	14	14	28	14	14	33	27	6	27	12 1/2	11	1	1
16 O.D.	16	15	30	15	16	36 1/2	30	6 1/2	30	13 3/4	11	1	1
18 O.D.	12	13	26	15 1/2	8	26	25	1	27 1/2	15	13 1/2	1 1/8	1 1/8
20 O.D.	14	14	28	17	10	28	27	1	29 1/2	16	13 1/2	1 1/8	1 1/8
24 O.D.	16	15	30	19	12	32	31 1/2	1/2	34 1/2	18 1/2	13 1/2	1 1/8	1 1/8
30 O.D.	20	18	36	23	14	39	39	0	42	......	......	......	......
36 O.D.	24	20	40	26	......	......	......	......	......	......	......	......	......
250 lb. Maximum Pressure (A.S.A. B16b, 1928)													
1	1	4	8	4	1	8 1/2	6 1/2	2	6 1/2	3 3/4	4	5/8	1/2
1 1/4	1 1/4	4 1/4	8 1/2	4 1/4	1 1/4	9 1/2	7 1/4	2 1/4	7 1/4	4	4	5/8	1/2
1 1/2	1 1/2	4 1/2	9	4 1/2	1 1/2	11	8 1/2	2 1/2	8 1/2	4 1/8	4 7/8	11/16	1/2
2	2	5	10	5	2	11 1/2	9	2 1/2	9	4 1/2	5 1/4	3/4	1/2
2 1/2	2 1/2	5 1/2	11	5 1/2	2 1/2	13	10 1/2	2 1/2	10 1/2	4 3/4	5 1/4	3/4	1/2
3	3	6	12	6	3	14	11	3	11	5 1/4	6 1/8	13/16	5/8
3 1/2	3 1/2	6 1/2	13	6 1/2	3 1/2	15 1/2	12 1/2	3	12 1/2	5 5/8	6 1/8	13/16	5/8
4	4	7	14	7	4	16 1/2	13 1/2	3	13 1/2	6	6 1/2	7/8	5/8
5	5	8	16	8	5	18 1/2	15	3 1/2	15	6 3/4	7 1/2	1	3/4
6	6	8 1/2	17	8 1/2	6	21 1/2	17 1/2	4	17 1/2	7 1/2	7 1/2	1	3/4
8	8	10	20	10	8	25 1/2	20 1/2	5	20 1/2	9	10	1 1/4	7/8
10	10	11 1/2	23	11 1/2	10	29 1/2	24	5 1/2	24	10 1/2	10	1 1/4	7/8
12	12	13	26	13	12	33 1/2	27 1/2	6	27 1/2	12	12 1/2	1 7/16	1
14 O.D.	14	15	30	15	14	37 1/2	31	6 1/2	31	13 1/2	12 1/2	1 7/16	1
16 O.D.	16	16 1/2	33	16 1/2	16	42	34 1/2	7 1/2	34 1/2	14 3/4	12 1/2	1 7/16	1 1/8
18 O.D.	12	14	28	17	8	34	31	3	32 1/2	16 1/4	15	1 5/8	1 1/8
20 O.D.	14	15 1/2	31	18 1/2	10	37	34	3	36	17 7/8	15	1 5/8	1 1/4
24 O.D.	16	17	34	21 1/2	12	44	41	3	43	20 3/4	17 1/2	1 7/8	1 1/4
30 O.D.	20	20 1/2	41	25 1/2	......	......	......	......	......	......	......	......	......

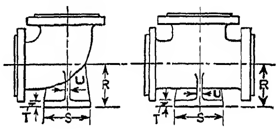


Fig. 26.

* These dimensions apply to straight and reducing sizes and long and short body patterns.

is parallel to the face. Spot facing must not reduce the thickness of the flange below that given in the tables.

Minimum metal thicknesses are based on an allowable fiber stress of 7000 lb. per sq. in., using the modified Barlow formula:

For pipes 1/4 to 5 in. diameter,

$$P = (2S/D)(t - 0.065) - 125,$$

and for pipes over 5 in. diameter, $P = (2S/D)(t - 0.1)$, where P = working pressure, lb.

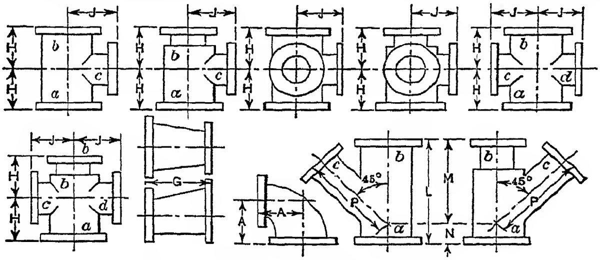


FIG. 27. Cast-iron Reducing Fittings

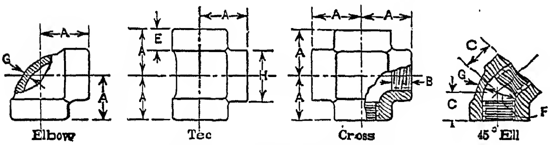


FIG. 28. Malleable Fittings

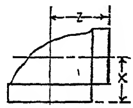


FIG. 29. Malleable Reducing Elbow

Table 36.—Dimensions of 300-lb. Malleable-iron Screw Fittings

Standards of Manufacturers' Standardization Society of Valve and Fitting Industry. (M.S.S. SP-31, 1934)

All dimensions in inches. See Fig. 28.

Nominal Pipe Size	Elbows, Tees and Crosses, Center to End A	45° Ells, Center to End C	Length Thread, min. B	Width of Band, min. E	Inside Diameter of Fitting Min. F	Inside Diameter of Fitting Max. F	Metal Thickness, min. G	Outside Diameter of Band, min. H
1/4	0.94	0.81	0.43	0.38	0.540	0.584	0.14	0.93
3/8	1.06	.88	.47	.44	.675	.719	.15	1.12
1/2	1.25	1.00	.57	.50	.840	.897	.16	1.34
3/4	1.44	1.13	.64	.56	1.050	1.107	.18	1.63
1	1.63	1.31	.75	.62	1.315	1.385	.20	1.95
1 1/4	1.94	1.50	.84	.69	1.660	1.730	.22	2.39
1 1/2	2.13	1.69	.87	.75	1.900	1.970	.24	2.68
2	2.50	2.00	1.00	.84	2.375	2.445	.26	3.28
2 1/2	2.94	2.25	1.17	.94	2.875	2.975	.31	3.86
3	3.38	2.50	1.23	1.00	3.500	3.600	.35	4.62

Table 37.—Dimensions of Malleable Iron and Bronze 90-deg. Reducing Elbows

Standards of Manufacturers' Standardization Society of Valve and Fitting Industry.

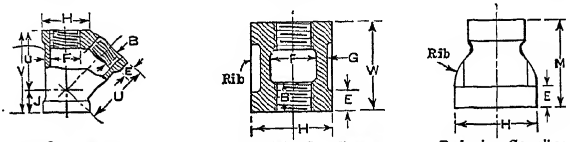
All dimensions in inches. See Fig. 29. For detail dimensions, see Tables 36 and 41.

Nominal Pipe Size	300-lb. Malleable M.S.S. SP-31 Center to end X	Z	125-lb. Bronze M.S.S. SP-10 Center to end X	Z	Nominal Pipe Size	300-lb. Malleable M.S.S. SP-31 Center to end X	Z	125-lb. Bronze M.S.S. SP-10 Center to end X	Z
1/4 × 1/8	......	......	0.65	0.60	1 1/4 × 1	1.75	1.81	1.52	1.60
3/8 × 1/4	......	......	.75	.78	1 1/2 × 1 1/4	2.00	2.06	1.72	1.81
1/2 × 3/8	1.19	1.19	.93	.90	2 × 1 1/2	2.25	2.38	1.89	2.07
3/4 × 1/2	1.31	1.38	1.08	1.11	2 1/2 × 2	2.69	2.75	2.39	2.60
3/4 × 3/8	......	......	1.00	1.00	3 × 2 1/2	3.06	3.31	2.83	2.99
1 × 3/4	1.50	1.56	1.30	1.31	4 × 3	......	......	3.30	3.60
1 × 1/2	......	......	1.20	1.24	......	......	......	......	......

per sq. in., t = thickness of pipe wall, in., D = actual outside diameter of pipe, in., S = fiber stress = 7000 lb. per sq. in.

An inspection limit of $\pm 1/32$ in. is allowed on all center to contact surface dimensions, and of $\pm 1/16$ in. on all contact surface to contact surface dimensions on all sizes to and including 10 in. The corresponding limits for sizes over 10 in. are $\pm 1/16$ in. and $\pm 1/8$ in., respectively. Reducing fittings have the same center to flange edge dimensions as straight size fittings corresponding to the largest opening.

Steel Flanged Fittings.—Physical and chemical requirements of material are given



45° Y-Branch — Straight Coupling — Reducing Coupling

FIG. 30. Malleable Fittings

Table 38.—Dimensions of Cast-iron and Malleable Screwed Fittings

American Standards Association

All dimensions in inches. See Figs. 28 and 30.

Nominal Pipe Size	Center to End			Inside Diameter of Fitting		Width of Band		Metal Thickness, min.		Outside Diameter of Band		Malleable Couplings, Fig. 30			Malleable Y-branches, Fig. 30	
														Length	Center to End	
Maximum Pressures { Cast Iron 125 lb. per sq. in., gage (A.S.A. B16d, 1927); Malleable 150 lb. per sq. in., gage (A.S.A. B16c, 1927)																
1/8 *	0.69	0.68	0.25	0.405	0.435	....	0.200	....	0.900	....	0.693	0.090	0.96	....	0.34	0.97	1.31
1/4	.81	.73	.32	.540	.584	0.38	.215	0.110	.095	0.93	.844	.095	1.06	1.00	.43	1.19	1.62
3/8	.95	.80	.36	.675	.719	.44	.230	.120	.100	1.12	1.015	.100	1.16	1.13	.50	1.43	1.93
1/2	1.12	.88	.43	.840	.897	.50	.249	.130	.105	1.34	1.197	.105	1.34	1.25	.61	1.71	2.32
3/4	1.31	.98	.50	1.050	1.107	.56	.273	.155	.120	1.63	1.458	.120	1.52	1.44	.72	2.05	2.77
1	1.50	1.12	.58	1.315	1.385	.62	.302	.170	.134	1.95	1.771	.134	1.67	1.69	.85	2.43	3.28
1 1/4	1.75	1.29	.67	1.660	1.730	.69	.341	.185	.145	2.39	2.153	.145	1.93	2.06	1.02	2.92	3.94
1 1/2	1.94	1.43	.70	1.900	1.970	.75	.368	.200	.155	2.68	2.427	.155	2.15	2.31	1.10	3.28	4.38
2	2.25	1.68	.75	2.375	2.445	.84	.422	.220	.173	3.28	2.963	.173	2.53	2.81	1.24	3.93	5.17
2 1/2	2.70	1.95	.92	2.875	2.975	.94	.478	.240	.210	3.86	3.589	.210	2.88	3.25	1.52	4.73	6.25
3	3.08	2.17	.98	3.500	3.600	1.00	.548	.260	.231	4.62	4.285	.231	3.18	3.69	1.71	5.55	7.26
3 1/2	3.42	2.39	1.03	4.000	4.100	1.06	.604	.280	.248	5.20	4.843	.248	3.43	4.00	1.85	6.25	8.10
4	3.79	2.61	1.08	4.500	4.600	1.12	.661	.310	.265	5.79	5.401	.265	3.69	4.38	2.01	6.97	8.98
5	4.50	3.05	1.18	5.563	5.663	1.18	.780	.380	.300	7.05	6.583	.300	4.22	5.12	2.34	8.43	10.77
6	5.13	3.46	1.28	6.625	6.725	1.28	.900	.430	.336	8.28	7.767	.336	4.75	5.86	2.66	9.81	12.47
8	6.56	4.28	1.47	8.625	8.725	1.47	1.125	.550	.403	10.63	9.995	.403	5.75	7.25	....	....	....
10†	8.08	5.16	1.68	10.750	10.850	1.68	....	.690	....	13.12	....	....	....	....	....	....	....
12†	9.50	5.97	1.88	12.750	12.850	1.88	....	.800	....	15.47	....	....	....	....	....	....	....
14 O.D.†	10.40	....	2.00	14.000	14.100	2.00	....	.880	....	16.94	....	....	....	....	....	....	....
16 O.D.†	11.82	....	2.20	16.000	16.100	2.20	....	1.000	....	19.30	....	....	....	....	....	....	....
1/4	0.94	0.81	0.43	0.540	0.584	0.49	....	0.18	....	1.17	....	....	....	....	....	....	....
3/8	1.06	.88	.47	.675	.719	.55	....	.18	....	1.36	....	....	....	....	....	....	....
1/2	1.25	1.00	.57	.840	.897	.60	....	.20	....	1.59	....	....	....	....	....	....	....
3/4	1.44	1.13	.64	1.050	1.107	.68	....	.23	....	1.88	....	....	....	....	....	....	....
1	1.63	1.31	.75	1.315	1.385	.76	....	.28	....	2.24	....	....	....	....	....	....	....
1 1/4	1.94	1.50	.84	1.660	1.730	.88	....	.33	....	2.73	....	....	....	....	....	....	....
1 1/2	2.13	1.69	.87	1.900	1.970	.97	....	.35	....	3.07	....	....	....	....	....	....	....
2	2.50	2.00	1.00	2.375	2.445	1.12	....	.39	....	3.74	....	....	....	....	....	....	....
2 1/2	2.94	2.25	1.17	2.875	2.975	1.30	....	.43	....	4.60	....	....	....	....	....	....	....
3	3.38	2.50	1.23	3.500	3.600	1.40	....	.48	....	5.36	....	....	....	....	....	....	....
3 1/2	3.75	2.63	1.28	4.000	4.100	1.49	....	.52	....	5.98	....	....	....	....	....	....	....
4	4.13	2.81	1.33	4.500	4.600	1.57	....	.56	....	6.61	....	....	....	....	....	....	....
5	4.88	3.19	1.43	5.563	5.663	1.74	....	.66	....	7.92	....	....	....	....	....	....	....
6	5.63	3.50	1.53	6.625	6.725	1.91	....	.74	....	9.24	....	....	....	....	....	....	....
8	7.00	4.31	1.72	8.625	8.725	2.24	....	.90	....	11.73	....	....	....	....	....	....	....
10	8.63	5.19	1.93	10.750	10.850	2.58	....	1.08	....	14.37	....	....	....	....	....	....	....
12	10.00	6.00	2.13	12.750	12.850	2.91	....	1.24	....	16.84	....	....	....	....	....	....	....
14 O.D.	11.00	....	2.25	14.000	14.100	3.10	....	1.33	....	18.40	....	....	....	....	....	....	....
16 O.D.	12.50	....	2.45	16.000	16.100	3.45	....	1.50	....	20.88	....	....	....	....	....	....	....

* Malleable fittings only. † Cast iron fittings only.

in Table 44. Nuts are of carbon steel. Washers are of forged or rolled carbon steel. Dimensions of companion flanges are given in Tables 45 and 46. Dimensions of fittings are given in Tables 47 and 48. The dimensional standards for steel castings are based on a product equal to that given in A.S.T.M. specification A95–29 for carbon steel castings for valves, flanges and fittings. For high temperature, castings shall be heat treated. Dimensional standards for steel forgings, other than companion flanges, are based on a product equal to Class C steel given in A.S.T.M. specification No. A105–28 for forged or rolled steel pipe flanges for high temperature service. Class A steel shall be used for forge welding. Class B steel may be used for companion flanges with hubs and is acceptable for fusion welding. Class B steel shall be used for companion flanges when made without hubs. Table 44 is based on the above-named A.S.T.M. specifications.

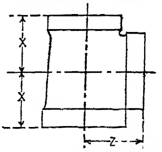


Fig. 31. Malleable Reducing Tee

Table 39.—Dimensions of Malleable-iron and Bronze Screwed Reducing Tees

Standards of Manufacturers' Standardization Society of Valve and Fittings Industry
All dimensions in inches. See Fig. 31. For detail dimensions, see Tables 36 and 38.

Nominal Pipe Size	300-lb. Malleable Iron M.S.S. SP–31			125-lb. Bronze M.S.S. SP–10		
	X	*Y*	*Z*	*X*	*Y*	*Z*
1/8 × 1/8 × 1/4	....	....	....	0.60	0.60	0.65
1/4 × 1/4 × 3/8	....	....	....	.78	.78	.75
1/4 × 1/4 × 1/8	....	....	....	.65	.65	.60
3/8 × 3/8 × 1/2	....	....	....	.90	.90	.93
3/8 × 3/8 × 1/4	....	....	....	.75	.75	.78
3/8 × 1/4 × 3/8	....	....	....	.82	.78	.82
3/8 × 1/4 × 1/4	....	....	....	.75	.71	.78
1/2 × 1/2 × 3/4	....	....	....	1.11	1.11	1.08
1/2 × 1/2 × 3/8	1.19	1.19	1.19	0.93	0.93	0.90
1/2 × 3/8 × 1/2	1.25	1.19	1.25	1.01	.90	1.01
1/2 × 3/8 × 3/8	....	....	....	0.93	.82	0.90
3/4 × 3/4 × 1	....	....	....	1.31	1.31	1.30
3/4 × 3/4 × 1/2	1.31	1.31	1.38	1.08	1.08	1.11
3/4 × 3/4 × 3/8	....	....	....	1.00	1.00	1.00
3/4 × 1/2 × 3/4	1.44	1.38	1.44	1.18	1.11	1.18
3/4 × 1/2 × 1/2	....	....	....	1.08	1.01	1.11
1 × 1 × 1 1/4	....	....	....	1.60	1.60	1.52
1 × 1 × 3/4	1.50	1.50	1.56	1.30	1.30	1.31
1 × 1 × 1/2	1.44	1.44	1.50	1.20	1.20	1.24
1 × 1 × 3/8	1.31	1.31	1.44	....	....	....
1 × 3/4 × 1	1.63	1.56	1.63	1.43	1.31	1.43
1 × 3/4 × 3/4	....	....	....	1.30	1.18	1.31
1 × 3/4 × 1/2	....	....	....	1.20	1.08	1.24
1 × 1/2 × 1	....	....	....	1.43	1.24	1.43
1 1/4 × 1 1/4 × 1 1/2	....	....	....	1.81	1.81	1.72
1 1/4 × 1 1/4 × 1	1.75	1.75	1.81	1.52	1.52	1.60
1 1/4 × 1 1/4 × 3/4	1.63	1.63	1.75	1.39	1.39	1.48
1 1/4 × 1 1/4 × 1/2	1.50	1.50	1.69	1.29	1.29	1.41
1 1/4 × 1 × 1 1/4	1.94	1.81	1.94	1.69	1.60	1.69
1 1/4 × 1 × 1	....	....	....	1.52	1.43	1.60
1 1/4 × 1 × 3/4	....	....	....	1.39	1.30	1.48
1 1/2 × 1 1/2 × 2	....	....	....	2.07	2.07	1.89
1 1/2 × 1 1/2 × 1 1/4	2.00	2.00	2.06	1.72	1.72	1.81
1 1/2 × 1 1/2 × 1	1.81	1.81	2.00	1.55	1.55	1.72
1 1/2 × 1 1/2 × 3/4	1.69	1.69	1.88	1.42	1.42	1.60
1 1/2 × 1 1/2 × 1/2	1.63	1.63	1.81	1.32	1.32	1.53
1 1/2 × 1 1/4 × 1 1/2	2.13	2.06	2.13	1.84	1.81	1.84
1 1/2 × 1 1/4 × 1 1/4	....	....	....	1.72	1.69	1.81
1 1/2 × 1 1/4 × 1	....	....	....	1.55	1.52	1.72
1 1/2 × 3/4 × 1 1/2	....	....	....	1.84	1.60	1.84
2 × 2 × 2 1/2	....	....	....	2.60	2.60	2.39
2 × 2 × 1 1/2	2.25	2.25	2.38	1.89	1.89	2.07
2 × 2 × 1 1/4	2.13	2.13	2.31	1.77	1.77	2.04
2 × 2 × 1	2.00	2.00	2.25	1.59	1.59	1.95
2 × 2 × 3/4	1.81	1.81	2.13	1.47	1.47	1.84
2 × 2 × 1/2	1.75	1.75	2.06	....	....	....
2 × 1 1/2 × 2	2.50	2.38	2.50	2.12	2.07	2.12
2 × 1 1/2 × 1 1/2	....	....	....	1.89	1.84	2.07
2 × 3/4 × 2	....	....	....	2.12	1.84	2.12
2 1/2 × 2 1/2 × 2	2.69	2.69	2.75			
2 1/2 × 2 1/2 × 1 1/2	2.44	2.44	2.63			
2 1/2 × 2 × 2 1/2	2.94	2.75	2.94			
3 × 3 × 2 1/2	3.06	3.06	3.31			
3 × 3 × 2	2.81	2.81	3.12			
3 × 2 1/2 × 3	3.38	3.31	3.38			

Sizes 2 1/2 to 4 in. inclusive are same as American Standards B16c. See Table 38.

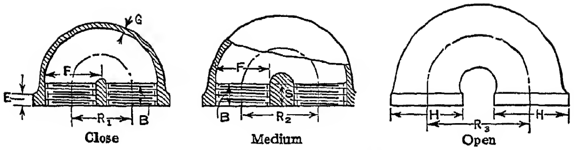


FIG. 32. Return Bends

Table 40.—Dimensions of Malleable Iron and Bronze Return Bends

Am. Standards Assoc. and Manufacturers' Standardization Society of Valve and Fittings Industry
All dimensions in inches. See Fig. 32

Nominal Pipe Size	Malleable, 150-lb. Pressure (A.S.A. B16c, 1927)									Bronze, M.S.S. SP-10*		
	Min. Length Thread	Min. Width Band	Inside Diameter		Metal Thickness	Outside Diam. Band	Center to Center			Center to Center		
			Min.	Max.			Close	Medium	Open	Close	Medium	Open
	B	E	F	F	G	H	R_1	R_2	R_3	R_1	R_2	R_3
1/2	0.43	0.249	0.840	0.897	0.116	1.197	1.000	1.25	1.50	1.00	1.25	1.50
3/4	.50	.273	1.050	1.107	.133	1.458	1.250	1.50	2.00	1.25	1.50	2.00
1	.58	.302	1.315	1.385	.150	1.771	1.500	1.875	2.50	1.50	1.88	2.50
1 1/4	.67	.341	1.660	1.730	.165	2.153	1.750	2.25	3.00	1.75	2.25	3.00
1 1/2	.70	.368	1.900	1.970	.178	2.427	2.188	2.50	3.50	2.19	2.50	3.50
2	.75	.422	2.375	2.445	.201	2.963	2.625	3.00	4.00	2.63	3.00	4.00
2 1/2	.92	.478	2.875	2.975	.244	3.589	3.250	3.75	4.50	3.25	3.75	4.50
3	.98	.548	3.500	3.600	.272	4.285	4.000	4.50	5.00	4.00	4.50	5.00
3 1/2	1.03	.604	4.000	4.100	.295	4.843	4.500	5.00	5.50			
4	1.08	.661	4.500	4.600	.318	5.401	5.000	5.50	6.00			
5	1.18	.780	5.563	5.663	.360	6.583	......	6.50	7.50			
6	1.28	.900	6.625	6.725	.403	7.767	......	7.50	8.50			

* For detail dimensions of Bronze Fittings see Table 41.

Table 41.—Dimensions of 250-lb. and 125-lb. Bronze Screw Fittings

Standards of Manufacturers' Standardization Society of Valve and Fitting Industry
All dimensions in inches. See Figs. 28 and 30

Nominal Pipe Size	All Fittings: Length of Thread, min. B	Width of Band, min.	Inside Diameter of Fitting Min. F	Inside Diameter of Fitting Max.	Metal Thick-	Outside Diameter of Band, min. H	Ells, Tees, Crosses. Center to End	45° Ells. Center to End	Reducers: Reducing 1	Reducers: End to End M_2	Coupling. End to End W	45° Y-branches: Center to End	45° Y-branches: Center to End, Outlet	45° Y-branches: End to End
250-lb. Fittings (M.S.S. SP-11, 1932)														
1/4	0.32	0.38	0.540	0.584	0.110	0.93	0.81	0.73	....	....	....	....	....	....
3/8	.36	.44	.675	.719	.120	1.12	.95	.80	....	....	....	....	....	....
1/2	.43	.50	.840	.897	.130	1.34	1.12	.88	....	....	....	....	....	....
3/4	.50	.56	1.050	1.107	.155	1.63	1.31	.98	....	....	....	....	....	....
1	.58	.62	1.315	1.385	.170	1.95	1.50	1.12	....	....	....	....	....	....
1 1/4	.67	.69	1.660	1.730	.185	2.39	1.75	1.29	....	....	....	....	....	....
1 1/2	.70	.75	1.900	1.970	.200	2.68	1.94	1.43	....	....	....	....	....	....
2	.75	.84	2.375	2.445	.220	3.28	2.25	1.68	....	....	....	....	....	....
2 1/2	.92	.94	2.875	2.975	.240	3.86	2.70	1.95	....	....	....	....	....	....
3	.98	1.00	3.500	3.600	.260	4.62	3.08	2.17	....	....	....	....	....	....
3 1/2	1.03	1.06	4.000	4.100	.280	5.20	3.42	2.39	....	....	....	....	....	....
4	1.08	1.12	4.500	4.600	.310	5.79	3.79	2.61	....	....	....	....	....	....
125-lb. Fittings (M.S.S. SP-10, 1932)														
1/8	0.25	0.14	0.41	0.44	0.08	0.67	0.54	0.42	....	....	0.80	....	....	....
1/4	.32	.16	.54	.58	.08	.81	.71	.56	0.88	....	.97	....	....	....
3/8	.36	.17	.68	.72	.09	1.00	.82	.63	1.01	0.92	1.05	0.50	1.28	1.75
1/2	.43	.19	.84	.90	.09	1.17	1.01	.78	1.17	1.13	1.29	.61	1.58	2.19
3/4	.50	.23	1.05	1.11	.10	1.42	1.18	.89	1.36	1.24	1.43	.72	1.90	2.62
1	.58	.27	1.32	1.39	.11	1.72	1.43	1.06	1.56	1.49	1.68	.85	2.33	3.18
1 1/4	.67	.31	1.66	1.73	.12	2.10	1.69	1.22	1.77	1.65	1.86	1.02	2.83	3.85
1 1/2	.70	.34	1.90	1.97	.13	2.38	1.84	1.30	1.89	1.80	1.92	1.10	3.14	4.24
2	.75	.41	2.38	2.45	.15	2.92	2.12	1.45	2.06	2.03	2.20	1.24	3.76	5.00
2 1/2	.92	.48	2.88	2.97	.17	3.49	2.70	1.95	3.25	3.25	2.88	1.52	4.73	6.25
3	.98	.55	3.50	3.60	.19	4.20	3.08	2.17	3.69	3.69	3.18	1.71	5.55	7.26
3 1/2	1.03	.60	4.00	4.10	.20	4.75	3.42	2.39	4.00	4.00	3.43	1.85	6.25	8.10
4	1.08	.66	4.50	4.60	.22	5.31	3.7	2.61	4.38	4.38	3.69	2.01	6.97	8.98

Table 42.—Dimensions of Standard Bronze Flanges

Pipe Size, In.	A (min.)	B	C	D	E	F	Bolt Diameter	Number of Bolts
	Dimensions in Inches. See Fig. 33							
	For 150 lb. per sq. in. Steam Pressure (M.S.S. SP–2a–1930)							
1/2	3 1/2	5/16	2 3/8	1 3/8	1 1/16	5/8	1/2	4
3/4	3 7/8	11/32	2 3/4	1 3/4	1 1/4	5/8	1/2	4
1	4 1/4	3/8	3 1/8	2 1/8	1 5/8	5/8	1/2	4
1 1/4	4 5/8	13/32	3 1/2	2 1/2	2	5/8	1/2	4
1 1/2	5	7/16	3 7/8	2 7/8	2 3/8	5/8	1/2	4
2	6	1/2	4 3/4	3 1/2	2 7/8	3/4	5/8	4
2 1/2	7	9/16	5 1/2	4 1/4	3 5/8	3/4	5/8	4
3	7 1/2	5/8	6	4 3/4	4 1/8	3/4	5/8	4
3 1/2	8 1/2	11/16	7	5 3/4	5 1/8	3/4	5/8	8
4	9	11/16	7 1/2	6 1/4	5 5/8	3/4	5/8	8
5	10	3/4	8 1/2	7	6 1/4	7/8	3/4	8
6	11	13/16	9 1/2	8	7 1/4	7/8	3/4	8
8	13 1/2	15/16	11 3/4	10 1/4	9 1/2	7/8	3/4	8
10	16	1	14 1/4	12 1/2	11 1/2	1	7/8	12
12	19	1 1/16	17	15 1/4	14 1/4	1	7/8	12
	For 250 lb. per sq. in. Steam Pressure (M.S.S. SP–2b–1930)							
1/2	3 3/4	13/32	2 5/8	1 5/8	1 1/8	5/8	1/2	4
3/4	4 5/8	7/16	3 1/4	2	1 3/8	3/4	5/8	4
1	4 7/8	1/2	3 1/2	2 1/4	1 5/8	3/4	5/8	4
1 1/4	5 1/4	17/32	3 7/8	2 5/8	2	3/4	5/8	4
1 1/2	6 1/8	9/16	4 1/2	3	2 1/4	7/8	3/4	4
2	6 1/2	5/8	5	3 3/4	3 1/8	3/4	5/8	8
2 1/2	7 1/2	11/16	5 7/8	4 3/8	3 5/8	7/8	3/4	8
3	8 1/4	3/4	6 5/8	5 1/8	4 3/8	7/8	3/4	8
3 1/2	9	13/16	7 1/4	5 3/4	5	7/8	3/4	8
4	10	7/8	7 7/8	6 3/8	5 5/8	7/8	3/4	8
5	11	15/16	9 1/4	7 3/4	7	7/8	3/4	8
6	12 1/2	1	10 5/8	9 1/8	8 3/8	7/8	3/4	12
8	15	1 1/8	13	11 1/4	10 1/4	1	7/8	12
10	17 1/2	1 3/16	15 1/4	13 1/4	12	1 1/8	1	16
12	20 1/2	1 1/4	17 3/4	15 1/2	14	1 1/4	1 1/8	16

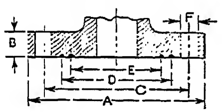


Fig. 33.

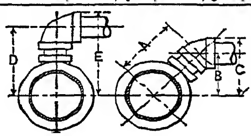


Fig. 34.

Table 43.—Minimum Height of Branch Connections with Screw Fittings

All dimensions in inches. See Fig. 34

Main	Branch	A	B	C	D	E	Main	Branch	A	B	C	D	E
1	3/4	2 13/16	2	2 3/4	3 1/8	3 7/8	3 1/2	2‡	5 3/4	4 1/16	5 9/16	6 13/16	8 1/4
1 1/4	3/4	3	2 1/8	2 7/8	3 5/16	4 1/4	3 1/2	2 1/2	5 13/16	4 1/8	5 15/16	6 5/8	8 3/16
1 1/4	1	3 3/16	2 1/4	3 1/8	3 9/16	4 7/16	3 1/2	3	6 3/4	4 3/4	6 15/16	7 5/8	9 13/16
1 1/2	3/4	3 1/8	2 1/4	2 15/16	3 7/16	4 3/16	4	1 1/2‡	6	4 3/8	5 7/16	6 7/16	7 11/16
1 1/2	1	3 5/16	2 3/8	3 1/4	3 11/16	4 9/16	4	2	5 1/2	3 7/8	5 3/8	6 1/16	7 9/16
1 1/2	1 1/4	3 9/16	2 1/2	3 5/8	4 1/16	5 1/8	4	2 1/2	6 1/8	4 5/16	6 1/4	6 7/8	8 11/16
2	3/4	3 3/8	2 3/8	3 1/4	3 11/16	4 7/16	4	3	6 5/16	4 7/16	6 5/8	7 3/16	9 3/8
2	1	3 9/16	2 1/2	3 3/8	3 15/16	4 1/2	4	3 1/2	7 7/16	5 1/4	6 11/16	8 1/2	10 15/16
2	1 1/4	3 13/16	2 11/16	3 3/4	2 11/16	3 3/4	5	1 1/2*	7 11/16	5 7/16	6 11/16	8 3/16	9 7/16
2	1 1/2	4 1/8	2 15/16	4 1/4	4 5/8	5 13/16	5	2†	7 1/4	5 1/8	6 5/8	7 13/16	9 5/16
2 1/2	3/4	3 11/16	2 5/8	2 11/16	4 1/16	4 3/4	5	2 1/2‡	7 3/16	5 3/16	7	5 5/8	7 7/16
2 1/2	1	3 7/8	2 3/4	3 5/8	4 1/4	5 3/16	5	3	6 15/16	4 7/8	7 1/16	7 13/16	10
2 1/2	1 1/4	4 1/8	2 15/16	4	4 5/8	5 11/16	5	3 1/2†	8 13/16	6 1/4	8 5/8	9 13/16	12 1/4
2 1/2	1 1/2	4 1/2	3 3/16	4 3/16	5	6 3/16	5	4‡	8 5/8	6 1/4	8 13/16	9 13/16	12 1/2
2 1/2	2	4 11/16	3 5/16	4 13/16	5 1/4	5 1/4	6	1 1/4†	7 3/8	5 1/4	6 5/16	7 13/16	8 15/16
3	1	4 3/8	2 15/16	3 1/16	4 9/16	5 7/16	6	1 1/2*	7 1/16	5	6 1/4	7 9/16	8 13/16
3	1 1/4	4 7/16	3 1/8	4 1/4	4 7/8	6	6	2	6 5/8	4 11/16	6 3/8	7 3/16	8 11/16
3	1 1/2	4 3/4	3 3/8	4 9/16	5 1/4	6 1/2	6	2 1/2	7 1/4	5 1/8	6 15/16	8	9 13/16
3	2	5	3 1/2	5	5 9/16	7	6	3	7 7/16	5 1/4	7 1/2	8 3/8	10 1/2
5	2 1/2	5 9/16	3 15/16	5 3/4	6 5/16	8 1/8	6	3 1/2‡	8 5/8	6 1/4	8 1/2	9 5/8	12 1/16
3 1/2	1 1/4*	5 5/16	3 13/16	4 7/8	6 15/16	8 1/16	6	4	8 7/16	6	8 11/16	9 5/8	12 5/16
3 1/2	1 1/2†	6 3/16	4 3/8	5 5/8	6 11/16	7 15/16	6	5‡	9 13/16	6 15/16	10 1/4	11 5/16	14 9/16

* Use 3 bushings in tee. † Use 2 bushings in tee. ‡ Use 1 bushing in tee.

Table 44.—Physical and Chemical Requirements of Steel Pipe Flanges and Fittings

American Standards Association, A.S.A., B16e, 1939

				Chemical Composition, percent					
				C max.		Mn	P max.		Mo
STEEL CASTINGS									
Carbon Steel.	70†	†	22†	0.15–.45	0.06	0.50†	0.05	0.20†	. . . . .
Carbon-Molybdenum Steel.	70†	45†	22†	0.35	0.06	1.00*	0.05	0.20†	0.40†
FLANGE FORGINGS									
150 & 350 lb. pressure, Class I. .	60†	30†	22†	0.35	0.05	0.40–.80	0.05		
150 & 350 lb. pressure, Class II.	70†	36†	18†		0.05	0.40–.80	0.05		
400–2500 lb. pressure,									
Class I, Carbon Steel.	60†	30†	25†		0.05	0.40–.8	0.05		
Class II, Carbon Steel.	70†	36†	22†		0.05	0.40–.8	0.05		
Grade F1, C-Mo Steel.	70†	45†	25†	0.35	0.05	0.30–.8	0.04	0.20–.50	0.40–.60
BOLTING MATERIAL									
Bolts 2 1/2 in. and less:									
Class A.	95†	70†	20†		0.05		0.045		
Class B.	105†	80†	20†		0.05		0.045		
Class C.	125†	105†	16†		0.05		0.045		
Bolts over 2 1/2 in. to 4 in. incl.									
Class A.	90†	65†	20†		0.05		0.045		
Class B.	100†	75†	20†		0.05		0.045		
	115†	95†	16†		0.05		0.045		

* Maximum. † Minimum.

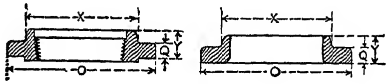


FIG. 35.

Table 45.—Dimensions of Steel Companion Flanges

American Standards Association (A.S.A. B16e, 1939). All dimensions in inches. See Fig. 35

Nominal Pipe Size	Outside Diameter O	Thickness,* Min. Q	Diameter of Hub † X	Length through Hub: Screwed Y	Length through Hub: Lapped Y	Diameter of Bolt Circle	Bolts: Number	Bolts: Size
Maximum Gage Pressure, 150 lb. per sq. in. Maximum Temperature, 500° F.								
1/2	3 1/2	7/16	1 3/16	5/8	5/8	2 3/8	4	1/2
3/4	3 7/8	1/2	1 1/2	5/8	5/8	2 3/4	4	1/2
1	4 1/4	9/16	1 15/16	11/16	11/16	3 1/8	4	1/2
1 1/4	4 5/8	5/8	2 5/16	13/16	13/16	3 1/2	4	1/2
1 1/2	5	11/16	2 9/16	7/8	7/8	3 7/8	4	1/2
2	6	3/4	3 1/16	1	1	4 3/4	4	5/8
2 1/2	7	7/8	3 9/16	1 1/8	1 1/8	5 1/2	4	5/8
3	7 1/2	15/16	4 1/4	1 3/16	1 3/16	6	4	5/8
3 1/2	8 1/2	15/16	4 13/16	1 1/4	1 1/4	7	8	5/8
4	9	15/16	5 5/16	1 5/16	1 5/16	7 1/2	8	5/8
5	10	15/16	6 7/16	1 7/16	1 7/16	8 1/2	8	3/4
6	11	1	7 9/16	1 9/16	1 9/16	9 1/2	8	3/4
8	13 1/2	1 1/8	9 11/16	1 3/4	1 3/4	11 3/4	8	3/4
10	16	1 3/16	12	1 15/16	1 15/16	14 1/4	12	7/8
12	19	1 1/4	14 3/8	2 3/16	2 3/16	17	12	7/8
14 O.D.	21	1 3/8	15 3/4	2 1/4	3 1/8	18 3/4	12	1
16 O.D.	23 1/2	1 7/16	18	2 1/2	3 7/16	21 1/4	16	1
18 O.D.	25	1 9/16	19 7/8	2 11/16	3 13/16	22 3/4	16	1 1/8
20 O.D.	27 1/2	1 11/16	22	2 7/8	4 1/16	25	20	1 1/8
24 O.D.	32	1 7/8	26 1/8	3 1/4	4 3/8	29 1/2	20	1 1/4
Maximum Pressure, 300 lb. per sq. in. Maximum Temperature, 750° F.‡								
2	6 1/2	7/8	3 5/16	1 5/16	1 5/16	5	8	5/8
2 1/2	7 1/2	1	3 15/16	1 1/2	1 1/2	5 7/8	8	3/4
3	8 1/4	1 1/8	4 5/8	1 11/16	1 11/16	6 5/8	8	3/4
3 1/2	9	1 3/16	5 1/4	1 3/4	1 3/4	7 1/4	8	3/4
4	10	1 1/4	5 3/4	1 7/8	1 7/8	7 7/8	8	3/4
5	11	1 3/8	7	2	2	9 1/4	8	3/4
6	12 1/2	1 7/16	8 1/8	2 1/16	2 1/16	10 5/8	12	3/4
8	15	1 5/8	10 1/4	2 7/16	2 7/16	13	12	7/8
10	17 1/2	1 7/8	12 5/8	2 5/8	3 3/4	15 1/4	16	1
12	20 1/2	2	14 3/4	2 7/8	4	17 3/4	16	1 1/8
14 O.D.	23	2 1/8	16 3/4	3	4 3/8	20 1/4	20	1 1/8
16 O.D.	25 1/2	2 1/4	19	3 1/4	4 3/4	22 1/2	20	1 1/4
18 O.D.	28	2 3/8	21	3 1/2	5 1/8	24 3/4	24	1 1/4
20 O.D.	30 1/2	2 1/2	23 1/8	3 3/4	5 1/2	27	24	1 1/4
24 O.D.	36	2 3/4	27 5/8	4 3/16	6	32	24	1 1/2

See end of table for footnotes.

Table 45.—Dimensions of Steel Companion Flanges—*Continued*

Nominal Pipe Size	Outside Diameter O	Thickness,* Min.	Diameter of Hub† X	Length through Hub: Screwed Y	Length through Hub: Lap Y	Diameter of Bolt Circle	Bolts: Number	Bolts: Size
Maximum Pressure, 400 lb. per sq. in. Maximum Temperature, 750° F.§								
4	10	1 3/8	5 3/4	2	2	7 7/8	8	7/8
5	11	1 1/2	7	2 1/8	2 1/8	9 1/4	8	7/8
6	12 1/2	1 5/8	8 1/8	2 1/4	2 1/4	10 5/8	12	7/8
8	15	1 7/8	10 1/4	2 11/16	2 11/16	13	12	1
10	17 1/2	2 1/8	12 5/8	2 7/8	4	15 1/4	16	1 1/8
12	20 1/2	2 1/4	14 3/4	3 1/8	4 1/4	17 3/4	16	1 1/4
14 O.D.	23	2 3/8	16 3/4	3 5/16	4 5/8	20 1/4	20	1 1/4
16 O.D.	25 1/2	2 1/2	19	3 11/16	5	22 1/2	20	1 3/8
18 O.D.	28	2 5/8	21	3 7/8	5 3/8	24 3/4	24	1 3/8
20 O.D.	30 1/2	2 3/4	23 1/8	4	5 3/4	27	24	1 1/2
24 O.D.	36	3	27 5/8	4 1/2	6 1/4	32	24	1 3/4
Maximum Pressure, 600 lb. per sq. in. Maximum Temperature, 750° F.								
1/2	3 3/4	9/16	1 1/2	7/8	7/8	2 5/8	4	1/2
3/4	4 5/8	5/8	1 7/8	1	1	3 1/4	4	5/8
1	4 7/8	11/16	2 1/8	1 1/16	1 1/16	3 1/2	4	5/8
1 1/4	5 1/4	13/16	2 1/2	1 1/8	1 1/8	3 7/8	4	5/8
1 1/2	6 1/8	7/8	2 3/4	1 1/4	1 1/4	4 1/2	4	3/4
2	6 1/2	1	3 5/16	1 7/16	1 7/16	5	8	5/8
2 1/2	7 1/2	1 1/8	3 15/16	1 5/8	1 5/8	5 7/8	8	3/4
3	8 1/4	1 1/4	4 5/8	1 13/16	1 13/16	6 5/8	8	3/4
3 1/2	9	1 3/8	5 1/4	1 15/16	1 15/16	7 1/4	8	7/8
4	10 3/4	1 1/2	6	2 1/8	2 1/8	8 1/2	8	7/8
5	13	1 3/4	7 7/16	2 3/8	2 3/8	10 1/2	8	1
6	14	1 7/8	8 3/4	2 5/8	2 5/8	11 1/2	12	1
8	16 1/2	2 3/16	10 3/4	3	3	13 3/4	12	1 1/8
10	20	2 1/2	13 1/2	3 3/8	4 3/8	17	16	1 1/4
12	22	2 5/8	15 3/4	3 5/8	4 5/8	19 1/4	20	1 1/4
14 O.D.	23 3/4	2 3/4	17	3 11/16	5	20 3/4	20	1 3/8
16 O.D.	27	3	19 1/2	4 3/16	5 1/2	23 3/4	20	1 1/2
18 O.D.	29 1/4	3 1/4	21 1/2	4 5/8	6	25 3/4	20	1 5/8
20 O.D.	32	3 1/2	24	5	6 1/2	28 1/2	24	1 5/8
24 O.D.	37	4	28 1/4	5 1/2	7 1/4	33	24	1 7/8
Maximum Pressure, 900 lb. per sq. in. Maximum Temperature, 750° F.¶								
3	9 1/2	1 1/2	5	2 1/8	2 1/8	7 1/2	8	7/8
4	11 1/2	1 3/4	6 1/4	2 3/4	2 3/4	9 1/4	8	1 1/8
5	13 3/4	2	7 1/2	3 1/8	3 1/8	11	8	1 1/4
6	15	2 3/16	9 1/4	3 3/8	3 3/8	12 1/2	12	1 1/8
8	18 1/2	2 1/2	11 3/4	4	4 1/2	15 1/2	12	1 3/8
10	21 1/2	2 3/4	14 1/2	4 1/4	5	18 1/2	16	1 3/8
12	24	3 1/8	16 1/2	4 5/8	5 5/8	21	20	1 3/8
14 O.D.	25 1/4	3 3/8	17 3/4	5 1/8	6 1/8	22	20	1 1/2
16 O.D.	27 3/4	3 1/2	20	5 1/4	6 1/2	24 1/4	20	1 5/8
18 O.D.	31	4	22 1/4	6	7 1/2	27	20	1 7/8
20 O.D.	33 3/4	4 1/4	24 1/2	6 1/4	8 1/4	29 1/2	20	2
24 O.D.	41	5 1/2	29 1/2	8	10 1/2	35 1/2	20	2 1/2
1	5 7/8	1 1/8	2 1/16	1 5/8	1 5/8	4	4	7/8
1 1/4	6 1/4	1 1/8	2 1/2	1 5/8	1 5/8	4 3/8	4	7/8
1 1/2	7	1 1/4	2 3/4	1 3/4	1 3/4	4 7/8	4	1
2	8 1/2	1 1/2	4 1/8	2 1/4	2 1/4	6 1/2	8	7/8
2 1/2	9 5/8	1 5/8	4 7/8	2 1/2	2 1/2	7 1/2	8	1
3	10 1/2	1 7/8	5 1/4	2 7/8	2 7/8	8	8	1 1/8
4	12 1/4	2 1/8	6 3/8	3 9/16	3 9/16	9 1/2	8	1 1/4
5	14 3/4	2 7/8	7 3/4	4 1/8	4 1/8	11 1/2	8	1 1/2
6	15 1/2	3 1/4	9	4 11/16	4 11/16	12 1/2	12	1 3/8
8	19	3 5/8	11 1/2	5 5/8	5 5/8	15 1/2	12	1 5/8
10	23	4 1/4	14 1/2	6 1/4	7	19	12	1 7/8
12	26 1/2	4 7/8	17 3/4	7 1/8	8 5/8	22 1/2	16	2
Maximum Pressure, 2500 lb. per sq. in. Maximum Temperature, 750° F.								
1/2	5 1/4	1 3/16	1 11/16	1 9/16	1 9/16	3 1/2	4	3/4
3/4	5 1/2	1 1/4	2	1 11/16	1 11/16	3 3/4	4	3/4
1	6 1/4	1 3/8	2 1/4	1 7/8	1 7/8	4 1/4	4	7/8
1 1/4	7 1/4	1 1/2	2 7/8	2 1/16	2 1/16	5 1/8	4	1
1 1/2	8	1 3/4	3 1/8	2 3/8	2 3/8	5 3/4	4	1 1/8
2	9 1/4	2	3 3/4	2 3/4	2 3/4	6 3/4	8	1
2 1/2	10 1/2	2 1/4	4 1/2	3 1/8	3 1/8	7 3/4	8	1 1/8
3	12	2 5/8	5 1/4	3 5/8	3 5/8	9	8	1 1/4
4	14	3	6 1/2	4 1/4	4 1/4	10 3/4	8	1 1/2
5	16 1/2	3 5/8	8	5 1/8	5 1/8	12 3/4	8	1 3/4
6	19	4 1/4	9 1/4	6	6	14 1/2	8	2
8	21 3/4	5	12	7	7	17 1/4	12	2
10	26 1/2	6 1/2	14 3/4	9	9	21 1/4	12	2 1/2
12	30	7 1/4	17 3/8	10	10	24 3/8	12	2 3/4

* A raised face of 1/16 in. is included in min. thickness of flange of 150- and 300-lb. flanges, but is not included in the flanges for higher pressures. † This dimension is for large end of hub, which may be tapered 5 deg. for draft. ‡ For sizes below 2 in., use dimensions of 600-lb. flanges. § For sizes below 4 in. use dimensions of 600-lb. flanges. ¶ For sizes below 3 in., use dimensions of 1500-lb. flanges.

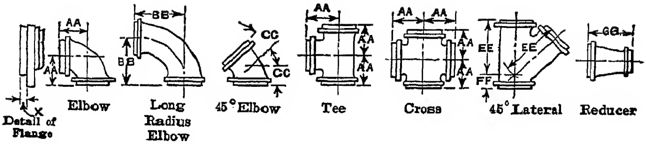


FIG. 36.

Table 46.—Dimensions of Steel Flanged Fittings, with Projecting Faces

American Standards Association (A.S.A. B16e, 1939)

All dimensions in inches. See Fig. 36. For dimensions of Base Fittings see Table 47. For dimensions of Faces, see Table 48.

Nominal Pipe Size	Fitting: Inside Diameter	Fitting: Min. Thick-Metal	Flange: Outside Diameter	Flange: Min. Thick-	Flange: Diameter of Bolt Circle	Bolts: Number	Bolts	Center to Contact Surface of Raised Face*: Elbow, Tee, Cross	Long Radius Ell BB	45-deg. Ell CC	Lateral EE	Lateral FF	Contact Surface to Contact Surface, Reducer* GG
Maximum Pressure of 150 lb. per sq. in., gage, at 500° F., and 100 lb. per sq. in. at 750° F. For sizes larger and smaller than those given below, use same dimensions as for 125-lb. cast-iron flanges and flanged fittings. See Table 34.													
2	2	1/4	6	5/8	4 3/4	4	5/8	4 1/2	6 1/2	2 1/2	8	2 1/2	5
2 1/2	2 1/2	1/4	7	11/16	5 1/2	4	5/8	5	7	3	9 1/2	2 1/2	5 1/2
3	3	1/4	7 1/2	3/4	6	4	5/8	5 1/2	7 3/4	3	10	3	6
3 1/2	3 1/2	1/4	8 1/2	13/16	7	8	5/8	6	8 1/2	3 1/2	11 1/2	3	6 1/2
4	4	1/4	9	15/16	7 1/2	8	5/8	6 1/2	9	4	12	3	7
5	5	9/32	10	15/16	8 1/2	8	3/4	7 1/2	10 1/4	4 1/2	13 1/2	3 1/2	8
6	6	9/32	11	1	9 1/2	8	3/4	8	11 1/2	5	14 1/2	3 1/2	9
8	8	5/16	13 1/2	1 1/8	11 3/4	8	3/4	9	14	5 1/2	17 1/2	4 1/2	11
10	10	11/32	16	1 3/16	14 1/4	12	7/8	11	16 1/2	6 1/2	20 1/2	5	12
12	12	3/8	19	1 1/4	17	12	7/8	12	19	7 1/2	24 1/2	5 1/2	14
14 O.D.	13 1/4	13/32	21	1 3/8	18 3/4	12	1	14	21 1/2	7 1/2	27	6	16
16 O.D.	15 1/4	7/16	23 1/2	1 7/16	21 1/4	16	1	15	24	8	30	6 1/2	18
18 O.D.	17 1/4	15/32	25	1 9/16	22 3/4	16	1 1/8	16 1/2	26 1/2	8 1/2	32	7	19
20 O.D.	19 1/4	1/2	27 1/2	1 11/16	25	20	1 1/8	18	29	9 1/2	35	8	20
24 O.D.	23 1/4	9/16	32	1 7/8	29 1/2	20	1 1/4	22	34	11	40 1/2	9	24
Maximum Pressure, 300 lb. per sq. in., gage, at 750° F. For sizes below 2 in., use the dimensions of 600-lb. fittings.													
2	2	1/4	6 1/2	7/8	5	8	5/8	5	6 1/2	3	9	2 1/2	5
2 1/2	2 1/2	1/4	7 1/2	1	5 7/8	8	3/4	5 1/2	7	3 1/2	10 1/2	2 1/2	5 1/2
3	3	9/32	8 1/4	1 1/8	6 5/8	8	3/4	6	7 3/4	3 1/2	11	3	6
3 1/2	3 1/2	9/32	9	1 3/16	7 1/4	8	3/4	6 1/2	8 1/2	4	12 1/2	3	6 1/2
4	4	5/16	10	1 1/4	7 7/8	8	3/4	7	9	4 1/2	13 1/2	3	7
5	5	3/8	11	1 3/8	9 1/4	8	3/4	8	10 1/4	5	15	3 1/2	8
6	6	3/8	12 1/2	1 7/16	10 5/8	12	3/4	8 1/2	11 1/2	5 1/2	17 1/2	4	9
8	8	7/16	15	1 5/8	13	12	7/8	10	14	6	20 1/2	5	11
10	10	1/2	17 1/2	1 7/8	15 1/4	16	1	11 1/2	16 1/2	7	24	5 1/2	12
12	12	9/16	20 1/2	2	17 3/4	16	1 1/8	13	19	8	27 1/2	6	14
14 O.D.	13 1/4	5/8	23	2 1/8	20 1/4	20	1 1/8	15	21 1/2	8 1/2	31	6 1/2	16
16 O.D.	15 1/4	11/16	25 1/2	2 1/4	22 1/2	20	1 1/4	16 1/2	24	9 1/2	34 1/2	7 1/2	18
18 O.D.	17	3/4	28	2 3/8	24 3/4	24	1 1/4	18	26 1/2	10	37 1/2	8	19
20 O.D.	19	13/16	30 1/2	2 1/2	27	24	1 1/4	19 1/2	29	10 1/2	40 1/2	8 1/2	20
24 O.D.	23	15/16	36	2 3/4	32	24	1 1/2	22 1/2	34	12	47 1/2	10	24
Maximum Pressure, 400 lb. per sq. in., gage, at 750° F. For sizes below 4 in., use the dimensions of 600-lb. fittings.													
4	4	3/8	10	1 3/8	7 7/8		7/8	8		/2	16	4 1/2	8 1/4
5	5	7/16	11	1 1/2	9 1/4		7/8	9		6	16 3/4	5	9 1/4
6	6	7/16	12 1/2	1 5/8	10 5/8		7/8	9 3/4		6 1/4	18 3/4	5 1/4	10
8	8	9/16	15	1 7/8	13			11 3/4		6 3/4	22 1/4	5 3/4	12
10	10	11/16	17 1/2	2 1/8	15 1/4		1/8	13 1/4		7 3/4	25 3/4	6 1/4	13 1/2
12	12	3/4	20 1/2	2 1/4	17 3/4		1/4	15		8 3/4	29 3/4	6 1/2	15 1/4
14 O.D.	13 1/8	13/16	23	2 3/8	20 1/4		1/4	16 1/4		9 1/	32 3/4	7	16 1/2
16 O.D.	15	7/8	25 1/2	2 1/2	22 1/2			17 3/4		/4	36 1/4	8	18 1/2
18 O.D.	17	15/16	28	2 5/8	24 3/4			19 1/4		10 3/4	39 1/4	8 1/2	19 1/2
20 O.D.	18 7/8	1 1/16	30 1/2	2 3/4	27			20 3/4		11 1/4	42 3/4	9	21
24 O.D.		1 3/16	36		32			24 1/4		12 3/4	50 1/4	10 1/2	24 1/2

See end of table for footnotes.

Table 46.—Dimensions of Steel Flanged Fittings, with Projecting Faces—*Continued*

Nominal Pipe Size	Fitting: Inside Diameter	Fitting: Min. Thickness of Metal	Flange: Outside Diameter	Flange: Min. Thickness *	Flange: Diameter of Bolt Circle	Bolts: Number	Bolts: Size	Center to Contact Surface of Raised Face*: Elbow, Tee, Cross *AA*	Long Radius Ell *BB*	45-deg. Ell *CC*	Lateral *EE*	Lateral *FF*	Contact Surface to Contact Surface, Reducer* *GG*
Maximum Pressure, 600 lb. per sq. in., gage, at 750° F.													
1/2	1/2	1/4	3 3/4		2 5/8		1/2	3 1/4		2	5 3/4	1 3/4	5
3/4	3/4	1/4	4 5/8		3 1/4		5/8	3 3/4		2 1/2	6 3/4	2	5
1	1	1/4	4 7/8	/16	3 1/2		5/8	4 1/4		2 1/2	7 1/4	2 1/4	5
1 1/4	1 1/4	1/4	5 1/4	13/16	3 7/8		5/8	4 1/2		2 3/4	8	2 1/2	5
1 1/2	1 1/2	1/4	6 1/8	7/8	4 1/2		3/4	4 3/4		3	9	2 3/4	5
2	2	5/16	6 1/2	1	5		5/8	5 3/4		4 1/4	10 1/4	3 1/2	6
2 1/2	2 1/2	3/8	7 1/2	1 1/8	5 7/8		3/4	6 1/2		4 1/2	11 1/2	3 1/2	6 3/4
3	3	3/8	8 1/4	1 1/4	6 5/8		3/4	7		5	12 3/4	4	7 1/4
3 1/2	3 1/2	7/16	9	1 3/8	7 1/4		7/8	7 1/2		5 1/2	14		7 3/4
4	4	1/2	10 3/4	1 1/2	8 1/2		7/8	8 1/2		6	16 1/2		8 3/4
5	5	9/16	13	1 3/4	10 1/2		1	10		7	19 1/2		10 1/4
6	6	5/8	14	1 7/8	11 1/2		1	11		7 1/2	21	/2	11 1/4
8	7 7/8	3/4	16 1/2	2 3/16	13 3/4		1 1/8	13		8 1/2	24 1/2	7	13 1/4
10	9 3/4	7/8	20	2 1/2	17		1 1/4	15 1/2		9 1/2	29 1/2	8	15 3/4
12	11 3/4		22	2 5/8	19 1/4		1 1/4	16 1/2		10	31 1/2	8 1/2	16 3/4
14 O.D.	12 7/8	/8	23 3/4	2 3/4	20 3/4		1 3/8	17 1/2		10 3/4	34 1/4	9	17 3/4
16 O.D.	14 3/4	1 1/4	27	3	23 3/4		1 1/2	19 1/2		11 3/4	38 1/2	10	19 3/4
18 O.D.	16 1/2	1 3/8	29 1/4	3 1/4	25 3/4		1 5/8	21 1/2		12 1/4	42	10 1/2	21 3/4
20 O.D.	18 1/4	/2	32	3 1/2	28 1/2		1 5/8	23 1/2		13	45 1/2	11	23 3/4
24 O.D.	22	1 3/4	37	4	33	24	1 7/8	27 1/2		14 3/4	53	13	27 3/4
im Pressure, 900 lb. per sq. in., gage, at 750° F.													
1/2	1/2	5/16	4 3/4	7/8	3 1/4		3/4	4 1/4		3			
3/4	11/16	5/16	5 1/8	1	3 1/2		3/4	4 1/2		3 1/4			
1	7/8		5 7/8	1 1/8	4		7/8	5		3 1/2	9	2 1/2	
1 1/4	1 1/8		6 1/4	1 1/8	4 3/8		7/8	5 1/2		4	10	3	/4
1 1/2	1 3/8	7/16	7	1 1/4	4 7/8		1	6		4 1/4	11	3 1/2	6 1/4
2	1 7/8	9/16	8 1/2	1 1/2	6 1/2		7/8	7 1/4		4 3/4	13 1/4	4	7 1/4
2 1/2	2 1/4	11/16	9 5/8	1 5/8	7 1/2		1	8 1/4		5 1/4	15 1/4	4 1/2	8 1/4
3	2 7/8	1/2	9 1/2	1 1/2	7 1/2		7/8	7 1/2		5 1/2	14 1/2	4 1/2	7 3/4
4	3 7/8	5/8	11 1/2	1 3/4	9 1/4		1 1/8	9		6 1/2	17 1/2	5 1/2	9 1/4
5	4 3/4	3/4	13 3/4	2	11		1 1/4	11		7 1/2	21	6 1/2	11 1/4
6	5 3/4	13/16	15	2 3/16	12 1/2		1 1/8	12		8	22 1/2	6 1/2	12 1/4
8	7 1/2	1 1/16	18 1/2	2 1/2	15 1/2		1 3/8	14 1/2		9	27 1/2	7 1/2	14 3/4
10	9 3/8	1 1/4	21 1/2	2 3/4	18 1/2		1 3/8	16 1/2		10	31 1/2	8 1/2	16 3/4
12	11 1/8	1 7/16	24	3 1/8	21		1 3/8	19		11	34 1/2	9	17 3/4
14 O.D.	12 1/4	1 9/16	25 1/4	3 3/8	22		1 1/2	20 1/4		11 1/2	36 1/2	9 1/2	19
16 O.D.	14	1 13/1	27 3/4	3 1/2	24 1/4		1 5/8	22 1/4		12 1/2	40 3/4	10 1/2	21
18 O.D.	15 3/4	2	31	4	27		1 7/8	24		13 1/4	45 1/2	12	24 1/2
20 O.D.	17 1/2	2 1/4	33 3/4	4 1/4	29 1/2		2	26		14 1/2	50 1/4	13	26 1/2
24 O.D.	21	2 5/8	41	5 1/2	35 1/2		2 1/2	30 1/2		18	60	15 1/2	30 1/2
Maximum Pressure, 1500 lb. per sq. in., gage, at 750° F.													
1	7/8	3/8	5 7/8	1 1/8	4		7/8	5		3 1/2	9	2 1/2	5
1 1/4	1 1/8	3/8	6 1/4	1 1/8	4 3/8		7/8	5 1/2		4	10	3	5 3/4
1 1/2	1 3/8	7/16	7	1 1/4	4 7/8		1	6		4 1/4	11	3 1/2	6 1/4
2	1 7/8	9/16	8 1/2	1 1/2	6 1/2		7/8	7 1/4		4 3/4	13 1/4	4	7 1/4
2 1/2	2 1/4	11/16	9 5/8	1 5/8	7 1/2		1	8 1/4		5 1/4	15 1/4	4 1/2	8 1/4
3	2 3/4	3/4	10 1/2	1 7/8	8		1 1/8	9 1/4		5 3/4	17 1/4	5	9 1/4
4	3 5/8	1	12 1/4	2 1/8	9 1/2		1 1/4	10 3/4		7 1/4	19 1/4	6	10 3/4
5	4 3/8	1 1/8	14 3/4	2 7/8	11 1/2		1 1/2	13 1/4		8 3/4	23 1/4	7 1/2	13 3/4
6	5 3/8	1 5/16	15 1/2	3 1/4	12 1/2		1 3/8	13 7/8		9 3/8		8 1/8	14 1/2
8	7	1 5/8	19	3 5/8	15 1/2		1 5/8	/8		10 7/8	/8	9 1/8	17
10	8 3/4	2	23	4 1/4	19		1 7/8	19 1/2		12	36	10 1/4	20 1/4
12	10 3/8	2 5/16	26 1/2	4 7/8	22 1/2		2	22 1/4		13 1/4	40 3/4	12	23
14 O.D.†	11 3/8	2 1/2	29 1/2	5 1/4	25		2 1/4	24 3/4		14 1/4	44	12 1/2	25 3/4
16 O.D.†	13	2 7/8	32 1/2	5 3/4	27 3/4		2 1/2	27 1/4		16 1/4	48 1/4	14 3/4	28 1/4
18 O.D.†	14 5/8	3 1/4	36	6 3/8	30 1/2		2 3/4	30 1/4		17 3/4	53 1/4	16 1/2	31 1/2
20 O.D.†	16 3	3 5/8	38 3/4	7	32 3/4		3	32 3/4		18 3/4	57 3/4	17 3/4	34
24 O.D.†	19 5/8	4 1/4	46		39		3 1/2	38 1/4		20 3/4	67 1/4	20 1/2	39 3/4

* See end of table for footnotes.

(*Table continued on following page*)

Table 46.—Dimensions of Steel Flanged Fittings, with Projecting Faces—*Continued*

Nominal Pipe Size	Fitting: Inside Diameter	Fitting: Min. Thick- Metal	Flange: Out- Diam- eter	Flange: Min. Thick-	Flange: Diam- eter of Bolt Circle	Bolts: Num- ber	Center to Contact Surface of Raised Face*: Elbow, Tee, Cross AA	Long Radius Ell BB	45-deg. Ell CC	Lat- eral EE	Lat- eral FF	Contact Surface to Contact Surface, Reducer* GG
					Maximum Pressure, 2500 lb. per sq. in., gage, at 750° F.							
1/2	7/16		5 1/4		3 1/2	3/4	5 3/16					
3/4	9/16	7/16	5 1/2	1 1/4	3 3/4	3/4	5 3/8					
1	3/4	1/2	6 1/4	1 3/8	4 1/4	7/8	6 1/16		4			
1 1/4	1	5/8	7 1/4	1 1/2	5 1/8		6 7/8		4 1/4			
1 1/2	1 1/8	11/16	8	1 3/4	5 3/4	1/8	7 9/16		4 3/4			
2	1 1/2	13/16	9 1/4	2	6 3/4		8 7/8		5 3/4	15 1/4	5 1/4	9 1/2
2 1/2	1 7/8	1	10 1/2	2 1/4	7 3/4	1/8	10		6 1/4	17 1/4	5 3/4	10 1/2
3	2 1/4	1 3/16	12	2 5/8	9	1/4	11 3/8		7 1/4		6 3/4	11 3/4
4	2 7/8	1 7/16	14	3	10 3/4	1/2	13 1/4		8 1/2	23	7 3/4	13 1/2
5	3 5/8	1 13/16	16 1/2	3 5/8	12 3/4	3/4			10	27 1/4	9 1/4	15 3/4
6	4 3/8	2 1/16	19	4 1/4	14 1/2		18		11 1/2	31 1/4	10 1/2	18
8	5 3/4	2 5/8	21 3/4	5	17 1/4		20 1/8		12 3/4	35 1/4	11 3/4	20 1/2
10	7 1/4	3 1/4	26 1/2	6 1/2	21 1/4	2 1/2	25		16	43 1/4	14 3/4	25 1/2
12	8 5/8	3 13/16	30	7 1/4	24 3/8		28			49 1/4	16 1/4	29

* A raised face of 1/16 in. (dimension *X*) is provided on the flange of each opening of the 150-lb. and 300-lb. fittings and is included in the minimum flange thickness, center to contact surface, and contact surface to contact surface dimensions. Reducing fittings have the same center to contact surface dimensions as straight fittings of the largest opening. On the 400-lb., 600-lb., 900-lb., 1500-lb., and 2500-lb. fittings a raised face of 1/4 in. (dimension *X*) *is not* included in the minimum flange thickness, but *is* included in the center to contact surface, and contact surface to contact surface dimensions. When facings other than the 1/16-in. and 1/4-in. raised faces, as given above, are used, the center to flange edge dimensions shall remain unchanged and new center to contact surface or contact surface to contact surface dimensions shall be established to suit the facing used.

Table 47.—Dimensions of Steel Flanged Base Fittings (A.S.A., B16e, 1932)

For dimensions not given see Table 46. All dimensions in inches. See Fig. 26.

Nominal Pipe Size	Center to Base *R*	Diameter of Base † *S*	Thickness of Base *T*	Thickness of Ribs *U*	Drilling for Base: Diameter of Bolt Circle	Drilling for Base: Number of Bolts	Drilling for Base: Size of Bolts
For 300 and 400 lb. per sq. in. at 750° F. For sizes below 4 in. in the 400-lb. fittings use the dimensions of 600-lb. fittings.							
2*	4 1/2	5 1/4	3/4	1/2	3 7/8	4	5/8
2 1/2*	4 3/4	5 1/4	3/4	1/2	3 7/8	4	5/8
3*	5 1/4	6 1/8	13/16	5/8	4 1/2	4	3/4
3 1/2*	5 5/8	6 1/8	13/16	5/8	4 1/2	4	3/4
4	6	6 1/2	7/8	5/8	5	8	5/8
5	6 3/4	7 1/2	1	3/4	5 7/8	8	3/4
6	7 1/2	7 1/2	1	3/4	5 7/8	8	3/4
8	9	10	1 1/4	7/8	7 7/8	8	3/4
10	10 1/2	10	1 1/4	7/8	7 7/8	8	3/4
12	12	12 1/2	1 7/16	1	10 5/8	12	3/4
14 O.D.	13 1/2	12 1/2	1 7/16	1	10 5/8	12	3/4
16 O.D.	14 3/4	12 1/2	1 7/16	1 1/8	10 5/8	12	3/4
18 O.D.	16 1/4	15	1 5/8	1 1/8	13	12	7/8
20 O.D.	17 7/8	15	1 5/8	1 1/4	13	12	7/8
24 O.D.	20 3/4	17 1/2	1 7/8	1 1/4	15 1/4	16	1
For 600 lb. per sq. in. at 750° F.							
2	4 3/4	6 1/8	13/16	5/8	4 1/2	4	3/4
2 1/2	5 1/4	6 1/8	13/16	5/8	4 1/2	4	3/4
3	5 3/4	6 1/2	7/8	3/4	5	8	5/8
3 1/2	6 1/2	6 1/2	7/8	3/4	5	8	5/8
4	7	7 1/2	1	3/4	5 7/8	8	3/4
5	8 1/4	10	1 1/4	3/4	7 7/8	8	3/4
6	9	10	1 1/4	3/4	7 7/8	8	3/4
8	11	12 1/2	1 7/16	1	10 5/8	12	3/4
10	12 1/2	12 1/2	1 7/16	1	10 5/8	12	3/4
12	13 1/4	15	1 5/8	1 1/8	13	12	7/8
14 O.D.	14 3/4	15	1 5/8	1 1/8	13	12	7/8
16 O.D.	16	15	1 5/8	1 1/4	13	12	7/8

(*Table continued on following page*)

Table 47.—Dimensions of Steel Flanged Base Fittings—*Continued*

Nominal Pipe Size	Center to Base R	Diameter of Base †	Thickness of Base T	Thickness of Ribs U	Drilling for Base: Diameter of Bolt Circle	Drilling for Base: Number of Bolts	Drilling for Base: Size of Bolts
For 900 lb. per sq. in. at 750° F.							
3	5 3/4	6 1/2	7/8	3/4	5	8	5/8
4	7	7 1/2	1	3/4	5 7/8	8	3/4
5	8 1/4	10	1 1/4	3/4	7 7/8	8	3/4
6	9	10	1 1/4	3/4	7 7/8	8	3/4
8	11	12 1/2	1 7/16	1	10 5/8	12	3/4
10	12 1/2	12 1/2	1 7/16	1	10 5/8	12	3/4
12	13 1/4	15	1 5/8	1 1/8	13	12	7/8
14 O.D.	14 3/4	15	1 5/8	1 1/8	13	12	7/8
16 O.D.	16	15	1 5/8	1 1/4	13	12	7/8
For 1500 lb. per sq. in. at 750° F.							
2	5 1/2	6 1/2	7/8	3/4	5	4	5/8
2 1/2	6	6 1/2	7/8	3/4	5	4	5/8
3	6 1/2	7 1/2	1	3/4	5 7/8	4	3/4
4	7 3/4	10	1 1/4	3/4	7 7/8	4	3/4
5	9	10	1 1/4	3/4	7 7/8	4	3/4
6	9 3/4	12 1/2	1 7/16	1	10 5/8	4	3/4
8	11 1/2	12 1/2	1 7/16	1	10 5/8	4	3/4
10	13 3/4	15	1 5/8	1 1/8	13	4	7/8
12	15 1/2	15	1 5/8	1 1/8	13	4	7/8
14	17 1/4	17 1/2	1 7/8	1 1/4	15 1/4	4	1
16	18 3/4	17 1/2	1 7/8	1 1/4	15 1/4	4	1

* Dimensions for these sizes apply only to 300-lb. fittings. For 400-lb. fittings under 4 in. use dimensions of 600-lb. fittings. † Bases are to be plain faced unless otherwise specified, and center to base dimension R is the finished dimension.

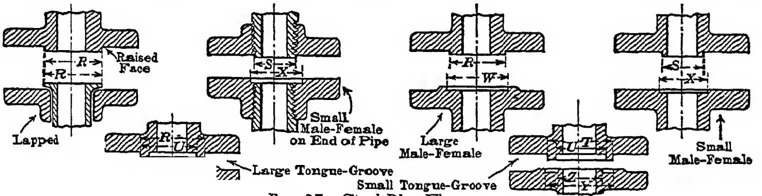


FIG. 37. Steel Pipe Flanges

Table 48.—Facing Dimensions for 150-, 300-, 400-, 600-, 900-, 1500-, and 2500-lb. Steel Flanges (A.S.A. B16e, 1939)

All dimensions in inches. See Fig. 37.

Nominal Pipe Size	Outside Diameter: Raised Face Lapped Large Male, and Large Tongue R	Outside Diameter: Small Male S	Outside Diameter: Small Tongue T	Inside Diameter of Large and Small Tongue U	Outside Diameter: Large Female* and Large Groove* W	Outside Diameter: Small Female* X	Outside Diameter: Small Groove* Y	Inside Diameter of Large and Small Groove* Z
1/2	1 3/8	23/32	1 3/8	1	1 7/16	25/32	1 7/16	15/16
3/4	1 11/16	15/16	1 11/16	1 5/16	1 3/4	1	1 3/4	1 1/4
1	2	1 3/16	1 7/8	1 1/2	2 1/16	1 1/4	1 15/16	1 7/16
1 1/4	2 1/2	1 1/2	2 1/4	1 7/8	2 9/16	1 9/16	2 5/16	1 13/16
1 1/2	2 7/8	1 3/4	2 1/2	2 1/8	2 15/16	1 13/16	2 9/16	2 1/16
2	3 5/8	2 1/4	3 1/4	2 7/8	3 11/16	2 5/16	3 5/16	2 13/16
2 1/2	4 1/8	2 11/16	3 3/4	3 3/8	4 3/16	2 3/4	3 13/16	3 5/16
3	5	3 5/16	4 5/8	4 1/4	5 1/16	3 3/8	4 11/16	4 3/16
3 1/2	5 1/2	3 13/16	5 1/8	4 3/4	5 9/16	3 7/8	5 3/16	4 11/16
4	6 3/16	4 5/16	5 11/16	5 3/16	6 1/4	4 3/8	5 3/4	5 1/8
5	7 5/16	5 3/8	6 13/16	6 5/16	7 3/8	5 7/16	6 7/8	6 1/4
6	8 1/2	6 3/8	8	7 1/2	8 9/16	6 7/16	8 1/16	7 7/16
8	10 5/8	8 3/8	10	9 3/8	10 11/16	8 7/16	10 1/16	9 5/16
10	12 3/4	10 1/2	12	11 1/4	12 13/16	10 9/16	12 1/16	11 3/16
12	15	12 1/2	14 1/4	13 1/2	15 1/16	12 9/16	14 5/16	13 7/16
14 O.D.	16 1/4	13 3/4	15 1/2	14 3/4	16 5/16	13 13/16	15 9/16	14 11/16
16 O.D.	18 1/2	15 3/4	17 5/8	16 3/4	18 9/16	15 13/16	17 11/16	16 11/16
18 O.D.	21	17 3/4	20 1/8	19 1/4	21 1/16	17 13/16	20 3/16	19 3/16
20 O.D.	23	19 3/4	22	21	23 1/16	19 13/16	22 1/16	20 15/16
24 O.D.	27 1/4	23 3/4	26 1/4	25 1/4	27 5/16	23 13/16	26 5/16	25 3/16

* Depth of groove or female, 3/16 in., for all sizes. Height of raised face, 150- and 300-lb. standards, 1/16 in. Height of raised face, 400-, 600-, 900-, 1500- and 2500-lb. standards, 1/4 in. A tolerance of ±0.016 inch (1/64 inch) is allowed on the inside and outside diameters of all facings.

Table 49.—Facing Dimensions for Ring-joint Flanges*

A.S.A., B16e, 1939 and A.P.I., 5G3, 1937 Standards

For dimensions not given see Table 45. All dimensions in inches. See Fig. 38.

Nominal Pipe Size	Pitch Diam., Ring and Groove *P*	Width of Groove *D*	Depth of Groove *L* †	Diam. of Raised Face, Ring or Lapped Joint *K*	Ring No. See Table 50	Approx. Distance between Flanges ‡	Pitch Diam., Ring and Groove *P*	Width of Groove *D*	Depth of Groove *L* †	Diam. of Raised Face, Ring or Lapped Joint *K*	Ring No. See Table 50	Approx. Distance between Flanges ‡
	For 150 lb. per sq. in.						**For 300 lb. per sq. in.**					
1/2	....	....	......	....	....	....	1 11/32	9/32	3/16	2	R 11	1/8
3/4	....	....	......	....	....	....	1 11/16	11/32	7/32	2 1/2	R 13	5/32
1	1 7/8	11/32	7/32	2 1/2	R 15	5/32	2	11/32	7/32	2 3/4	R 16	5/32
1 1/4	2 1/4	11/32	7/32	2 7/8	R 17	5/32	2 3/8	11/32	7/32	3 1/8	R 18	5/32
1 1/2	2 9/16	11/32	7/32	3 1/4	R 19	5/32	2 11/16	11/32	7/32	3 9/16	R 20	5/32
2	3 1/4	11/32	1/4	4	R 22	5/32	3 1/4	15/32	5/16	4 1/4	R 23	7/32
2 1/2	4	11/32	1/4	4 3/4	R 25	5/32	4	15/32	5/16	5	R 26	7/32
3	4 1/2	11/32	1/4	5 1/4	R 29	5/32	4 7/8	15/32	5/16	5 3/4	R 31 §	7/32
3 1/2	5 3/16	11/32	1/4	6 1/16	R 33	5/32	5 3/16	15/32	5/16	6 1/4	R 34	7/32
4	5 7/8	11/32	1/4	6 3/4	R 36	5/32	5 7/8	15/32	5/16	6 7/8	R 37	7/32
5	6 3/4	11/32	1/4	7 5/8	R 40	5/32	7 1/8	15/32	5/16	8 1/4	R 41	7/32
6	7 5/8	11/32	1/4	8 5/8	R 43	5/32	8 5/16	15/32	5/16	9 1/2	R 45	7/32
8	9 3/4	11/32	1/4	10 3/4	R 48	5/32	10 5/8	15/32	5/16	11 7/8	R 49	7/32
10	12	11/32	1/4	13	R 52	5/32	12 3/4	15/32	5/16	14	R 53	7/32
12	15	11/32	1/4	16	R 56	5/32	15	15/32	5/16	16 1/4	R 57	7/32
14 O.D.	15 5/8	11/32	1/4	16 3/4	R 59	1/8	16 1/2	15/32	5/16	18	R 61	7/32
16 O.D.	17 7/8	11/32	1/4	19	R 64	1/8	18 1/2	15/32	5/16	20	R 65	7/32
18 O.D.	20 3/8	11/32	1/4	21 1/2	R 68	1/8	21	15/32	5/16	22 5/8	R 69	7/32
20 O.D.	22	11/32	1/4	23 1/2	R 72	1/8	23	17/32	3/8	25	R 73	7/32
24 O.D.	26 1/2	11/32	1/4	28	R 76	1/8	27 1/4	21/32	7/16	29 1/2	R 77	1/4
	For 400 lb. per sq. in.						**For 600 lb. per sq. in.**					
1/2	1 11/32	9/32	3/16	2	R 11	1/8	1 11/32	9/32	3/16	2	R 11	1/8
3/4	1 11/16	11/32	7/32	2 1/2	R 13	5/32	1 11/16	11/32	7/32	2 1/2	R 13	5/32
1	2	11/32	7/32	2 3/4	R 16	5/32	2	11/32	7/32	2 3/4	R 16	5/32
1 1/4	2 3/8	11/32	7/32	3 1/8	R 18	5/32	2 3/8	11/32	7/32	3 1/8	R 18	5/32
1 1/2	2 11/16	11/32	7/32	3 9/16	R 20	5/32	2 11/16	11/32	7/32	3 9/16	R 20	5/32
2	3 1/4	15/32	5/16	4 1/4	R 23	3/16	3 1/4	15/32	5/16	4 1/4	R 23	3/16
2 1/2	4	15/32	5/16	5	R 26	3/16	4	15/32	5/16	5	R 26	3/16
3	4 7/8 §	15/32	5/16	5 3/4	R 31 §	3/16	4 7/8	15/32	5/16	5 3/4	R 31 §	3/16
3 1/2	5 3/16	15/32	5/16	6 1/4	R 34	3/16	5 3/16	15/32	5/16	6 1/4	R 34	3/16
4	5 7/8	15/32	5/16	6 7/8	R 37	7/32	5 7/8	15/32	5/16	6 7/8	R 37	3/16
5	7 1/8	15/32	5/16	8 1/4	R 41	7/32	7 1/8	15/32	5/16	8 1/4	R 41	3/16
6	8 5/16	15/32	5/16	9 1/2	R 45	7/32	8 5/16	15/32	5/16	9 1/2	R 45	3/16
8	10 5/8	15/32	5/16	11 7/8	R 49	7/32	10 5/8	15/32	5/16	11 7/8	R 49	3/16
10	12 3/4	15/32	5/16	14	R 53	7/32	12 3/4	15/32	5/16	14	R 53	3/16
12	15	15/32	5/16	16 1/4	R 57	7/32	15	15/32	5/16	16 1/4	R 57	3/16
14 O.D.	16 1/2	15/32	5/16	18	R 61	7/32	16 1/2	15/32	5/16	18	R 61	3/16
16 O.D.	18 1/2	15/32	5/16	20	R 65	7/32	18 1/2	15/32	5/16	20	R 65	3/16
18 O.D.	21	15/32	5/16	22 5/8	R 69	7/32	21	15/32	5/16	22 5/8	R 69	3/16
20 O.D.	23	17/32	3/8	25	R 73	7/32	23	17/32	3/8	25	R 73	3/16
24 O.D.	27 1/4	21/32	7/16	29 1/2	R 77	1/4	27 1/4	21/32	7/16	29 1/2	R 77	7/32
	For 900 lb. per sq. in.						**For 1500 lb. per sq. in.**					
1/2	1 9/16	11/32	7/32	2 3/8	R 12	5/32	1 9/16	11/32	7/32	2 3/8	R 12	5/32
3/4	1 3/4	11/32	7/32	2 5/8	R 14	5/32	1 3/4	11/32	7/32	2 5/8	R 14	5/32
1	2	11/32	7/32	2 13/16	R 16	5/32	2	11/32	7/32	2 13/16	R 16	5/32
1 1/4	2 3/8	11/32	7/32	3 3/16	R 18	5/32	2 3/8	11/32	7/32	3 3/16	R 18	5/32
1 1/2	2 11/16	11/32	7/32	3 5/8	R 20	5/32	2 11/16	11/32	7/32	3 5/8	R 20	5/32
2	3 3/4	15/32	5/16	4 7/8	R 24	1/8	3 3/4	15/32	5/16	4 7/8	R 24	1/8
2 1/2	4 1/4	15/32	5/16	5 3/8	R 27	1/8	4 1/4	15/32	5/16	5 3/8	R 27	1/8
3	4 7/8	15/32	5/16	6 1/8	R 31	5/32	5 3/8	15/32	5/16	6 5/8	R 35	1/8
4	5 7/8	15/32	5/16	7 1/8	R 37	5/32	6 3/8	15/32	5/16	7 5/8	R 39	1/8
5	7 1/8	15/32	5/16	8 1/2	R 41	5/32	7 5/8	15/32	5/16	9	R 44	1/8
6	8 5/16	15/32	5/16	9 1/2	R 45	5/32	8 5/16	17/32	3/8	9 3/4	R 46	1/8
8	10 5/8	15/32	5/16	12 1/8	R 49	5/32	10 5/8	21/32	7/16	12 1/2	R 50	5/32
10	12 3/4	15/32	5/16	14 1/4	R 53	5/32	12 3/4	21/32	7/16	14 5/8	R 54	5/32
12	15	15/32	5/16	16 1/2	R 57	5/32	15	29/32	9/16	17 1/4	R 58	3/16
14 O.D.	16 1/2	21/32	7/16	18 3/8	R 62	5/32	16 1/2	1 1/16	5/8	19 1/4	R 63	7/32
16 O.D.	18 1/2	21/32	7/16	20 5/8	R 66	5/32	18 1/2	1 3/16	11/16	21 1/2	R 67	5/16
18 O.D.	21	25/32	1/2	23 3/8	R 70	3/16	21	1 3/16	11/16	24 1/8	R 71	5/16
20 O.D.	23	25/32	1/2	25 1/2	R 74	3/16	23	1 5/16	11/16	26 1/2	R 75	3/8
24 O.D.	27 1/4	1 1/16	5/8	30 3/8	R 78	7/32	27 1/4	1 7/16	13/16	31 1/4	R 79	7/16

*See footnotes at end of table.

(*Table continued on following page*)

Table 49.—Facing Dimensions for Ring-joint Flanges—*Continued*

Nominal Pipe Size	Pitch Diam., Ring and Groove	Width of Groove	Depth of Groove L †	Diam. of Raised Face, Ring or Lapped Joint	Ring No. See Table 50	Approx. Distance between Flanges ‡
			For 2500 lb. per sq. in.			
	1 11/16		7/32	2 9/16	R 13	
	2		7/32	2 7/8	R 16	/32
	2 3/		7/32	3 1/	R 18	5/32
	2 27/32			4	R 21	1/8
	3 1/4			4 1/2	R 23	1/8
	4			5 1/4	R 26	1/8
2 1/2	4 3/			5	R 28	1/8
3			3/8		R 32	1/8
4			7/16		R 38	5/32
5			1/2	9	R 42	5/32
6			1/2	11	R 47	5/32
8	11		9/	13 3/8	R 51	3/16
10	13 1/2			16 3/4	R 55	1/4
12	16			19 1/2	R 60	5/16

† Depth of groove is added to minimum flange thickness Q. See Table 45.
‡ For ring joints when joint is compressed. This dimension must be added in calculating "laying lengths."
§ For ring joints with lapped flanges make P = 4 5/8 in. and use ring R30.

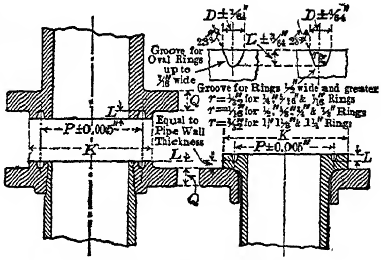


FIG. 38. Ring-joint Flange

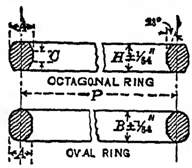


FIG. 39. Rings for Ring Joints

Table 50.—Dimensions of Rings for Ring Pipe Joints

A.S.A., B16e, 1939 and A.P.I., 5G3, 1937 Standards

P = pitch diameter of ring; A = width of ring; B = height of oval ring; H = height of octagonal ring; C = height of flat on octagonal ring. See Fig. 39. All dimensions in inches.

No.				H	C	No.				H	C	No.	P				
R 11	..	1/4	/16			R 34	5 3/16	7/16	/16			R 57	5	7/16			
R 12	9/16	/16	/16			R 35	5 3/8	7/16	11/16			R 58	5	7/8	1/8	1/16	
R 13	11/1	5/16	/16			R 36	5 7/8	5/16	9/16			R 59	5 5/8	5/16	9/16		
R 14	3/4	5/16	9/16			R 37	5 7/8	7/16	11/16			R 60	6	1/4	1 9/16	1/2	5/8
R 15	7/8	5/16	9/16			R 38	6 3/16	5/8	7/8	13/16	5/16	R 61	6 1/2	7/16	11/16		
R 16	2	5/16	9/16			R 39	6 3/8	7/16	11/16			R 62	6 1/2	5/8	7/8	13/16	/16
R 17	2 1/4	5/16	9/16			R 40	6 3/4	5/16	9/16			R 63	6 1/2		1 5/16	1/4	1/2
R 18	2 3/8	5/16	9/16			R 41	7 1/8	7/16	11/16			R 64	7 7/8	5/16	9/16		
R 19	2 9/16	5/16	9/16			R 42	7 1/2	3/4	1	15/16	/16	R 65	8 1/2	7/16	11/16		
R 20	2 11/16	5/16	9/16			R 43	7 5/8	5/16	9/16			R 66	8 1/2	5/8	7/8	13/16	/16
R 21		7/16	11/1			R 44	7 5/8	7/16	11/16			R 67	8 1/2	1 1/8	1 7/16		9/16
R 22	3 1/4	5/16	9/16			R 45	8 5/16	7/16	11/16			R 68	20 3/8	5/16	9/16		
R 23	3 1/4	7/16	11/16			R 46		1/2	3/4	11/16		R 69		7/16	11/16		
R 24	3 3/4	7/16	11/16			R 47		3/4	1	15/16		R 70		3/4		1	5/16
R 25	4	5/16	9/16			R 48	9 3/4	5/16	9/16			R 71	21	1/8	1 7/16	3/8	9/16
R 26	4	7/16	11/16			R 49	10 5/8	7/16	11/16			R 72	22	5/16	9/16		.....
R 27	4 1/4	7/1	11/16			R 50	10 5/8	5/8	7/8	13/1		R 73	23	1/2	3/4		5/16
R 28	4 3/8	1/2	3/4			R 51	11	7/8	1 1/8	1 1/		R 74	23		1	15/1	5/16
R 29	4 1/2	5/1	9/16			R 52	12	5/16	9/16			R 75	23				5/8
R 30	4 5/8		11/16			R 53	12 3/4	7/1	11/1			R 76	26 1/2		9/16		.....
R 31	4 7/8		11/16			R 54	12 3/4	5/8	7/8	13/1		R 77	27 1/		7/8		5/16
R 32			3/4			R 55	13 1/2	1 1/8	1 7/16	1 3/8		R 78	27 1/4		1 5/16	1/4	1/2
R 33			9/16			R 56		5/1				R 79	27 1/	1 3/8	1 3/4	1 5/8	11/16

Table 51.—Dimensions of Steel Welding Neck Flanges

American Standards Association, A.S.A., B16e, 1939

For dimensions not given see Table 45. All dimensions in inches. See Fig. 40.

For 150 lb. per sq. in., gage

Nominal Pipe Size	Flange: Outside Diam. O	Flange: Thickness, min. Q	Hub Diameter: Outside X	Hub Diameter: Beginning of Chamfer H	Length through Hub Y
1/2	3 1/2	7/16	1 3/16	0.84	1 7/8
3/4	3 7/8	1/2	1 1/2	1.05	2 1/16
1	4 1/4	9/16	1 15/16	1.32	2 3/16
1 1/4	4 5/8	5/8	2 5/16	1.66	2 1/4
1 1/2	5	11/16	2 9/16	1.90	2 7/16
2	6	3/4	3 1/16	2.38	2 1/2
2 1/2	7	7/8	3 9/16	2.88	2 3/4
3	7 1/2	15/16	4 1/4	3.50	2 3/4
3 1/2	8 1/2	15/16	4 13/16	4.00	2 13/16
4	9	15/16	5 5/16	4.50	3
5	10	15/16	6 7/16	5.56	3 1/2
6	11	1	7 9/16	6.63	3 1/2
8	13 1/2	1 1/8	9 11/16	8.63	4
10	16	1 3/16	12	10.75	4
12	19	1 1/4	14 3/8	12.75	4 1/2
14 O.D.	21	1 3/8	15 3/4	14.00	5
16 O.D.	23 1/2	1 7/16	18	16.00	5
18 O.D.	25	1 9/16	19 7/8	18.00	5 1/2
20 O.D.	27 1/2	1 11/16	22	20.00	5 11/16
24 O.D.	32	1 7/8	26 1/8	24.00	6

For 300 lb. per sq. in., gage

Nominal Pipe Size	Flange: Outside Diam. O	Flange: Thickness, min. Q	Hub Diameter: Outside X	Hub Diameter: Beginning of Chamfer H	Length through Hub Y
1/2	3 3/4	9/16	1 1/2	0.84	2 1/16
3/4	4 5/8	5/8	1 7/8	1.05	2 1/4
1	4 7/8	11/16	2 1/8	1.32	2 7/16
1 1/4	5 1/4	3/4	2 1/2	1.66	2 9/16
1 1/2	6 1/8	13/16	2 3/4	1.90	2 11/16
2	6 1/2	7/8	3 5/16	2.38	2 3/4
2 1/2	7 1/2	1	3 15/16	2.88	3
3	8 1/4	1 1/8	4 5/8	3.50	3 1/8
3 1/2	9	1 3/16	5 1/4	4.00	3 3/16
4	10	1 1/4	5 3/4	4.50	3 3/8
5	11	1 3/8	7	5.56	3 7/8
6	12 1/2	1 7/16	8 1/8	6.63	3 7/8
8	15	1 5/8	10 1/4	8.63	4 3/8
10	17 1/2	1 7/8	12 5/8	10.75	4 5/8
12	20 1/2	2	14 3/4	12.75	5 1/8
14 O.D.	23	2 1/8	16 3/4	14.00	5 5/8
16 O.D.	25 1/2	2 1/4	19	16.00	5 3/4
18 O.D.	28	2 3/8	21	18.00	6 1/4
20 O.D.	30 1/2	2 1/2	23 1/8	20.00	6 3/8
24 O.D.	36	2 3/4	27 5/8	24.00	6 5/8

For 400 lb. per sq. in., gage

Nominal Pipe Size	Flange: Outside Diam. O	Flange: Thickness, min. Q	Hub Diameter: Outside X	Hub Diameter: Beginning of Chamfer H	Length through Hub Y
1/2	3 3/4	9/16	1 1/2	0.84	2 1/16
3/4	4 5/8	5/8	1 7/8	1.05	2 1/4
1	4 7/8	11/16	2 1/8	1.32	2 7/16
1 1/4	5 1/4	13/16	2 1/2	1.66	2 5/8
1 1/2	6 1/8	7/8	2 3/4	1.90	2 3/4
2	6 1/2	1	3 5/16	2.38	2 7/8
2 1/2	7 1/2	1 1/8	3 15/16	2.88	3 1/8
3	8 1/4	1 1/4	4 5/8	3.50	3 1/4
3 1/2	9	1 3/8	5 1/4	4.00	3 3/8
4	10	1 3/8	5 3/4	4.50	3 1/2
5	11	1 1/2	7	5.56	4
6	12 1/2	1 5/8	8 1/8	6.63	4 1/16
8	15	1 7/8	10 1/4	8.63	4 5/8
10	17 1/2	2 1/8	12 5/8	10.75	4 7/8
12	20 1/2	2 1/4	14 3/4	12.75	5 3/8
14 O.D.	23	2 3/8	16 3/4	14.00	5 7/8
16 O.D.	25 1/2	2 1/2	19	16.00	6
18 O.D.	28	2 5/8	21	18.00	6 1/2
20 O.D.	30 1/2	2 3/4	23 1/8	20.00	6 5/8
24 O.D.	36	3	27 5/8	24.00	6 7/8

For 600 lb. per sq. in., gage

Nominal Pipe Size	Flange: Outside Diam. O	Flange: Thickness, min. Q	Hub Diameter: Outside X	Hub Diameter: Beginning of Chamfer H	Length through Hub Y
1/2	3 3/4	9/16	1 1/2	0.84	2 1/16
3/4	4 5/8	5/8	1 7/8	1.05	2 1/4
1	4 7/8	11/16	2 1/8	1.32	2 7/16
1 1/4	5 1/4	13/16	2 1/2	1.66	2 5/8
1 1/2	6 1/8	7/8	2 3/4	1.90	2 3/4
2	6 1/2	1	3 5/16	2.38	2 7/8
2 1/2	7 1/2	1 1/8	3 15/16	2.88	3 1/8
3	8 1/4	1 1/4	4 5/8	3.50	3 1/4
3 1/2	9	1 3/8	5 1/4	4.00	3 3/8
4	10 3/4	1 1/2	6	4.50	4
5	13	1 3/4	7 7/16	5.56	4 1/2
6	14	1 7/8	8 3/4	6.63	4 5/8
8	16 1/2	2 3/16	10 3/4	8.63	5 1/4
10	20	2 1/2	13 1/2	10.75	6
12	22	2 5/8	15 3/4	12.75	6 1/8
14 O.D.	23 3/4	2 3/4	17	14.00	6 1/2
16 O.D.	27	3	19 1/2	16.00	7
18 O.D.	29 1/4	3 1/4	21 1/2	18.00	7 1/4
20 O.D.	32	3 1/2	24	20.00	7 1/2
24 O.D.	37	4	28 1/4	24.00	8

For 900 lb. per sq. in., gage

Nominal Pipe Size	Flange: Outside Diam. O	Flange: Thickness, min. Q	Hub Diameter: Outside X	Hub Diameter: Beginning of Chamfer H	Length through Hub Y
1/2	4 3/4	7/8	1 1/2	0.84	2 3/8
3/4	5 1/8	1	1 3/4	1.05	2 3/4
1	5 7/8	1 1/8	2 1/16	1.32	2 7/8
1 1/4	6 1/4	1 1/8	2 1/2	1.66	2 7/8
1 1/2	7	1 1/4	2 3/4	1.90	3 1/4
2	8 1/2	1 1/2	4 1/8	2.38	4
2 1/2	9 5/8	1 5/8	4 7/8	2.88	4 1/8
3	9 1/2	1 1/2	5	3.50	4
4	11 1/2	1 3/4	6 1/4	4.50	4 1/2
5	13 3/4	2	7 1/2	5.56	5
6	15	2 3/16	9 1/4	6.63	5 1/2
8	18 1/2	2 1/2	11 3/4	8.63	6 3/8
10	21 1/2	2 3/4	14 1/2	10.75	7 1/4
12	24	3 1/8	16 1/2	12.75	7 7/8
14 O.D.	25 1/4	3 3/8	17 3/4	14.00	8 3/8
16 O.D.	27 3/4	3 1/2	20	16.00	8 1/2
18 O.D.	31	4	22 1/4	18.00	9
20 O.D.	33 3/4	4 1/4	24 1/2	20.00	9 3/4
24 O.D.	41	5 1/2	29 1/2	24.00	11 1/2

For 1500 lb. per sq. in., gage

Nominal Pipe Size	Flange: Outside Diam. O	Flange: Thickness, min. Q	Hub Diameter: Outside X	Hub Diameter: Beginning of Chamfer H	Length through Hub Y
1/2	4 3/4	7/8	1 1/2	0.84	2 3/8
3/4	5 1/8	1	1 3/4	1.05	2 3/4
1	5 7/8	1 1/8	2 1/16	1.32	2 7/8
1 1/4	6 1/4	1 1/8	2 1/2	1.66	2 7/8
1 1/2	7	1 1/4	2 3/4	1.90	3 1/4
2	8 1/2	1 1/2	4 1/8	2.38	4
2 1/2	9 5/8	1 5/8	4 7/8	2.88	4 1/8
3	10 1/2	1 7/8	5 1/4	3.50	4 5/8
4	12 1/4	2 1/8	6 3/8	4.50	4 7/8
5	14 3/4	2 7/8	7 3/4	5.56	6 1/8
6	15 1/2	3 1/4	9	6.63	6 3/4
8	19	3 5/8	11 1/2	8.63	8 3/8
10	23	4 1/4	14 1/2	10.75	10
12	26 1/2	4 7/8	17 3/4	12.75	11 1/8
14 O.D.	29 1/2	5 1/4	19 1/2	14.00	11 3/4
16 O.D.	32 1/2	5 3/4	21 3/4	16.00	12 1/4
18 O.D.	36	6 3/8	23 1/2	18.00	12 7/8
20 O.D.	38 3/4	7	25 1/4	20.00	14
24 O.D.	46	8	30	24.00	16

(Table continued on following page)

Table 51.—Dimensions of Steel Welding Neck Flanges—*Continued*

Nominal Pipe Size	Flange: Outside Diam. O	Flange: Thickness	Hub Diameter: Outside X	Hub Diameter: Beginning of Chamfer H	through Hub
		For 2500 lb. per sq. in., gage			
1/2	5 1/4	1 3/16	1 11/16	0.84	[illegible]
	5 1/2	1 1/4	2	1.05	3 1/8
	6 1/4	1 3/8	2 1/4	1.32	3 1/2
	7 1/4	1 1/2	2 7/8	1.66	3 3/4
	8	1 3/4	3 1/8	1.90	4 3/8
2	9 1/4	2	3 3/4	2.38	5
2 1/2	10 1/2	2 1/4	4 1/2	2.88	5 5/8
3	12	2 5/8	5 1/4	3.50	6 5/8
4	14	3	6 1/2	4.50	7 1/2
5	16 1/2	3 5/8	8	5.56	9
6	19	4 1/4	9 1/4	6.63	10 3/4
8	21 3/4	5	12	8.63	12 1/2
10	26 1/2	6 1/2	14 3/4	10.75	16 1/2
12	30	7 1/4	17 3/8	12.75	18 1/4

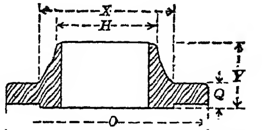


FIG. 40. Steel Welding Neck Flange

8. PIPING DETAILS

EXPANSION IN PIPING is provided for by the flexibility of the piping system as a whole, expansion bends, expansion joints, and by so introducing changes of direction that expansion will swivel the pipe in screwed fittings. Walker and Crocker (Piping Handbook) give the following formula for the expansion of piping:

$$L_t = L_0[\{1 + a(t - 32)/1000\} + b\{(t - 32)/1000\}^2],$$

where L_t, L_0 = length at temperature t and at 32° F., respectively; t = final temperature of pipe, deg. F.; a, b = constants whose values are:

	a	b
Cast Iron....	0.005441	0.001747
Steel.........	0.006212	0.001623
Wrought Iron.	0.006503	0.001622
Copper	0.009278	0.001244

The curves of Fig. 41 have been derived from this formula.

Stresses caused by expansion may exceed those due to the internal fluid pressure. In high-temperature work, creep stress also must be considered. Maximum permissible stress resulting from internal pressure, expansion, and dead weight between supports is 15,000 lb. per sq. in., up to temperatures of 700° F., with a reduction for higher temperatures. See p. 5–28.

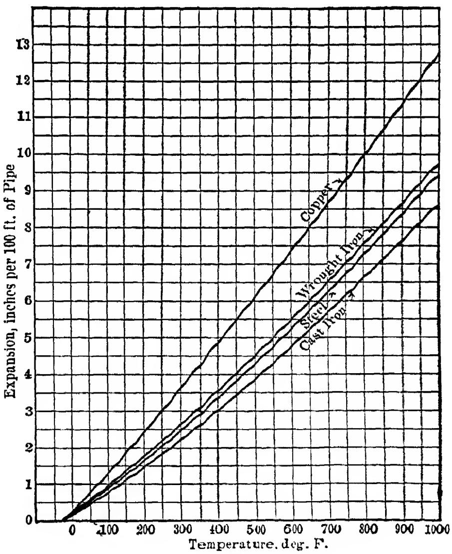


FIG. 41. Expansion of Piping

References.—Walker and Crocker, Piping Handbook, McGraw-Hill; Crocker and Sanford, Elasticity of Pipe Bends, *Trans.* A.S.M.E., xliv, 1922, p. 547; A. M. Wahl, Stresses and Reactions in Expansion Pipe Bends, *Trans.* A.S.M.E., FSP-50-49, 1928; W. H.

Shipman, Design of Steam Piping to Care for Expansion, *Trans.* A.S.M.E., FSP-51-52, 1929; C. T. Mitchell, Graphic Method for Determining Stresses in Pipe Lines, *Trans.* A.S.M.E., FSP-52-25, 1930; E. T. Cope and E. A. Wert, Load Deflection for Large, Plain, Corrugated and Creased Pipe Bends, *Trans.* A.S.M.E., FSP-54-12, 1932; S. W. Spielvogel and S. Kameros, *Trans.* A.S.M.E., FSP-57-10, 1935; H. M. Hovgaard, Elastic Deformation of Pipe Bends and Deformation of Plane Pipes, and Further Research on Pipe Bends, Publication of Mass. Inst. of Tech., vol. 62, No. 64 (Nov., 1926) and vol. 64, No. 59 (Dec., 1928); A. M. Wahl, J. W. Bowley and G. Back, Stresses in Turbine Pipe Bends, *Mech. Engg.*, Nov., 1929; J. H. Walker, Steam Piping for 1000 deg., *Power Plant Engg.*, June, 1931.

Expansion Joints.—Expansion joints may be either single- or double-slip expansion joints, Fig. 44, or corrugated copper joints, Figs. 42 and 43. The number of joints to be

Table 52.—Traverse of Single-slip Expansion Joints

Pipe size, in.	2	2 1/2	3	3 1/2	4	4 1/2
Traverse, in.	2 1/2	2 1/2	2 3/4	3	3 1/2	3 1/2

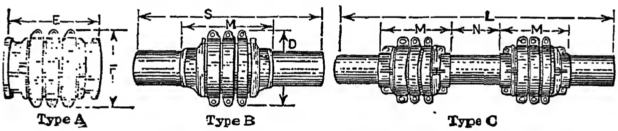


FIG. 42. Corrugated Copper Expansion Joint

Table 53.—Dimensions and Weights of Copper Expansion Joints

(E. B. Badger & Sons Co., Boston)

See Fig. 42. Dimensions in inches; weights in pounds

Traverse, in.		Pipe Size, in. 3	4	5	6	8	10	12	14	16	18	20
	E	......	11 1/4	11 1/4	12 1/2	12 1/2	12 1/2	13	13 1/2	13 1/2	14	15
	F	......	11	12	13 1/2	16 3/4	18 3/4	23	25	27	29	31
	D	10 1/2	11 1/2	11 1/4	14	17	19 1/4	23 1/2	25	27	29	31
	N	8	8	11	11 1/4	12 1/2	13 1/4	13 1/4	14 1/2	15 1/4	16 3/4	17
1	*M*	9	9	8 1/2	10 3/4	9 3/4	9 3/4	11 1/2	11 1/2	11 1/2	11 1/2	11 1/2
	S	26 1/2	26 1/2	26 3/4	29 1/2	29 1/2	29 1/2	31	31 1/2	33	33 1/2	34
	Wt. A	......	90	102	139	162	208	305	415	534	612	702
	" B	71	94	100	170	212	275	393	423	532	631	678
1 1/2	*M*	11 1/2	11 1/2	11 1/4	12 1/2	12	12	14 1/2	14 1/2	14 1/2	14 1/2	14 1/2
	S	29	29	29 1/2	31 1/2	31 1/2	31 1/2	34	34 1/2	36	36 1/2	37
	Wt. A	......	109	122	174	207	262	379	488	618	710	805
	" B	99	120	125	205	257	329	467	497	616	728	781
2	*L*	43 1/2	43 1/2	46	51 3/4	51 3/8	52 1/4	55 3/4	57 1/2	59 3/4	61 3/4	62 1/2
	M	14	14	14	16	14	14	17 1/2	17 1/2	17 1/2	17 1/2	17 1/2
	S	31 1/2	31 1/2	32 1/4	35	33 1/2	33 1/2	37	37 1/2	39	39 1/2	40
	Wt. A	......	132	145	236	251	316	445	562	702	807	908
	" B	123	143	150	267	301	383	533	570	700	826	884
	" C	154	193	190	327	407	542	765	822	1031	1231	1319
3	*L*	48 1/2	48 1/2	51 1/2	55 1/4	55 3/4	56 1/2	61 3/4	63 1/2	65 3/4	67 3/4	68 1/2
	M	19	19	18	19 1/2	16 3/4	16 3/4	21	21	21	21	21
	S	36 1/2	36 1/2	36 1/4	38 1/2	36 1/2	36 1/2	40 1/2	41	42 1/2	43	43 1/2
	Wt. A	......	179	191	299	296	371	509	635	786	906	1012
	" B	170	190	190	330	346	438	597	644	784	925	988
	" C	190	231	240	397	496	650	914	969	1199	1426	1525
4	*L*	53 1/2	53 1/2	57	62 1/4	60	60 3/4	67 3/4	69 1/2	71 3/4	73 3/4	74 1/2
	M	......	......	20 3/4	23 1/4	21	21	27	27	27	27	27
	S	......	......	39 1/4	42 1/2	40 1/2	40 1/2	46 1/2	47	48 1/2	49	49 1/2
	Wt. A	......	......	......	361	340	425	574	709	871	1005	1116
	" B	......	......	215	392	390	492	662	717	869	1024	1092
	" C	237	278	290	522	585	759	1045	1114	1368	1621	1731
6	*L*	63 1/2	63 1/2	65 1/4	69 1/4	65 1/4	66	74 3/4	76 1/2	78 3/4	80 3/4	81 1/2
	M	19*	19*	18	19 1/2	16 3/4	16 3/4	21	21	21	21	21
	Wt. C	331	371	370	647	674	868	1174	1263	1536	1819	1939
8	*L*	......	......	71 1/4	76 3/4	73 3/4	74 1/2	86 3/4	88 1/2	90 3/4	92 3/4	93 1/2
	M	......	......	20 3/4	23 1/4	21	21	27	27	27	27	27
	Wt. C	......	......	420	872	763	977	1303	1412	1705	2059	2147

* Type C only.

installed or the lineal feet of pipe for which each joint will compensate is determined by the traverse or movement of the joint. Table 52 gives the traverse of single-slip joints, and Table 53 the expansion and dimensions of corrugated copper expansion joints.

A corrugated copper expansion joint with standard flanges will withstand 125 lb. per sq. in. working pressure. With extra heavy flanges it will withstand a maximum of 250 lb. per sq. in. working pressure. A copper expansion joint never should be installed with superheat unless fitted with Monel metal sleeves. An expansion joint will not function properly unless the pipe line is securely anchored and has the required number of pipe guides. Anchors should be placed between every pair of joints and at every elbow.

Fig. 43. Corrugated Expansion Joint

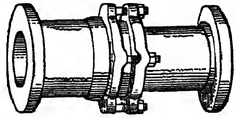


Fig. 44. Single Slip Expansion Joint

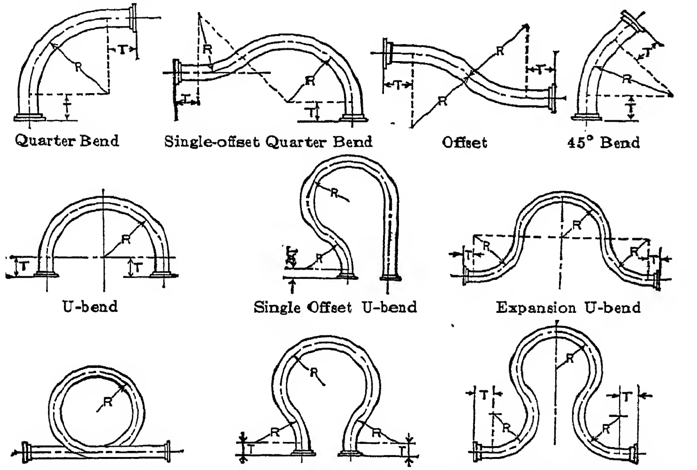


Fig. 45. Pipe Bends

Expansion Bends are of various forms as shown in Fig. 45 and Table 54. For extreme flexibility bends are corrugated or creased. The expansion taken up by expansion bends

Table 54. Dimensions of Pipe Bends

(See Fig. 45)

Size of Pipe, in.	Radius of Bend, in. R	Length of Tangent, in. T	Lineal Feet of Pipe in			Minimum Radius of Bends, Extra Strong Pipe, in.
			U-Bend	90° Bend	45° Bend	
2	12 1/2	3	3′ 9 1/4″	2′ 1 3/4″	1′ 3 7/8″	6
3	15	4	4′ 7 1/8″	2′ 7 3/4″	1′ 7 7/8″	12
3 1/2	17 1/2	5	5′ 5″	3′ 1 1/2″	1′ 11 3/4″	12
4	20	5	6′ 1″	3′ 5 1/2″	2′ 1 3/4″	14
5	25	6	7′ 6 5/8″	4′ 3 1/4″	2′ 7 1/2″	14
6	30	7	9′ 0 1/4″	5′ 1 1/8″	3′ 1 1/2″	15
8	40	9	11′ 11 3/4″	6′ 9″	4′ 1 1/2″	23
10	50	12	15′ 1 1/8″	8′ 6 1/2″	5′ 3 1/4″	30
12	60	14	18′ 0 1/4″	10′ 0 1/4″	6′ 3 1/4″	36
14	70	16	21′ 0″	11′ 10″	7′ 3″	48
16	96	18	.. ..	15′ 6 3/4″	9′ 3 1/2″	60
18	108	18	.. ..	17′ 1 3/4″	10′ 0 7/8″	66
20	120	18	.. ..	18′ 8 1/2″	10′ 10 1/4″	72
24	144	18	.. ..	21′ 10″	12′ 0 5/8″	108

Table 55.—Expansion Taken Up by Pipe Bends

(Crane Co., Chicago, 1929)

Figures in the table give the expansion, in inches, in a pipe line taken up by a quarter bend. To obtain the expansion taken up by other types of bends, multiply values in table by the following factors: U-bends, multiply by 2; single offset bends or expansion U-bends, multiply by 4; double offset bends, circle bends, or double offset expansion U-bends, multiply by 5.

Pipe Size, in.	Mean Radius of Bend, in.												
	12	15	20	30	40	50	60	70	80	90	100	110	120
1	1/4	3/8	3/4	1 3/4	3 1/8	.....	.....	.....	.....	.....	.....	.....	.....
2	1/8	1/4	1/2	1	1 3/4	2 3/4	3 7/8	5 3/8	.....	.....	.....	.....	.....
2 1/2	.....	1/4	3/8	7/8	1 1/2	2 1/4	3 1/4	4 1/2	5 3/4	.....	.....	.....	.....
3	.....	1/8	3/8	5/8	1 1/8	1 7/8	2 5/8	3 5/8	4 3/4	6	.....	.....	.....
3 1/2	.....	.....	1/4	5/8	1	1 5/8	2 3/8	3 1/8	4 1/8	5 1/4	.....	.....	.....
4	.....	.....	1/4	1/2	1	1 1/2	2	2 7/8	3 3/4	4 3/4	5 3/4	.....	.....
4 1/2	.....	.....	.....	1/2	7/8	1 3/8	1 7/8	2 1/2	3 3/8	4 1/4	5 1/4	.....	.....
5	.....	.....	.....	3/8	3/4	1 1/8	1 5/8	2 1/4	3	3 3/4	4 5/8	5 5/8	.....
6	.....	.....	.....	3/8	5/8	1	1 3/8	1 7/8	2 1/2	3 1/8	3 7/8	4 3/4	5 5/8
8	.....	.....	.....	.....	1/2	3/4	1	1 1/2	1 7/8	2 1/2	3	3 5/8	4 3/8
10	.....	.....	.....	.....	.....	5/8	7/8	1 1/8	1 1/2	2	2 3/8	2 7/8	3 1/2
12	.....	.....	.....	.....	.....	.....	3/4	1	1 3/8	1 5/8	2	2 1/2	2 7/8
14	.....	.....	.....	.....	.....	.....	.....	7/8	1 1/8	1 3/8	1 3/4	2 1/8	2 1/2
16	.....	.....	.....	.....	.....	.....	.....	.....	7/8	1 1/4	1 1/2	1 7/8	2 1/4
18	.....	.....	.....	.....	.....	.....	.....	.....	.....	.....	.....	1 5/8	1 7/8
20	.....	.....	.....	.....	.....	.....	.....	.....	.....	.....	.....	.....	1 3/4

is given in Table 55. These values may be large, and at high temperatures may introduce undesirable stresses in the pipe. They should be checked by a mathematical analysis of the pipe line as a whole before being accepted for design.

VAN STONE PIPE JOINT.—The original Van Stone joint was produced by expanding and turning over the end of the pipe, the resulting flange providing a bearing for a loose steel flange in which the bolt holes were drilled. The present form, as made by the M. W. Kellogg Co., New York is shown in Fig. 46. The upset end of the pipe is made of the same thickness as the pipe itself, but can be made as much thicker as desired, up to 200 percent of the pipe thickness. The square corner is formed in the upsetting process. The fillet radius on the back of the joint and on the face of the loose flange are concentric, thus bringing the bearing of the flange on the joint as close as possible to the pipe, and reducing the lever arm, tending to bend the joint, to the distance of the clearance between the pipe and flange. Flanges are made to the dimensions of the American Standard for flanged fittings. The radius of the fillet on the back of the joint is 3/8 in. for all sizes of pipe. The following clearances are allowed between pipe and flange: Pipe sizes 2 to 6 in. inclusive, 3/64 in.; 7 to 12 in. inclusive, 1/16 in.; 14 in. O.D. and larger, 5/64 in.

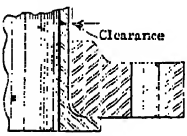


FIG. 46. Van Stone Joint

BENT AND COILED PIPE.—Table 56 shows minimum dimensions of pipe bends and coils. Machine-made bends of a single radius must be flat in one plane. Consecutive bends should have a straight portion at least equal to diameter of pipe between them. Coils requiring more than one length of tube or pipe cannot be made of as small diameter as those requiring no joints. The position or type of ends often limits the size of coils, and manufacturers should be consulted in unusual cases. The limiting factors for bends

Table 56.—Approximate Minimum Dimensions of Pipe Bends and Coils

National Pipe Bending Co., New Haven, Conn.

R = center radius, 90-deg. bends, in.; D = center diameter, 180-deg. or U-bends, in.; d = outside diameter of spiral coils, in.

Nominal Pipe Size, in.	Standard			Extra Strong			Nominal Pipe Size, in.	Standard			Extra Strong		
	R	D	d	R	D	d		R	D	d	R	D	d
1/8	7/8	1 3/4	2 1/2	3/4	1 1/2	2	2 1/2	6	12	20	4 1/2	9	15
1/4	1	2	3	7/8	1 3/4	2 1/2	3	8	16	30	6	12	20
3/8	1 1/8	2 1/4	4	1	2	3	3 1/2	14	28	....	9	18	....
1/2	1 1/4	2 1/2	5	1 1/8	2 1/4	4	4	18	36	....	12	24	....
3/4	1 1/2	3	6	1 1/4	2 1/2	5	4 1/2	20	40	....	15	30	....
1	2	4	8	1 1/2	3	6	5	24	48	....	18	36	....
1 1/4	2 1/2	5	10	2	4	8	6	30	60	....	21	42	....
1 1/2	3	6	12	2 1/2	5	10	8	40	80	....	30	60	....
2	4	8	15	3	6	12	.....	....	....	....	.....	....	....

Table 57.—Dimensions for Bends in Iron or Steel Pipes
Whitlock Coil Pipe Co., Hartford, Conn.

	Smallest A Radius, i		mallest Radius, E Bends, in.	th est	ortest Leng for Flangin	f Pipe, O in.	est Advisabl dius, in.*	evelop Bend Radi		Developed Length 90 Bend, Smallest Radius, Expansion Bend, in.	
3	12	19	18	28 1/2	3	14	60	94 1/2	112	177	7
3 1/2	13	20 1/2	21	33	3 1/2	16	70	110	128	202	7
4	15	23 1/2	24	38	3 1/2	18	80	126	144	227	7
4 1/2	17	27	27	42 1/2	4	20	90	142			8
5	20	31 1/2	30	47 1/2	4	22	100	157			8
6	23	36	36	57	4	24	110	173			8
7	26	41	48	75 1/2	5	26	120	189			10
8	30	47 1/2	54	85	5	28	140	220			10
9	36	57	60	94 1/2	5	30	160	251			10
10	42	66	72	114	6						
12	48	75 1/2	84	133	6						

* The radii given are to the center of the pipe. Radii can be reduced below those given in the table by the use of extra heavy pipe. Under usual circumstances steel pipe can be bent to a smaller radius than iron pipe.

in steel, brass or copper tubes are as follows, the outside diameter of the pipe being multiplied by the factor to obtain the desired dimension:

	Least Ordinary	Difficult	Limit
Center radius, 90-deg. bends...	2	1 1/2	1
Center diameter, U-bends.....	4	3	2
Center diameter, coils........	8	6	4
Outside diameter, coils.......	9	7	5

Steel tubing cannot be bent to the absolute limits of copper and brass. Thickness of wall of any tubing must be proportioned to the radius of curvature.

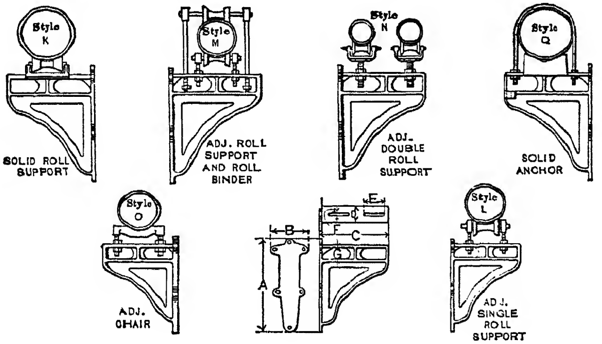
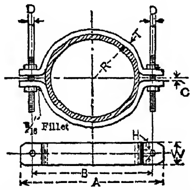


Fig. 47. Pipe Support Brackets

PIPE SUPPORTS.—Pipes may be supported on brackets or suspended from hangers. Figs. 47 to 51, and Tables 58 to 61 show designs and dimensions of brackets and hangers. Pipe on brackets should be carried on rollers formed to the pipe contour to guide its movement, due to expansion, in the desired direction. Long, heavy, suspended pipes, particularly those subjected to high temperature should be spring supported, to avoid the expansion of the pipe lifting the support from its bearing. Spring support is especially desirable for long vertical runs. A form of spring support is shown in Fig. 51.

Table 58.—Dimensions of Pipe Support Brackets
All dimensions in inches. See Fig. 47

Size No.	Safe Load Tons	Size of Pipe	Wall Plate Height *A*	Wall Plate Width *B*	Bracket Length *C*	Bracket Width *D*	Bracket Depth *G*	Slot Length *E*	Slot Width *F*
11	1	5 to 8	34	12	25	6	5 1/2	8 1/2	1 1/4
12	2	9 to 14	40	14	30	6	6	9	1 3/8
13	3	15 to 18	45	16	34	6	6 1/2	9 1/4	1 1/2
14	4	20 to 24	51 1/2	19	40	6	7	11 1/8	1 5/8
15	Special	20 to 30	64	19	44 1/8	6	7	12 1/4	1 5/8



. 48. Horizontal Pipe Clamp

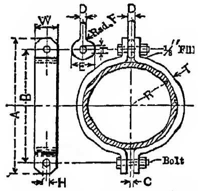


Fig. 49. Vertical Pipe Clamp

Table 59.—Dimensions of Horizontal Pipe Clamps
All dimensions in inches. See Fig. 48

Size of Pipe	Pipe Clamps: Overall *A*	Pipe Clamps: Spacing, Hanger Rods *B*	Pipe Clamps: Inside Radius, Clamp *R*	Fitting Clamps: Overall *A*	Fitting Clamps: Spacing, Hanger Rods *B*	Fitting Clamps: Inside Radius, Clamp *R*	All Clamps: Offset, Half Clamp *C*	All Clamps: Diam. Hanger Rod *D*	All Clamps: Diam. Hanger Rod Holes *H*	All Clamps: Thickness, Clamp *T*	All Clamps: Width of Clamp *W*
4	10 1/4	8	2 1/4	11 3/4	9 1/4	2 7/8	3/16	3/4	7/8	3/4	1 3/4
4 1/2	10 3/4	8 1/2	2 1/2	12 1/4	9 3/4	3 1/8	3/16	3/4	7/8	3/4	1 3/4
5	11 3/8	9 1/8	2 13/16	12 3/4	10 1/4	3 3/8	3/16	3/4	7/8	3/4	2
6	12 3/8	10 1/8	3 5/16	13 7/8	11 3/8	3 15/16	1/4	3/4	7/8	3/4	2
7	13 3/8	11 1/8	3 13/16	14 7/8	12 3/8	4 7/16	1/4	3/4	7/8	3/4	2
8	14 3/8	12 1/8	4 5/16	16	13 1/2	5	1/4	3/4	7/8	3/4	2
9	15 3/8	13 1/8	4 13/16	17	14 1/2	5 1/2	1/4	3/4	7/8	3/4	2 1/4
10	16 1/2	14 1/4	5 3/8	18 1/8	15 5/8	6 1/16	1/4	3/4	7/8	3/4	2 1/4
12	19 1/2	17	6 3/8	21 1/2	18 3/4	7 1/4	5/16	7/8	1	1	2 1/2
14 / 15 O.D.	21 3/4	19 1/4	7 1/2	23 3/4	21	8 3/8	5/16	7/8	1	1	2 1/2
15 / 16 O.D.	22 3/4	20 1/4	8	24 7/8	22 1/8	8 15/16	5/16	7/8	1	1	2 1/2
16 / 17	23 3/4	21 1/4	8 1/2	26	23 1/4	9 1/2	5/16	7/8	1	1	2 1/2

Table 60.—Dimensions of Vertical Pipe Clamps
All dimensions in inches. See Fig. 49

Size of Pipe	Pipe Clamps: Overall *A*	Pipe Clamps: Spacing Bolt Holes *B*	Pipe Clamps: Inside Radius, Clamp *R*	Fitting Clamps: Overall *A*	Fitting Clamps: Spacing Bolt Holes *B*	Fitting Clamps: Inside Radius, Clamp *R*	All Clamps: Offset, Half Clamp *C*	All Clamps: Diam. Hanger Rod *D*	All Clamps: Diam. Hanger Rod Eye *E*	All Clamps: Diam. Bolt Holes *H*	All Clamps: Thickness, Clamp *T*	All Clamps: Width of Clamp *W*	All Clamps: Bolts Diam.	All Clamps: Bolts Length
4	9 7/8	7 3/8	2 1/4	11 1/8	8 5/8	2 7/8	1/2	3/4	2 1/2	1	3/8	1 3/4	7/8	3
4 1/2	10 3/8	7 7/8	2 1/2	11 5/8	9 1/8	3 1/8	1/2	3/4	2 1/2	1	3/8	1 3/4	7/8	3
5	11	8 1/2	2 13/16	12 1/8	9 5/8	3 3/8	1/2	3/4	2 1/2	1	3/8	2	7/8	3
6	12	9 1/2	3 5/16	13 1/4	10 3/4	3 15/16	1/2	3/4	2 1/2	1	3/8	2	7/8	3
7	13	10 1/2	3 13/16	14 1/4	11 3/4	4 7/16	1/2	3/4	2 1/2	1	3/8	2	7/8	3
8	14 5/8	11 7/8	4 5/16	16	13 1/4	5	9/16	7/8	2 7/8	1 1/8	7/16	2	1	3 1/2
9	15 5/8	12 7/8	4 13/16	17	14 1/4	5 1/2	9/16	7/8	2 7/8	1 1/8	7/16	2 1/4	1	3 1/2
10	16 3/4	14	5 3/8	18 1/8	15 3/8	6 1/16	9/16	7/8	2 7/8	1 1/8	7/16	2 1/4	1	3 1/2
12	19 3/8	16 3/8	6 3/8	21 1/8	18 1/8	7 1/4	5/8	1	3 1/4	1 1/4	1/2	2 1/2	1 1/8	4
14 / 15 O.D.	21 5/8	18 5/8	7 1/2	23 3/8	20 3/8	8 3/8	5/8	1	3 1/4	1 1/4	1/2	2 1/2	1 1/8	4
15 / 16 O.D.	22 5/8	19 5/8	8	24 1/2	21 1/2	8 15/16	5/8	1	3 1/4	1 1/4	1/2	2 1/2	1 1/8	4
16 / 17	23 5/8	20 5/8	8 1/2	25 5/8	22 5/8	9 1/2	5/8	1	3 1/4	1 1/4	1/2	2 1/2	1 1/8	4

Table 61.—Dimensions of I-Beam Clamps for Pipe Hangers

All dimensions in inches. See Fig. 50

Dimension	Loop Type, Size of Beam					Open Type, Size of Beam					
	3	4	5	6	7	8	9	10	12	15	18
A	2 3/4	3 1/8	3 1/2	3 3/4	4	......	......	......	......	......	......
D	3 1/8	4 1/8	5 1/8	6 1/8	7 1/8	......	......	......	......	......	......
E	......	......	......	......	......	5/16	5/16	5/16	3/8	7/16	1/2
					3/4-in. Rod *						
B	1 5/8	1 5/8	1 5/8	1 5/8	1 5/8	1 3/4	1 3/4	1 3/4	1 3/4	1 3/4	1 3/4
C	7/8	7/8	7/8	7/8	7/8	......	......	......	......	......	......
L	15/16	1 1/8	1 5/16	1 7/16	1 9/16	1 9/16	1 3/4	1 7/8	2 1/16	2 5/16	2 9/16
T	1/2	1/2	1/2	1/2	1/2	5/8	5/8	5/8	5/8	5/8	5/8
W	2	2	2	2	2	2 1/2	2 1/2	2 1/2	2 1/2	2 1/2	2 1/2
Bolt Diam.	7/8	7/8	7/8	7/8	7/8	7/8	7/8	7/8	7/8	7/8	7/8
Bolt Length	3 1/4	3 1/4	3 1/4	3 1/4	3 1/4	3 3/4	3 3/4	3 3/4	3 3/4	3 3/4	3 3/4
					7/8-in. Rod *						
B	1 7/8	1 7/8	1 7/8	1 7/8	1 7/8	2	2	2	2	2	2
C	1	1	1	1	1	......	......	......	......	......	......
L	7/8	1 1/16	1 1/4	1 3/8	1 1/2	1 1/2	1 11/16	1 13/16	2	2 1/4	2 1/2
T	5/8	5/8	5/8	5/8	5/8	3/4	3/4	3/4	3/4	3/4	3/4
W	2 1/4	2 1/4	2 1/4	2 1/4	2 1/4	2 1/2	2 1/2	2 1/2	2 1/2	2 1/2	2 1/2
Bolt Diam.	1	1	1	1	1	1	1	1	1	1	1
Bolt Length	3 3/4	3 3/4	3 3/4	3 3/4	3 3/4	4 1/4	4 1/4	4 1/4	4 1/4	4 1/4	4 1/4
					1-in. Rod *						
B	2 1/8	2 1/8	2 1/8	2 1/8	2 1/8	2 1/4	2 1/4	2 1/4	2 1/4	2 1/4	2 1/4
C	1 1/8	1 1/8	1 1/8	1 1/8	1 1/8	......	......	......	......	......	......
L	13/16	1	1 3/16	1 5/16	1 7/16	1 7/16	1 5/8	1 3/4	1 15/16	2 3/16	2 7/16
T	3/4	3/4	3/4	3/4	3/4	7/8	7/8	7/8	7/8	7/8	7/8
W	2 1/2	2 1/2	2 1/2	2 1/2	2 1/2	2 1/2	2 1/2	2 1/2	2 1/2	2 1/2	2 1/2
Bolt Diam.	1 1/8	1 1/8	1 1/8	1 1/8	1 1/8	1 1/8	1 1/8	1 1/8	1 1/8	1 1/8	1 1/8
Bolt Length	4 1/4	4 1/4	4 1/4	4 1/4	4 1/4	4 3/4	4 3/4	4 3/4	4 3/4	4 3/4	4 3/4

* The size of the rod is determined by the size of the pipe. See Tables 59 and 60.

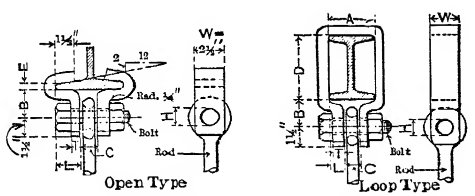


Fig. 50. I-Beam Clamps for Pipe Hanger

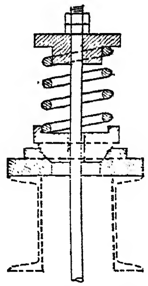


Fig. 51. Spring Support Pipe Hanger

9. STEAM VALVES

Valves for steam lines carrying pressures above 250 lb. per sq. in., but not over 1500 lb., and temperature above 450° F., but not above 750° F., should be of cast or forged steel. For pressures not over 250 lb. per sq. in. and temperatures not over 450° F., they may be of cast or forged steel, cast iron, malleable iron or non-ferrous metal.

Tables 62 to 69 give the dimensions and working pressures of representative valves suitable for steam and boiler feed lines. These tables are much abbreviated and represent but a small portion of the valves that are commercially available. The catalogs of valve manufacturers should be consulted for more detailed information.

Table 62.—Dimensions of Ferrosteel Body Globe, Angle and Cross Valves

(Crane Co., Chicago)

All dimensions in inches. See Fig. 52

Diam. of Port	End to End, Globe, Cross: Flanged* B	End to End, Globe, Cross: Screwed	Center to End, Angle, Cross: Flanged* B/2	Center to End, Angle, Cross: Screwed C/2	Center to Top, Open: Globe K	Center to Top, Open: Angle, Cross L	Diam. of Hand Wheel F	Flange Dimensions (raised face): Diam. D	Thick-	Diam. Raised Face	Diam. Bolt Circle	No. of Bolts	Diam. of Bolts
175-lb. Valves													
2	9	7 3/4	4 1/2	3 7/8	11 1/2	11 1/2	8	6 1/2	7/8	4 3/16	5	8	5/8
2 1/2	10	8	5	4	12 1/2	12 1/2	9	7 1/2	1	4 15/16	5 7/8	8	3/4
3	11	8 1/4	5 1/2	4 1/8	14 1/4	14 1/4	10	8 1/4	1 1/8	5 11/16	6 5/8	8	3/4
3 1/2	12	9 1/2	6	4 3/4	15 1/2	15 1/2	10	9	1 3/16	6 5/16	7 1/4	8	3/4
4	13	10 1/2	6 1/2	5 1/4	16 1/4	16 1/4	10	10	1 1/4	6 15/16	7 7/8	8	3/4
5	14 1/2	12 1/4	7 1/4	6 1/8	18 1/4	18 1/4	12	11	1 3/8	8 5/16	9 1/4	8	3/4
6	16	14	8	7	20 1/4	20 1/4	14	12 1/2	1 7/16	9 11/16	10 5/8	12	3/4
8	20	18 1/2	10	9 1/4	24 3/4	24 3/4	16	15	1 5/8	11 15/16	13	12	7/8
10	22 1/2	....	11 1/4	....	28 1/2	28 1/2	20	17 1/2	1 7/8	14 1/16	15 1/4	16	1
12	25 1/2	....	12 3/4	....	31 1/4	31 1/4	20	20 1/2	2	16 7/16	17 3/4	16	1 1/8
250-lb. Valves													
2	10 1/2	9 1/2	5 1/4	4 3/4	13 3/4	13	9	6 1/2	7/8	4 3/16	5	8	5/8
2 1/2	11 1/2	10 3/4	5 3/4	5 3/8	14 3/4	13 3/4	10	7 1/2	1	4 15/16	5 7/8	8	3/4
3	12 1/2	11 3/4	6 1/4	5 7/8	16 1/4	15 1/4	10	8 1/4	1 1/8	5 11/16	6 5/8	8	3/4
3 1/2	13 1/4	12 1/4	6 5/8	6 1/8	17	17 1/4	10	9	1 3/16	6 5/16	7 1/4	8	3/4
4	14	13	7	6 1/2	18 3/4	17 1/2	12	10	1 1/4	6 15/16	7 7/8	8	3/4
5	15 3/4	15	7 7/8	7 1/2	21	19 1/2	14	11	1 3/8	8 5/16	9 1/4	8	3/4
6	17 1/2	16 1/2	8 3/4	8 1/4	23 1/2	22	16	12 1/2	1 7/16	9 11/16	10 5/8	12	3/4
8	21	....	10 1/2	....	28 3/4	26 1/4	20	15	1 5/8	11 15/16	13	12	7/8
10	24 1/2	....	12 1/4	....	33	29 3/4	24	17 1/2	1 7/8	14 1/16	15 1/4	16	1

* Includes raised face.

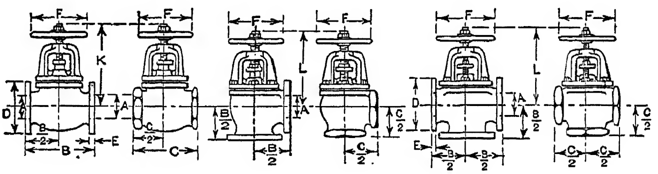


Fig. 52. Globe, Angle and Cross Valves

Table 63.—Dimensions of Iron Body Globe, Angle and Cross Valves

(Crane Co., Chicago)

All dimensions in inches. See Fig. 52

Diam. of Port A	End to End, Globe, Cross: Flanged B	End to End, Globe, Cross: Screwed C	Center to End, Angle, Cross: Flanged B/2	Center to End, Angle, Cross: Screwed C/2	Center to Top, Open: Globe K	Center to Top, Open: Angle, Cross L	Diam. of Hand Wheel F	Flange Dimensions (plain face): Diam. D	Thickness E	Diam. Bolt Circle	No. of Bolts	Diam. of Bolts
2	8	6 1/2	4	3 1/4	11 1/4	11 1/4	8	6	5/8	4 3/4	4	5/8
2 1/2	8 1/2	7	4 1/4	3 1/2	11 3/4	11 3/4	8	7	11/16	5 1/2	4	5/8
3	9 1/2	8	4 3/4	4	13 1/4	13	9	7 1/2	3/4	6	4	5/8
3 1/2	10 1/2	9	5 1/4	4 1/2	13 1/2	13 1/2	9	8 1/2	13/16	7	8	5/8
4	11 1/2	10	5 3/4	5	15 1/2	15	10	9	15/16	7 1/2	8	5/8
5	13	11 1/4	6 1/2	5 5/8	17 1/4	17	10	10	15/16	8 1/2	8	3/4
6	14	13	7	6 1/2	19 1/2	19 1/4	12	11	1	9 1/2	8	3/4
8	19 1/2	18 1/2	9 3/4	9 1/4	25 1/4	22 1/2	16	13 1/2	1 1/8	11 3/4	8	3/4
10	24 1/2	....	12 1/4	....	30 1/2	26 1/2	18	16	1 3/16	14 1/4	12	7/8
12	27 1/2	....	13 3/4	....	33 1/2	30	20	19	1 1/4	17	12	7/8
14	31	....	15 1/2	....	38 1/2	33 1/2	24	21	1 3/8	18 3/4	12	1
16	36	....	18	....	42 3/4	37 3/4	27	23 1/2	1 7/16	21 1/4	16	1

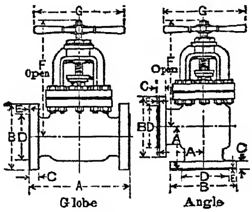


Fig. 53.

Table 64.—Dimensions of Cast-steel Flanged Globe and Angle Valves

(Consolidated Ashcroft Hancock Co., Bridgeport, Conn.)

All dimensions in inches. See Fig. 53

	Globe		Angle		Flanges					Bolts		
	Face to Face	Center to Top, Open F	Center to Face H	Center to Top, Open F	Diam. B	Thick-	Raised Face, Diam. D	Raised Face, Height E	Diam. Bolt Circle	No.	Diam.	Hand Wheel Diam. G
250 lb. Standard Steam, 300 lb., 750° F.; boiler feed, 325 lb., 450° F.; water (non-shock) 500 lb., 100° F.												
1/2 *	6 1/8	5 7/8	3 1/16	5 1/4	3 3/4	9/16	1 3/8	1/16	2 5/8	4	1/2	3 3/4
3/4 *	7 1/8	6 1/2	3 9/16	5 7/8	4 5/8	5/8	1 11/16	1/16	3 1/4	4	5/8	4 3/8
1 *	8 1/8	7 3/4	4 1/16	7	4 7/8	11/16	2	1/16	3 1/2	4	5/8	5 1/4
1 1/4 *	8 5/8	8 7/8	4 5/16	7 1/2	5 1/4	13/16	2 1/2	1/16	3 7/8	4	5/8	6 1/4
1 1/2 *	9 1/8	10	4 9/16	8 3/8	6 1/8	7/8	2 7/8	1/16	4 1/2	4	3/4	7 1/8
2 †	10	11 1/2	5	11 1/2	6 1/2	7/8	3 5/8	1/16	5	8	5/8	8
2 1/2 †	11	13 1/2	5 1/2	13 1/4	7 1/2	1	4 1/8	1/16	5 7/8	8	3/4	9 1/4
3 †	12	15	6	15	8 1/4	1 1/8	5	1/16	6 5/8	8	3/4	10 3/4
3 1/2 †	13	18	6 1/2	18 1/2	9	1 3/16	5 1/2	1/16	7 1/4	8	3/4	12
4 †	14	20	7	20 1/2	10	1 1/4	6 3/16	1/16	7 7/8	8	3/4	12 1/2
5 †	16	23	8	24	11	1 3/8	7 5/16	1/16	9 1/4	8	3/4	17
6 †	17	26 1/2	8 1/2	27	12 1/2	1 7/16	8 1/2	1/16	10 5/8	12	7/8	21
400 lb. Standard Steam, 450 lb., 750° F.; boiler feed, 500 lb., 450° F.; water (non-shock) 750 lb., 100° F.												
1/2 †	6 1/2	6 3/8	3 1/4	7	3 3/4	9/16	1 3/8	1/4	2 5/8	4	1/2	4 3/8
3/4 †	7 1/2	8	3 3/4	8 1/2	4 5/8	5/8	1 11/16	1/4	3 1/4	4	5/8	5 1/4
1 †	8 1/2	9	4 1/4	9 3/4	4 7/8	11/16	2	1/4	3 1/2	4	5/8	6 1/4
1 1/4 †	9	10 1/4	4 1/2	10 1/2	5 1/4	13/16	2 1/2	1/4	3 7/8	4	5/8	7 1/8
1 1/2 †	9 1/2	11 1/4	4 3/4	10 3/4	6 1/8	7/8	2 7/8	1/4	4 1/2	4	3/4	8
2 †	11 1/2	14 3/4	5 3/4	14 3/4	6 1/2	1	3 5/8	1/4	5	8	5/8	9 1/4
2 1/2 †	13	17 1/4	6 1/2	17 1/4	7 1/2	1 1/8	4 1/8	1/4	5 7/8	8	3/4	10 3/4
3 †	14	20	7	20	8 1/4	1 1/4	5	1/4	6 5/8	8	3/4	12 1/2
3 1/2 †	15	22	7 1/2	22	9	1 3/8	5 1/2	1/4	7 1/4	8	7/8	12 1/2
4 †	16	25	8	25	10	1 3/8	6 3/16	1/4	7 7/8	8	7/8	14 3/4
5 †	18	28 1/2	9	28 1/4	11	1 1/2	7 5/16	1/4	9 1/4	8	7/8	17
6 †	19 1/2	32 1/2	9 3/4	32 1/2	12 1/2	1 5/8	8 1/2	1/4	10 5/8	12	7/8	21
600 lb. Standard Steam, 650 lb., 750° F.; boiler-feed, 720 lb., 450° F.; water (non-shock), 1000 lb., 100° F.												
1/2 †	6 1/2	5 7/8	3 1/4	7	3 3/4	9/16	1 3/8	1/4	2 5/8	4	1/2	4 3/8
3/4 †	7 1/2	8	3 3/4	8 1/2	4 5/8	5/8	1 11/16	1/4	3 1/4	4	5/8	5 1/4
1 †	8 1/2	9	4 1/4	9 3/4	4 7/8	11/16	2	1/4	3 1/2	4	5/8	6 1/4
1 1/4 †	9	10 1/4	4 1/2	10 1/2	5 1/4	13/16	2 1/2	1/4	3 7/8	4	5/8	7 1/8
1 1/2 †	9 1/2	11 1/4	4 3/4	10 3/4	6 1/8	7/8	2 7/8	1/4	4 1/2	4	3/4	8
2 †	11 1/2	14 3/4	5 3/4	14 3/4	6 1/2	1	3 5/8	1/4	5	8	5/8	9 1/4
2 1/2 †	13	17 1/8	6 1/2	17	7 1/2	1 1/8	4 1/8	1/4	5 7/8	8	3/4	10 3/4
3 †	14	19 3/4	7	20	8 1/4	1 1/4	5	1/4	6 5/8	8	3/4	12 1/2
3 1/2 †	15	21 3/4	7 1/2	22	9	1 3/8	5 1/2	1/4	7 1/4	8	7/8	12
4 †	17	25	8 1/2	25 1/4	10 3/4	1 1/2	6 3/16	1/4	8 1/2	8	7/8	14 3/4
5 †	20	28 1/4	10	27 1/2	13	1 3/4	7 5/16	1/4	10 1/2	8	1	17
6 †	22	32 1/2	11	32	14	1 7/8	8 1/2	1/4	11 1/2	12	1	21

* Screwed outside screw and yoke bonnet. † Flanged outside screw and yoke bonnet.

Table 65.—Dimensions of Iron Body Wedge Gate Valves, without By-pass

(Crane Co., Chicago)

All dimensions in inches. See Fig. 54. For valves with by-pass, see Table 69

Diam. of Port	End to End		Center to Top, Open		Diam. Hand Wheel	Flange Dimensions (plain face)				
	Flanged	Screwed	Non-rising Stem	Outside Screw and Yoke		Diameter	Thickness	Diam. Bolt Circle	No. of Bolts	Diam. of Bolts
A	*B*	*C*	*G*	*J*	*F*	*D*	*E*	*P*	*N*	*d*
2	7	5 7/16	11 1/4	15	8	6	5/8	4 3/4	4	5/8
2 1/2	7 1/2	5 7/8	12 3/4	16 1/2	8	7	11/16	5 1/2	4	5/8
3	8	6 1/8	14 3/4	19 1/2	9	7 1/2	3/4	6	4	5/8
3 1/2	8 1/2	6 1/2	15 1/4	21 1/2	9	8 1/2	13/16	7	8	5/8
4	9	6 7/8	16 1/4	24 3/4	10	9	15/16	7 1/2	8	5/8
5	10	7 3/8	19	29 1/2	12	10	15/16	8 1/2	8	3/4
6	10 1/2	7 3/4	21 1/4	32 1/2	12	11	1	9 1/2	8	3/4
8	11 1/2	8 3/4	26	41	14	13 1/2	1 1/8	11 3/4	8	3/4
10	13	9 7/8	31	50	16	16	1 3/16	14 1/4	12	7/8
12	14	11 5/8	36	57 1/4	18	19	1 1/4	17	12	7/8
14	15	......	39 1/4	67 3/4	20	21	1 3/8	18 3/4	12	1
16	16	......	44 1/4	76 1/4	22	23 1/2	1 7/16	21 1/4	16	1
18	17	......	48 3/4	83 1/2	24	25	1 9/16	22 3/4	16	1 1/8
20	18	......	52 1/2	91 1/4	24	27 1/2	1 11/16	25	20	1 1/8
24	20	......	63 1/2	109	30	32	1 7/8	29 1/2	20	1 1/4
30	24	......	75 1/2	137	36	38 3/4	2 1/8	36	28	1 1/4
36	28	......	......	158 1/2	*	46	2 3/8	42 3/4	32	1 1/2

* Geared valve.

Table 66.—Dimensions of Ferrosteel Wedge Gate Valves without By-pass

(Crane Co., Chicago)

All dimensions in inches. See Fig. 54. For valves with by-pass see Table 67

Diam. of Port	End to End		Center to Top, Open		Diam. of Hand Wheel	Flange Dimensions (raised face)					
	Flanged*	Screwed	Non-rising Stem	Outside Screw and Yoke		Diameter	Thickness*	Diameter of Flange	Diameter of Raised Face	No. of Bolts	Diameter of Bolts
	B	*C*						*P*		*N*	*d*
					250-lb. Valves						
1 1/4	6 1/2	5 1/2	9 1/4	11 1/2	7	5 1/4	3/4	3 7/8	3 1/16	4	5/8
1 1/2	7 1/2	6 1/4	9 3/4	13 1/4	8	6 1/8	13/16	4 1/2	3 9/16	4	3/4
2	8 1/2	7	11	14 1/4	8	6 1/2	7/8	5	4 3/16	8	3/4
2 1/2	9 1/2	8	13 1/4	17 1/2	9	7 1/2	1	5 7/8	4 15/16	8	3/4
3	11 1/8	9	14 3/4	20 1/2	10	8 1/4	1 1/8	6 5/8	5 11/16	8	3/4
3 1/2	11 7/8	10	15 1/2	22	10	9	1 3/16	7 1/4	6 5/16	8	3/4
4	12	11	17 1/2	24 3/4	12	10	1 1/4	7 7/8	6 15/16	8	3/4
5	15	13 1/2	20 1/4	29 3/4	14	11	1 3/8	9 1/4	8 5/16	8	3/4
6	15 7/8	15 7/8	23	34 1/4	16	12 1/2	1 7/16	10 5/8	9 11/16	12	3/4
8	16 1/2	16 1/2	28 3/4	42 3/4	20	15	1 5/8	13	11 15/16	12	7/8
10	18	18	33 3/4	52 3/4	22	17 1/2	1 7/8	15 1/4	14 1/16	16	1
12	19 3/4	19 3/4	37 1/4	60	24	20 1/2	2	17 3/4	16 7/16	16	1 1/8
14	22 1/2	....	42 3/4	69 1/4	24	23	2 1/8	20 1/4	18 15/16	20	1 1/8
16	24	....	45 3/4	75 1/4	27	25 1/2	2 1/4	22 1/2	21 1/16	20	1 1/4
18	26	....	50	83	30	28	2 3/8	24 3/4	23 5/16	24	1 1/4
20	28	....	53 3/4	93	30	30 1/2	2 1/2	27	25 9/16	24	1 1/4
24	31	....	66	112 1/4	36	36	2 3/4	32	30 5/16	24	1 1/2
					800-lb. Valves						
2	11 1/2	11 1/2		18 1/2	10	6 1/2	1 1/4	5	3 5/8	8	5/8
2 1/2	13	13 1/2		20 3/4	12	7 1/2	1 3/8	5 7/8	4 1/8	8	3/4
3	14	14 1/2		23 1/2	14	8 1/4	1 1/2	6 5/8	5	8	3/4
4	17	16 1/2		28	16	10 3/4	1 7/8	8 1/2	6 3/16	8	7/8
5	20	18 1/2		32 1/4	18	13	2 1/8	10 1/2	7 5/16	8	1
6	22	20		36 3/4	20	14	2 1/4	11 1/2	8 1/2	12	1
8	26			46 1/4	24	16 1/2	2 1/2	13 3/4	10 5/8	12	1 1/8
10	31			54 3/4	27	20	2 7/8	17	12 3/4	16	1 1/4
12	33			63 3/4	30	22	3	19 1/4	15	20	1 1/4

* Includes raised face of 1/16 in. on 250-lb. valves and of 1/4 in. on 800-lb. valves, finished with concentric grooves, approximately 16 grooves per inch.

Table 67.—Dimensions of Ferrosteel Wedge Gage Valves with By-pass

(Crane Co., Chicago)

All dimensions in inches. See Fig. 56. For dimensions of flanges see Table 66

Diam. of Port	End to End, Flanged B	By-pass Size Bolted	By-pass Size Built-up	By-pass Center to Outside Bolted C	By-pass Center to Outside Built-up U	Bevel-geared Non-rising Stem Center to Center E	Bevel-geared Non-rising Stem Center to Top	Bevel-geared Outside Screw and Yoke Center to Center	Bevel-geared Outside Screw and Yoke Center Top	Spur-geared Non-rising Stem Center to Center J	Spur-geared Non-rising Stem Center to Top K	Spur-geared Outside Screw and Yoke Center to Center L	Spur-geared Outside Screw and Yoke Center to Top M	Diam. of Hand Wheel, Geared Valves N
250-lb. Valves														
5	15	1 1/4			11 3/4									
6	15 7/8	1 1/4			12 1/2									
8	16 1/2	1 1/2		17	13 1/4	24	14 3/4		15	6 1/4	35 1/2		41 1/4	18
10	18	1 1/2		18 3/4	16 1/2	27 1/2	18 1/4	42 1/2	19	6 1/4	40 1/4	14 3/4	51 1/4	20
12	19 3/4	2		20 3/4	17 3/4	30 3/4	19 3/4	47 1/4	20	6 1/4	43 3/4	14 3/4	56 1/4	22
14	22 1/2	2		22 1/4	20	36 1/2	19 3/4	54 1/4	20	8 11/16	51	14 3/4	63 1/4	22
16	24	3		25 3/4	21 1/2	39 1/4	20		22 1/4	8 11/16	53 3/4	17 1/4	68 1/4	24
18	26	3		27	23	43 1/4	21	65	23 1/4	8 11/16	58 1/4	19	74	27
20	28	3	1 1/2	28 1/4	24 1/4	47		71 3/4	23 1/4	8 11/16	62	19		27
24		4	2			59 1/2	29	86 1/2	24	14 5/16	76 1/2	22 1/2	98 1/2	30
800-lb. Valves														
5	20	1 1/4	.....	14 1/4	.....	....	....	....	....	....	....	....	....	....
6	22	1 1/4	.....	15	.....	....	....	30 1/4	15	....	....	12	36 1/4	18
8	26	1 1/2	.....	18	.....	....	....	37 1/4	20	....	....	14 3/4	46 1/4	22
10	31	1 1/2	.....	19 3/4	.....	....	....	44	22 1/4	....	....	17 1/4	52 1/2	24
12	33	2	.....	21 3/4	.....	....	....	50 1/2	23	....	....	17 1/4	59 3/4	27

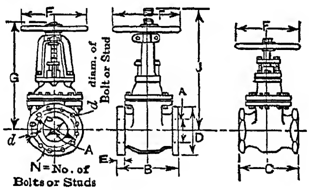


FIG. 54.

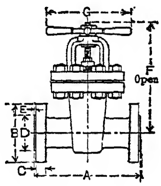


FIG. 55.

Table 68.—Dimensions of Flanged Cast Steel Gate Valves

(Consolidated Ashcroft Hancock Co., Bridgeport, Conn.)

All dimensions in inches. See Fig. 55. For dimensions of flanges, see Table 66

Size	300 lb. Steam, 750° F.* Face to Face A	300 lb. Steam, 750° F.* Center to Top, Open F	300 lb. Steam, 750° F.* Diam. Hand Wheel G	400 lb. Steam, 750° F.† Face to Face A	400 lb. Steam, 750° F.† Center to Top, Open F	400 lb. Steam, 750° F.† Diam. Hand Wheel G	600 lb. Steam, 750° F.‡ Face to Face A	600 lb. Steam, 750° F.‡ Center to Top, Open F	600 lb. Steam, 750° F.‡ Diam. Hand Wheel G
1	6	6 3/4	4	8 1/2	9 5/8	7 1/8	8 1/2	9 5/8	7 1/8
1 1/4	6 5/8	7 7/8	4 1/2	9	10 1/2	7 1/8	9	10 1/2	7 1/8
1 1/2	9 1/8	8 3/4	5 3/16	9 1/2	10 3/4	8	9 1/2	10 3/4	8
2	10	10	6 1/4	11 1/2	15	9 1/4	11 1/2	15	9 1/4
2 1/2	11	12 1/8	7 1/2	13	17	11	13	17	11
3	12	13 1/4	7 1/2	14	19	11	14	19	11

* 375 lb. boiler feed, 400° F.; 500 lb. water (non-shock), 100° F.
† 500 lb. boiler feed, 450° F.; 750 lb. water (non-shock), 100° F.
‡ 720 lb. boiler feed, 450° F.; 1000 lb. water (non-shock), 100° F.

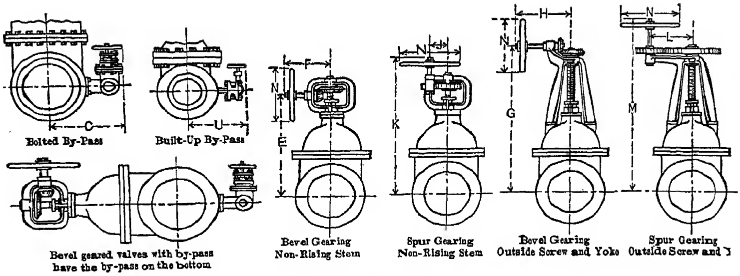


FIG. 56. Gate Valves with By-pass

Table 69.—Dimensions of Iron Body Wedge Gage Valves with By-pass

(Crane Co., Chicago)

All dimensions in inches. See Fig. 56. For dimensions of flanges, see Table 65

Diam. of Port	End to End, Flanged	By-pass		Bevel-geared Valves				Spur-geared Valves				Diam. Hand Wheel, Geared Valves
				Non-rising Stem		Outside Screw and Yoke		Non-rising Stem		Outside Screw and Yoke		
		Size	Center to Out-side	Center to Center	Center to Top	Center to Center	Center to Top	Center to Center	Center to Top	Center to Center	Center to Top	
A	*B*		*C*	*E*	*F*	*G*	*H*	*J*	*K*	*L*	*M*	*N*
12	14	2	19	31 1/2	14 1/2	45 3/4	14 3/4	6 1/4	43	12	51 3/4	16
14	15	2	20 1/4	36 1/4	14 3/4	53 1/2	15	6 1/4	47 3/4	12	59 3/4	18
16	16	3	23 3/4	41 1/2	19	60	19	8 11/16	55 1/2	14 3/4	68 1/2	20
18	17	3	25 1/4	43 1/2	19 1/2	64 1/2	20	8 11/16	57 3/4	14 3/4	73 1/2	22
20	18	3	26 1/4	47	19 1/2	70 1/4	20	8 11/16	61 1/4	14 3/4	79 1/4	22
24	20	4	29 1/4	55 1/2	21	84 1/2	22 1/2	8 11/16	70 1/2	17 1/4	93 1/4	27
30	30	4	34	68 1/2	29	105 1/2	29	14 5/16	85 3/4	19	116 3/4	30
36	32	6	41 3/4	83	31	122 1/2	31	16 1/4	100 1/2	19	133	30

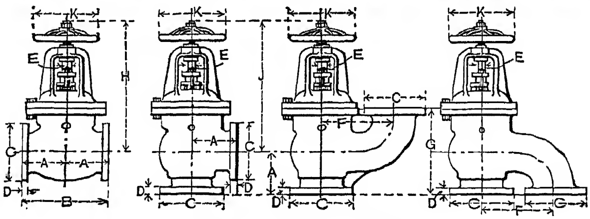


FIG. 57. Non-return Valves

Table 70.—Dimensions of Non-Return Valves, Plain and Triple Acting

(Golden-Anderson Valve Specialty Co., Pittsburgh)

All dimensions in inches. See Fig. 57

Size	*A*	*B*	*C*	*D*	*E*	*F*	*G*	*H*	*J*	*K*	Shell Thick-ness
2 1/2	5 3/4	11 1/2	7 1/2	7/8	7/8	9	11 1/2	17 1/2	16	8	1/2
3	6 1/4	12 1/2	8 1/4	1	1	10 1/2	12 1/2	19 1/2	19	8	5/8
4	7	14	10	1 1/4	1 1/4	12 1/2	14	24	24	8	3/4
5	7 7/8	15 3/4	11	1 3/8	1 3/8	13 1/2	15 3/4	28	27	12	7/8
6	8 3/4	17 1/2	12 1/2	1 7/16	1 1/2	15	17 1/2	29 1/2	30	12	1
7	9 5/8	19 1/4	14	1 1/2	1 5/8	16	19 1/4	31 1/2	30	15	1 1/16
8	10 1/2	21	15	1 13/16	1 3/4	17 1/4	21	35	33	15	1 1/8
10	12 1/4	24 1/2	17 1/2	1 7/8	2	20 1/2	25 1/4	39 1/2	37	15	1 1/4
12	14	28	20 1/2	2	2 1/4	24	30	45	45	20	1 3/8

Section 6

THE STEAM BOILER

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THE STEAM BOILER *

Steam boilers are of many types, starting, in so far as steaming capacity goes, with small miniature boilers such as are used in tailor shops, and ending with the large boilers used in central power plants. Steam pressures vary from 15 lb. per sq. in. as a maximum for house heating boilers to 1600 lb. per sq. in. and over for central power plants. In special cases forced-circulation boilers are built to operate at pressures up to the maximum at which steam can be generated, *i.e.*, the critical pressure of 3226 lb. per sq. in. In such boilers pressure may be even higher although critical pressure is exceeded.

UNITS OF CAPACITY AND EFFICIENCY

MEASURES FOR COMPARING PERFORMANCE OF BOILERS.—Horsepower. —The term Boiler Horsepower was standard for many years, but is being used to a lesser and lesser extent. It no longer is included in standard codes. A boiler horsepower is the evaporation of 34.5 lb. of water per hour from a temperature of 212° F. into dry saturated steam at the same temperature. This unit never has been used in Europe or for marine boilers built in the U. S.

Builders' Rated Horsepower.—In stationary practice, 10 sq. ft. of heating surface has been considered as equivalent to 1 Boiler Horsepower, based on the evaporation of 3.45 lb. of water per hour from and at 212° F. per sq. ft. of heating surface.

Evaporation.—Water evaporated into steam per square foot of heating surface per hour. Evaporation usually is reduced to equivalent evaporation, *i.e.*, evaporation from water at 212° F. to steam at 212° F.

The *Factor of Evaporation, F*, is the ratio of actual B.t.u. absorption from feed temperature per lb. of steam, whether wet, saturated or superheated, to the latent heat of steam at atmospheric pressure, or 970.2 B.t.u. per lb. $F = (H - h)/970.2$, where H = heat content of 1 lb. of steam, and h = heat content of 1 lb. of feedwater. Equivalent evaporation = (actual evaporation $\times$ factor of evaporation).

Rapidity of Combustion, measured by fuel burned per hour per square foot of grate surface, per cubic foot of furnace volume, or per foot of furnace width.

Efficiency is measured by the ratio (B.t.u. per lb. of fuel absorbed in evaporating and superheating the steam) $\div$ (B.t.u. per lb. of fuel). It will be the same whether based on fuel-as-fired or on dry fuel. In the U. S., efficiency is based on the higher heating value of the fuel. In certain cases, however, it is based on the lower heating value, because the latent heat of moisture, formed by burning the hydrogen in the fuel, is not available to generate steam in a boiler. The lower heating value frequently is used abroad.

With certain fuels, the refuse may contain considerable unconsumed combustible. While the boiler itself cannot be charged with failure to absorb heat not released from the fuel, nevertheless heat represented by unconsumed combustible must be charged against the combined steam-generating unit. Efficiency as ordinarily stated, therefore, is combined efficiency of boiler, superheater, furnace and fuel burning equipment.

A boiler manufacturer guaranteeing performance of a combined unit, including design and operation of elements (stokers, grates, burners, etc.) supplied by others, must require, if efficiency is to have a practical value, certain performance guarantees relative to those elements. Usually such guarantees state maximum or minimum limitations of excess air (or CO_2) and CO in flue gases, and of unconsumed combustibles in ash or refuse, as well as in flue gases.

Factors Affecting Efficiency.—Operation of a steam-generating unit involves the processes of, *a*, converting potential energy of fuel into heat energy, and *b*, transferring this heat energy to a medium (steam) which may be applied to useful purposes. In practice certain characteristic losses of energy occur. The summation of these losses,

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* The section on Steam Boilers has been revised by the editor-in-chief in collaboration with Dr. D. S. Jacobus and Mr. Robert K. Behr, except where otherwise noted in the text.

in terms of percent of original energy in the fuel, are represented by the difference between the percent efficiency and 100 percent. Unavoidable losses result from the necessity of discharging products of combustion at a temperature above that of the temperature of the fuel and of the air for combustion, and from hydrogen and moisture in the fuel. Avoidable losses, which can be controlled to a certain extent, result from excess air that is heated to temperature of exit gases; from moisture accompanying such excess air; from unconsumed solid combustible in ash and refuse; from unconsumed gaseous combustible in exit gases; and from heat radiated from the setting. Characteristic heat losses in a boiler or steam-generating unit usually are identified as: *

1. Loss Due to Moisture in Fuel.—This loss is due to evaporating moisture in the fuel and superheating it to temperature of the flue gas. Temperature at which evaporation begins may be quite low, due to the low partial pressure of water vapor in gaseous products of combustion. Heat lost is the difference between total heat of water vapor at exit gas temperature and heat of the liquid water at initial temperature, which is taken as fuel temperature. B.t.u. lost per lb. of fuel $= W_m(1090.7 - t_f + 0.455\,T)$ where $W_m =$ lb. of moisture per lb. of fuel; $T =$ temperature of flue gases, deg. F.; $t_f =$ temperature of fuel, deg. F.

2. Loss Due to Water from Combustion of Hydrogen.—Hydrogen in any fuel burns to water, and is discharged with the flue gases as superheated steam. The loss is the difference in the total heat of water vapor at exit temperature and of liquid water at fuel temperature. B.t.u. lost per lb. of fuel $= 9H(1090.7 - t_f + 0.455\,T)$ where $H =$ weight of hydrogen in fuel, lb. per lb. of fuel, other notation as above.

In Europe, latent heat of vapor formed by combustion of hydrogen often is deducted from heating value of the fuel, to obtain the so-called lower heat of combustion. This may result in an increase in efficiency of, say, 3% with a volatile coal, and 6% with oil, over efficiency obtained by using the higher heat value, which is the standard in the U. S.

If, in the ultimate analysis of fuel-as-fired, free moisture is included as hydrogen and oxygen, the above formula will give total moisture loss, and calculation of loss due to free, or surface, moisture as outlined under (1) may be omitted.

3. Loss Due to Moisture in the Air.—This moisture enters the furnace as water vapor. Heat lost is the difference in total heat of water vapor at exit gas temperature and at temperature of air for combustion. Amount of moisture per 1 lb. of air supplied for combustion is determined from wet and dry bulb thermometer readings and the use of a psychrometric chart. B.t.u. lost per lb. of fuel $= w \times m \times 0.47(T - t)$, where $w =$ lb. of dry air supplied per lb. of fuel; $m =$ lb. of moisture per lb. of dry air; $T =$ temperature of flue gases, deg. F.; $t =$ temperature of air surrounding boiler, or re-entering air heater. This loss is small and often is included in unaccounted losses.

4. Loss Due to Dry Chimney Gases is a loss of sensible heat carried by " dry " constituents of flue gases which leave the unit at a temperature higher than that of the surrounding air. B.t.u. lost per lb. of fuel $= 0.24\,W(T - t)$, where $W =$ lb. of dry flue gas per lb. of fuel; $T =$ temperature of flue gases, deg. F.; $t =$ temperature of air surrounding boiler or entering air heater, deg. F. (customarily accepted as datum temperature).

5. Loss Due to Incomplete Combustion of Carbon.—The presence of CO as detected by flue gas analysis indicates incomplete combustion. A small amount is indicative of an appreciable heat loss. B.t.u. lost per lb. of fuel $= (10{,}160 \times C \times CO) \div (CO_2 + CO)$ where $C =$ lb. of carbon burned per lb. of fuel $=$ lb. of carbon per lb. of fuel (from ultimate analysis) minus lb. of combustible in refuse per lb. of fuel; CO_2 and $CO =$ percentages by volume of carbon dioxide and carbon monoxide in the flue gases; $10{,}160 =$ difference in B.t.u. evolved in burning 1 lb. of carbon to CO_2 and to CO.

6. Loss Due to Unconsumed Combustible in Refuse.—Usually, this loss is based on unconsumed combustible matter discharged with furnace ash and refuse. However, for accuracy, the loss due to the unconsumed combustible particles carried into the setting, or up the stack, should be included. The calculation of this loss is quite essential in large boiler units and pulverized fuel installations. B.t.u. lost per lb. of fuel $= 14{,}600(R - A)$, where $14{,}600 =$ the heat value per lb. of carbon (all combustible assumed to be carbon); $R =$ lb. of refuse per lb. of fuel; $A =$ lb. of ash per lb. of fuel (from ultimate analysis).

7. Loss Due to Radiation, Unconsumed Hydrogen and Hydrocarbons, and Unaccounted Losses.—Radiation losses, due to the lack of a satisfactory means of determination, usually are grouped with unaccounted losses, and those due to unconsumed hydrogen and hydrocarbons to form a single item. Loss due to this combined item, in B.t.u. per lb. of fuel, is the difference between B.t.u. per lb. of fuel and the summation of the heat absorbed by the boiler and the six heat losses listed above. A heat balance may be given upon either a dry-fuel or a fuel-as-fired basis. Numerical values in all computations must be consistent with the basis used.

* See also Heat Balance under Rules for Conducting Boiler Tests, A.S.M.E. Power Test Code.

Capacity.—The capacity, or output, of a steam boiler, with or without integral superheater and economizer, may be stated as: 1. Total heat transferred through heating surfaces in B.t.u. per hr. (heat output in steam). 2. Units of evaporation per hour; a unit is 1000 B.t.u. and is called a kilo B.t.u. (*k*B). The mega B.t.u. (*m*B) of 1,000,000 B.t.u. sometimes is used for large installations. (3) Actual evaporation, lb. of steam per hr. at observed steam pressure and quality, or temperature, and observed feedwater temperature. This latter quantity, although useful, does not permit direct comparison of capacities of steam-generating units, because of the extreme range in operating conditions of boiler pressure, quality, superheat and feedwater temperature.

MEASUREMENT OF HEATING SURFACE.—Heating surface of a boiler is all that surface that is in contact with water or steam on one side and with gases or refractory being cooled on the other. Heating surface is measured on the side receiving heat. For details, see Test Code for Steam-Generating Apparatus, p. 16–13. If tubes are covered by blocks, studs, refractories, etc., special rules must be applied, for which no standards as yet (1935) exist.

DESIGN OF STEAM-GENERATING UNITS is largely an economic problem of how to so dispose the total heating surface as to give maximum efficiency at the lowest cost. The economic factors involved in determining the highest practical efficiency are: Load factor; type of load, whether constant or fluctuating; fuel cost; space limitations. Consideration of these factors fixes the desirable final temperature of the gases. The practicability of this temperature depends on the dew-point of the gases. A final temperature at maximum load of at least 50° to 75° F. above the dew-point of the gases is desirable to allow for decreased final gas temperature with decreased loads. With fuels high in sulphur, moisture condensed from the gases will combine with the SO_2 in them and initiate corrosion.

After the lowest practical final temperature of gases at maximum load has been fixed, the most economical disposition of heating surface, as between boiler, economizer and air heater is determined on an economic basis. For instance, the same final temperature may be obtained by using a large air heater and a small economizer, or *vice versa*, and the relative cost of the two arrangements may be the deciding factor.

EFFICIENCY AND PERFORMANCE OF STEAM BOILERS

RELATION BETWEEN BOILER EFFICIENCY AND CAPACITY.—In a well-designed boiler, steaming capacity per sq. ft. of heating surface is limited only by the amount of fuel that can be burned. The amount of heating surface provided for a given evaporation could be much smaller than is customary if other economic factors did not control such reduction in size. Factors limiting high evaporation rates from small surfaces are discussed below.

Efficiency.—Maximum efficiency may occur at a load above or below boiler rated capacity, and may remain practically constant over a considerable range of capacity. Capacity at which maximum efficiency is attained varies with size and type of boiler, type of fuel, method of firing, conditions of operation, etc. With hand-fired boilers, maximum efficiency is at or about boiler rated capacity; with stoker-fired boilers it generally is above normal rated capacity. Large stokers, common in central stations, tend to give maximum efficiency appreciably above boiler rated capacity; however, at low capacities combustion rate is so low that it is difficult to operate with a minimum of excess air.

Economizers tend to keep combined efficiency constant over a considerable range of capacities, due to the fact that at high capacity, increase in economizer efficiency compensates for decrease in boiler efficiency. The most economical load at which a boiler should operate is greater than the capacity which results in maximum combined thermal efficiency.

Gas Friction.—At high capacities, frictional resistance to flow of gases increases rapidly, resulting in an increase in power required to provide draft. In addition there is a probable loss due to a portion of the fuel being carried through the setting and discharged from the stack.

Feedwater must be practically free of ingredients that will cause foaming, priming or scale formation. At high capacities, distilled water may be necessary to avoid these difficulties.

Heat Liberation.—After maximum rate of combustion for any fuel has been established, the only method of increasing capacity is to burn more fuel. The amount is limited by firing equipment and furnace volume.

Maintenance.—At high capacities, cost of maintenance may be excessive, and may offset any gain effected by operating at such capacities.

Cleanliness.—Soot deposited on the tubes will seriously impair efficiency at high capacities, and provision must be made for removal of soot at frequent intervals.

LIMITATIONS OF CAPACITY.—The most important limitation on the steaming capacity of a boiler is the amount of fuel that can be burned in the furnace, the effect of arrangement and extent of heating surface being much less important. This is particularly true of boilers absorbing heat primarily through convection, where maximum evaporation per sq. ft. of heating surface decreases as amount of heating surface per sq. ft. of furnace width increases. The development of water walls, exposed to radiant heat of the furnace, has greatly increased evaporation per sq. ft. of heating surface. Another development, the steaming economizer, also has led to increased evaporation rates. The percent of rated capacity, based on the evaporation of 3.45 lb. of water from and at 212° F. per sq. ft. of heating surface per hr., has increased greatly, particularly in high-capacity units in which the boiler is practically a water screen protecting the superheater.

TYPES OF STEAM BOILERS AND THEIR APPLICATION

Steam boilers may be classified as to construction as water-tube or fire-tube boilers; as to material, as steel or cast-iron boilers; as to service, as power or heating boilers; as to use, as stationary, automotive or marine boilers.

The cylindrical structure, including the ends, of a fire-tube boiler, usually is called the shell. The cylinder superimposed on the tubes of a water-tube boiler is called the steam and water drum. Shells of Scotch-type marine boilers have been built as large as 16 ft. diameter. The largest steam and water drums of water-tube boilers built to date (1935) are 72 in. diameter.

1. WATER-TUBE BOILERS IN CENTRAL STATION SERVICE

ADVANCES IN STEAM BOILER PRACTICE.—Table 1 compares boilers installed in central power plants 30 years ago with present (1935) practice. The increase in pressure and capacity are due largely to the demands of the steam turbine.

The higher steam pressures and steam temperatures in use in 1935, and the higher rates of fuel burning, and consequently increased rates of evaporation, require a correspondingly higher grade of workmanship and the use of special materials, and have removed construction of steam generating equipment from the old-time boiler shop to a high-grade machine shop. The availability for service of large boiler units has become as high as that of the turbine units; the added efficiency has resulted in an advance on the economic side. The efficiency obtained with such large boilers and appurtenances is 85% and over, based on the higher heating value of the fuel. Should the lower heating value be used, efficiency of large generating units would be from 88 to 90%. Steam generating capacity has increased as well as efficiency, and boiler units with a single furnace have been built to evaporate 1,000,000 lb. of water into steam per hour as compared to about $^1/_3$ of this amount 10 or 15 years ago. Furnace design has formed an important part of the development. Water-cooled furnace walls and improved methods of burning fuel make it possible to operate for long periods of time without shut-down caused by furnace difficulties.

Fig. 1 shows a typical installation of a large central power plant boiler with 7214 sq. ft. of heating surface, 12,900 sq. ft. of superheater surface, a maximum evaporative capacity of 275,000 lb. of steam per hr. and built for a working pressure of 1400 lb. per sq. in. The furnace is water cooled, being arranged with Bailey blocks to form Bailey furnace walls (see p. 6–93). The boiler is fired with pulverized coal inter-tube burners, and has a Bailey water-cooled slag-tap furnace. Temperature of steam from the primary superheater is maintained at, substantially, 750° F. by passing a portion of it through a de-superheater at the higher loads, and returning heat from the steam to the boiler. Steam from an intermediate stage of the steam turbine passes to a reheater in the boiler setting. This,

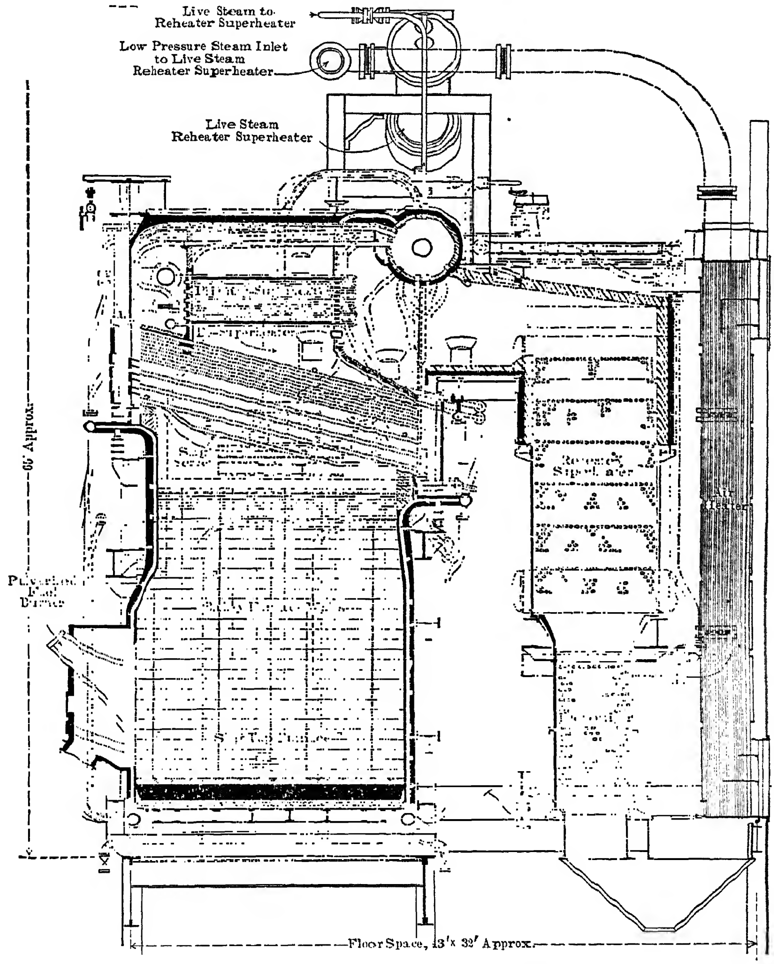
Table 1.—Comparison of Central Power Plant Boilers in 1905 and 1935

	1905	1935
Heating surface, largest unit	6,040	53,926
Heating surface, boiler, superheater, economizer and air heater, sq. ft.	7,000	144,402
Working pressure, lb. per sq. in.	225	650 to 1,600
Temperature superheated steam, deg. F. max.	550	700 to 900
Maximum rate of evaporation, lb. per hr. per boiler unit	30,000	1,000,000
Volume of boiler setting for largest boilers, cu. ft.	7,650	42,255*
Height, bottom of walls of setting to center of steam and water drum for largest boilers, ft.	19	78

* Includes two boilers set over a single furnace.

together with a live steam reheater, maintains a total temperature of superheated steam of about 750° F. irrespective of varying temperature of steam leaving the turbine. To accomplish this, steam from an intermediate stage of the turbine first passes through a live-steam reheater and then through the reheater in the boiler setting, where it is heated by the combustion gases.

The boiler unit is 65 ft. high measured from top of foundations. Area of the base is 1375 sq. ft. The tubes of boiler, superheater, de-superheater, live steam reheater, reheaters using gases as the heating medium, economizer, air heater and furnace placed end to end would extend approximately 23 miles. To drive the induced and forced blast fans requires a maximum of 400 Hp.; the feed pump requires 650 Hp. A special semi-manual apparatus controls combustion. Units of this type are in a class by themselves,



, 1. Central Station Boiler with Superheater Reheater, De-superheater and Economizer

and involve elements far beyond those that had to be considered in building smaller boilers in the old-time boiler shop.

Some engineers prefer to use, say, 900 lb. pressure and about 900° F. temperature of superheated steam, with no reheater to reheat steam between turbine stages. Others prefer 1400 lb. pressure with, say, 1200 lb. at the turbine throttle. In earlier practice, this would require using a reheater to avoid undue moisture in the low-pressure stages of the turbines. Even with a turbine operating at 1200 lb. steam pressure, present tendency is to heat the steam high enough to make a reheater unnecessary. As pressures increase, the use of a forced circulation or series type boiler instead of a natural circulation boiler may become advisable for certain types of work.

2. WATER-TUBE BOILERS FOR INDUSTRIAL USE

Water-tube boilers for industrial use are shown in Figs. 2, 3 and 4.

Fig. 2 shows a Stirling boiler with a chain grate stoker, a water-cooled furnace, an economizer and an air heater. Fig. 3 shows a Babcock & Wilcox boiler with a superheater between the first and second passes, equipped with an air heater, an underfeed stoker, and water walls in the furnace. A slag screen consisting of tubes spaced at double the distance apart of the tubes of the first bank, placed where the hot gases first impinge on the boiler, cools the gases and the contained particles of molten slag to a point where the slag will not adhere to the more closely spaced tubes. The double spacing is obtained

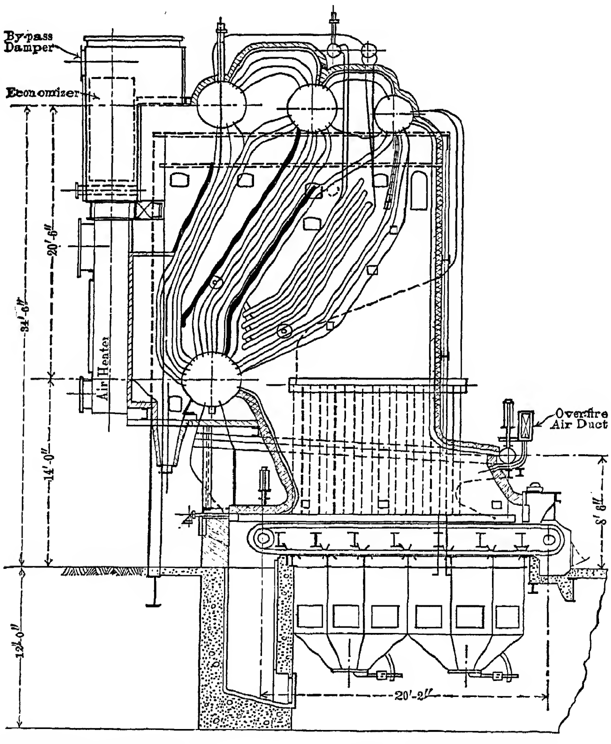


FIG. 2. Stirling Boiler and Chain Grate Stoker, Slag Screen, Water-cooled Furnace, Economizer and Air Heater

by bending outward alternate tubes in the front row. The superheater is placed back of the first bank of tubes.

Some engineers prefer straight-tube boilers, such as the Babcock & Wilcox boiler shown in Fig. 3, while others prefer to use bent-tube boilers similar to the Stirling. No rule can be given as to the best design of boiler for general use. Every case must be considered individually.

Fig. 4 shows a boiler with an integral water-tube furnace, particularly adaptable to burning pulverized coal, oil or gas. These boilers have been designed to evaporate from 4000 to 175,000 lb. of steam per hour. The gases make three horizontal passes through the boiler, giving a high efficiency with minimum draft drop.

The foregoing boilers all are fitted with superheaters, which is common practice for power boilers. For process work, where constant temperature is required, superheaters are omitted.

Waste Heat Boilers

Waste heat is sensible heat in gases discharged, usually from some high-temperature process, as a heating furnace. A number of Babcock & Wilcox and Stirling boilers, as

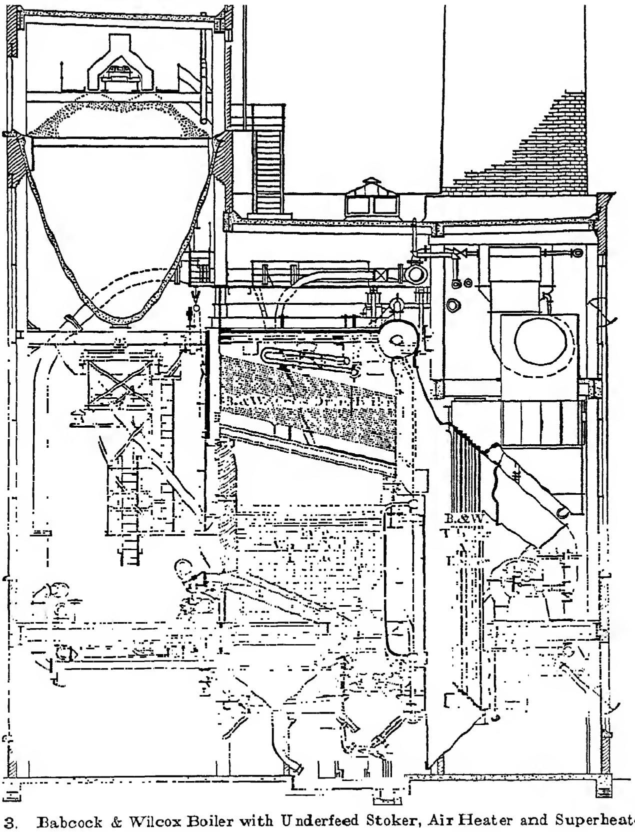


Fig. 3. Babcock & Wilcox Boiler with Underfeed Stoker, Air Heater and Superheater

shown in Figs. 2 and 3, have been used as waste heat boilers. Stirling boilers have been used in the smelter industry, with baffles removed, or arranged to give minimum draft drop, so that the high stacks used would provide enough draft to operate the smelting furnaces and draw waste gases through the boilers. In later practice, as in waste heat boilers for open hearth furnaces, a higher velocity of gases over the tubes was found to be of advantage. This involves a higher draft loss through the boiler and necessitates using induced draft fans. Special types of boilers, Figs 5 and 6, were later developed.

In the arrangement shown in Fig. 5 the gases pass longitudinally over vertical tubes. In Fig. 6 they pass in a generally horizontal direction through the tubes of a fire-tube boiler.

3. BOILERS USING BLAST-FURNACE GAS

Earlier steel mill power plants comprised a number of small boiler units, some designed for using blast-furnace gas exclusively, others being fired with coal on stokers to supplement the gas-fired units, and to carry the entire load when the gas went off. Pulverized coal firing and improvement in gas burners and furnace design and construction has

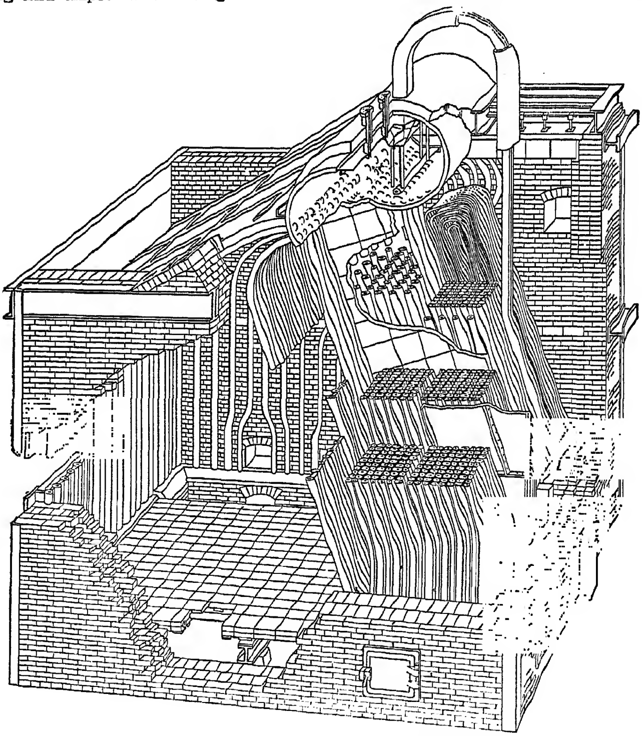


Fig. 4. Boiler with Integral Water-tube Furnace

made possible the firing of blast furnace gas and of coal in the same furnace. This decreases capital investment in plant for a given power requirement, and results in more complete and efficient use of available gas supply.

BURNER DESIGN.—The efficiency of blast-furnace gas burners is due primarily to methods of breaking the gas into thin streams and so introducing air for combustion as

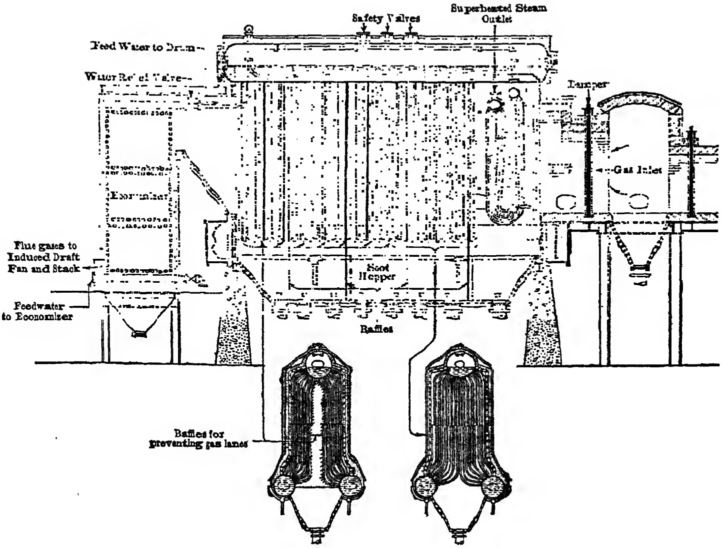


FIG. 5. Three-drum, Vertical-tube Waste Heat Boiler

to insure thorough mixture before ignition. Originally, aspirating burners were used, but the trend is toward a design in which both gases and air for combustion are brought to the burner under pressure. The air supply can be adjusted for varying gas conditions, and can be maintained at a minimum regardless of gas pressure in the main. Blast-furnace gas may be burned with as little as 15% excess air, instead of 100% excess air required in early box-type burners. Systems of automatic control permit speeds of induced and forced draft fans to be regulated by pressure in the gas main. Fig. 7 is a plan of a Babcock & Wilcox cross-tube blast-furnace gas burner.

FURNACE AND BOILER DESIGN.—Because of its low calorific value, blast-furnace gas burns slowly. Temperatures attained in the furnace are much lower than with other common fuels. Relatively large furnace volumes are required to insure complete burning before gases enter the tube bank, and when firing blast-furnace gas alone, no water-cooling

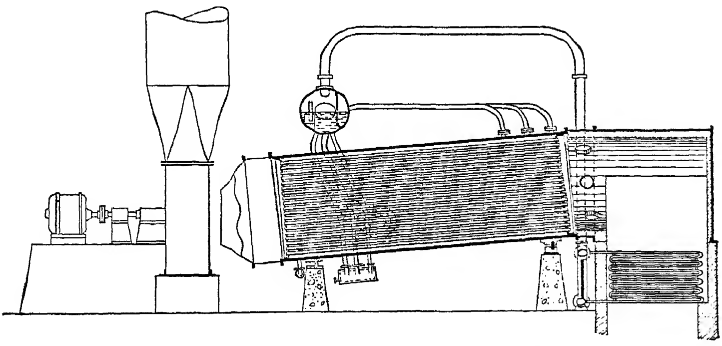


FIG. 6. Horizontal Fire-tube Waste Heat Boiler

other than that of the boiler is required in furnaces. When pulverized coal is fired in combination with gas, or is fired in the same furnace with " gas off ", water-cooled wall construction is necessary in certain zones to prevent destruction of brickwork and to facilitate removal of ash. Present practice (1935) indicates furnace volumes of 50 to 60 cu. ft. per million B.t.u. released per hr., compared with 30 to 40 cu. ft. in early installations. Water cooling of rear wall and hopper floor is customary, although the extent of such water-cooled surfaces should be governed by anticipated loads and ratios of firing gas and auxiliary fuel. Preheated air and the turbulence given by well-designed burners produce an appreciably shorter flame than heretofore obtained.

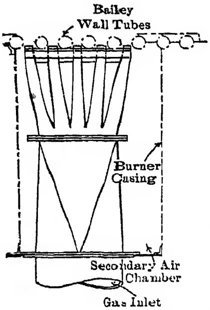


FIG. 7. Burner for Blast-furnace Gas

Since high furnace temperatures are favorable to high overall rates of heat absorption it is necessary to offset the lower furnace temperature obtained with blast-furnace gas if efficiencies are to be comparable to those of boilers fired with other fuels. Heat absorbed from direct radiation in a blast-furnace gas-fired boiler is about 40% of that absorbed in a similar stoker-fired boiler. If total absorption is to be comparable in both cases, absorption by convection must be increased in the boiler fired by blast-furnace gas. As in waste heat practice, high gas velocities will result in high rates of heat transfer by convection. High gas velocities are inherent in boilers fired with blast-furnace gas, since even with best combustion conditions, products of combustion weigh approximately 70 lb. per developed horsepower, as compared to about 45 lb. in stoker-fired boilers. Gas velocity may be further increased by decreasing the ratio (gas passage area through boiler ÷ boiler heating surface). The high gas velocity develops high frictional resistance, which usually necessitates using induced draft fans.

ECONOMIZERS USED WITH BLAST-FURNACE GAS BOILERS.—Despite low exit gas temperatures from boilers fired with blast-furnace gas, economizers, when installed, have less heating surface than those in stoker-fired practice. This is due to the fact that high gas velocity gives a high ratio of gas weight to weight of water in the economizer. The resultant heat transfer rates are sufficiently higher than those in stoker-fired practice to attain a given feed-water temperature with a smaller amount of economizer surface.

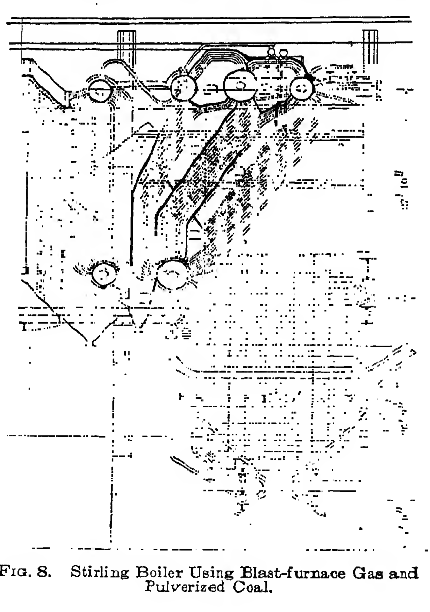


FIG. 8. Stirling Boiler Using Blast-furnace Gas and Pulverized Coal.

TYPICAL BOILERS.—A typical installation (1928) consisted of 1460 Hp. Stirling boilers equipped with superheaters, economizers, air heaters, air-cooled side walls, water-cooled hopper floors and bare tubes protecting the front wall. Units are fired both with blast-furnace gas and pulverized coal. At working pressures of 375 lb. per sq. in. and capacities of 175,000 lb. of steam per hr., overall efficiency has been about 83%. Draft loss through the unit was 7.7 in. of water. A unit installed in the same plant in 1932 (see Fig. 8) shows the trend toward higher capacities. The unit was designed for a capacity of 200,000 lb. of steam per hr., efficiency of 85%, and draft loss of 6.8 in. when burning blast-furnace gas, and a capacity of 300,000 lb. of steam per hr., efficiency of over 86% and draft loss of 6.7 in. when burning pulverized coal. In this unit refractory-faced water-cooled side walls have been added to permit operation at the higher rates with pulverized coal.

4. FIRE-TUBE AND SMALL WATER-TUBE BOILERS

HORIZONTAL RETURN TUBULAR BOILERS are used widely for small heating plants where working pressures ordinarily do not exceed 125 or 150 lb. per sq. in. In certain instances, they are built for considerably higher pressures. Fig. 9 shows a typical arrangement of a hand-fired horizontal return tubular boiler. This is the standard setting recommended by the Hartford Steam Boiler Inspection and Insurance Co.

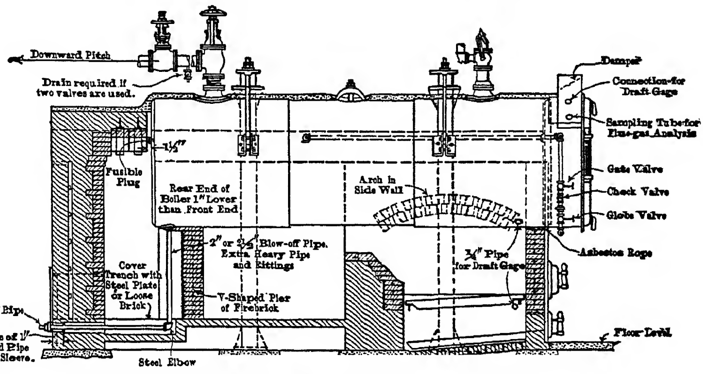


Fig. 9. Setting of Horizontal Return Tubular Boiler

The arrangement of tubes in horizontal return tubular boilers varies. Typical arrangements are shown in Fig. 10. The usual sizes of boilers and the heating surface available with different arrangement of tubes are given in Table 1. Weights of boilers are given in

Table 1.—Heating Surface of Horizontal Return Tubular Boilers, with Manholes below Tubes

(*Boiler Book* of Hartford Steam Boiler Inspection and Insurance Co.)

Boiler Diam., in.	Tubes			Heating Surface, sq. ft.			
	Length, ft.	Diam., in.	Number	Tubes	Shell	Rear Head	Total
54	14	3	52	531	99	8	638
	16	3	52	607	113	8	728
	14	3 1/2	44	526	99	8	633
	16	3 1/2	44	602	113	8	723
	14	4	36	494	99	7	601
	16	4	36	565	113	7	686
60	16	3	74	864	125	9	998
	18	3	74	972	141	9	1122
	16	3 1/2	52	711	125	10	846
	18	3 1/2	52	800	141	10	951
	16	4	46	722	125	9	856
	18	4	46	812	141	9	962
66	16	3	94	1098	138	11	1247
	18	3	94	1235	156	11	1402
	16	3 1/2	74	1012	138	11	1161
	18	3 1/2	74	1138	156	11	1305
	16	4	56	878	138	11	1027
	18	4	56	988	156	11	1155

Boiler Diam., in.	Tubes			Heating Surface, sq. ft.			
	Length, ft.	Diam., in.	Number	Shell	Tubes	Rear Head	Total
72	16	3	122	1425	151	13	1589
	18	3	122	1603	170	13	1786
	20	3	122	1781	188	13	1982
	16	3 1/2	98	1340	151	12	1503
	18	3 1/2	98	1508	170	12	1690
	20	3 1/2	98	1675	188	12	1875
	16	4	74	1161	151	12	1324
	18	4	74	1306	170	12	1488
	20	4	74	1451	188	12	1651
78	16	3	146	1705	163	15	1883
	18	3	146	1918	184	15	2117
	20	3	146	2131	204	15	2350
	16	3 1/2	112	1532	163	15	1710
	18	3 1/2	112	1723	184	15	1922
	20	3 1/2	112	1915	204	15	2134
	16	4	91	1427	163	14	1604
	18	4	91	1606	184	14	1804
	20	4	91	1784	204	14	2002
84	18	3	176	2312	198	17	2527
	20	3	176	2569	220	17	2806
	18	3 1/2	138	2123	198	16	2337
	20	3 1/2	138	2359	220	16	2595
	18	4	108	1906	198	16	2120
	20	4	108	2118	220	16	2354

Table 2. The weights given do not include fronts, trimmings, grates or other castings. The weight of water will vary with the number and diameter of tubes. The weights of

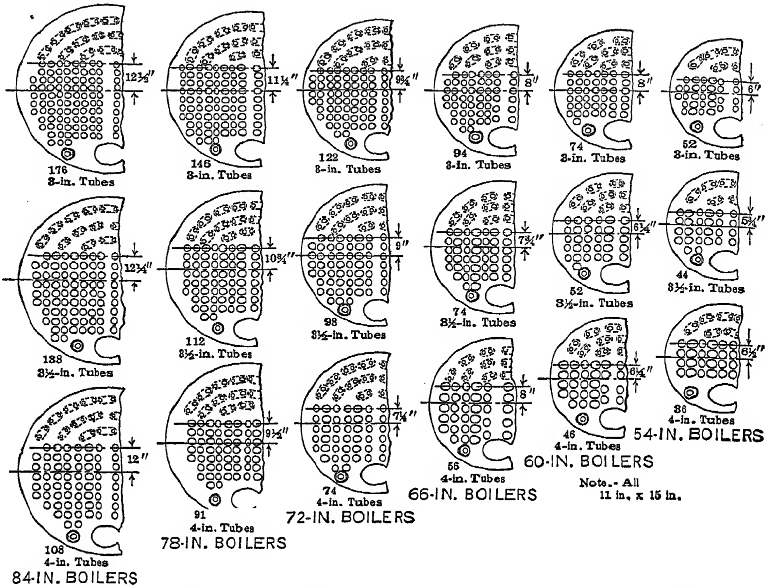


Fig. 10. Arrangement of Tubes and Braces in Horizontal Return Tubular Boilers

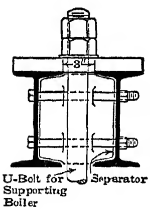


Fig. 11. I-Beam Suspension for Horizontal Tubular Boilers

water given in Table 2 are for average cases and are based on a weight of water of 62.5 lb. per cu. ft.

Suspension of Horizontal Tubular Boilers.—Horizontal tubular boilers are suspended from pairs of I-beams, as shown in Fig. 9, carried on columns. Round or square cast-iron columns or built-up plate-and-angle columns sometimes are used, but present practice uses the H-section as a column. Table 3 shows the sizes and weights of I-beams and H-sections recommended for various sizes of boilers. The size of I-beams is based on a fiber stress of 12,500 lb. per sq. in. The beams should be bolted together in pairs as shown in Fig. 11. The maximum ratio of slenderness of columns (l/r) has been taken as 120. Table 3 gives a heavier column than would be given by the standard tables of structural sections, since the stresses in boiler columns are not balanced nor distributed equally. The maximum length of column given in Table 3 should not be exceeded. The table assumes that the col-

Table 2.—Approximate Weight of Horizontal Tubular Boilers

(*Boiler Book* of Hartford Steam Boiler Inspection and Insurance Co.)

Diam. of Boiler, in.	Length of Tubes, ft.	Weight of Bare Boiler, lb. 125 lb. Pressure	Weight of Bare Boiler, lb. 150 lb. Pressure	Weight of Water, lb.	Diam. of Boiler, in.	Length of Tubes, ft.	Weight of Bare Boiler, lb. 125 lb. Pressure	Weight of Bare Boiler, lb. 150 lb. Pressure	Weight of Water, lb.
54	14	9,100	9,700	11,700	72	16	18,400	20,000	22,500
54	16	10,100	10,800	13,400	72	18	20,000	21,700	25,300
60	16	12,400	13,400	16,300	72	20	21,700	23,300	28,100
60	18	13,600	14,800	18,400	78	18	25,000	26,400	29,600
66	16	14,900	16,300	19,100	78	20	27,100	28,600	32,900
66	18	16,400	17,800	21,500					

umns are not built into the brickwork of the setting, or otherwise braced against lateral flexure.

Table 3.—Suspension Beams and Supporting Columns for Horizontal Return Tubular Boilers

Diam. of Boiler, in.	Length of Tubes, ft.	I-beam Suspension Beams						H-beam Supporting Columns						
		1 Boiler		2 Boilers		3 Boilers		Max. Length, in.	1 Boiler		2 Boilers		3 Boilers	
		Size, in.	Wt., lb.	Size, in.	Wt., lb.	Size, in.	Wt., lb.		Size, in.	Wt., lb.	Size, in.	Wt., lb.	Size, in.	Wt., lb.
54	16	6	12 1/4	10	30	15	42	126	5	18.7	5	18.7	6	23.8
60	16	7	15	12	31 1/2	18	55	132	5	18.7	6	23.8	8	34.0
60	18	7	15	12	35	18	55	132	5	18.7	6	23.8	8	34.0
66	16	7	15	12	40	18	55	144	5	18.7	8	34.0	8	34.0
66	18	8	18	15	42	18	60	144	5	18.7	8	34.0	.....	....
72	16	8	18	15	42	20	65	156	6	23.8	8	34.0	.....	....
72	18	8	18	15	42	20	65	156	6	23.8	8	34.0	.....	....
72	20	8	18	18	55	24	80	156	6	23.8	8	34.0	.....	....
78	16	8	18	18	55	24	80	162	6	23.8	8	34.0	.....	....
78	18	9	21	18	55	24	80	162	6	23.8	8	34.0	.....	....
78	20	9	21	18	55	24	80	162	8	34.0	.....	.....	.....	....
84	18	9	21	18	55	24	80	168	8	34.0	.....	.....	.....	....
84	20	9	21	20	65	24	90	168	8	34.0	.....	.....	.....	....

The I-beam section is not recommended as a boiler column, because its shape is not well adapted to use as a column. Data on built-up plate-and-angle columns and on round and square cast-iron columns are given in the *Boiler Book* of the Hartford Steam Boiler Inspection and Insurance Co., Hartford, Conn.

For other details of construction of horizontal return tubular boilers, including dimensions of boiler tubes, braces, stays, etc., see Boiler Construction, pp. 6–18 to 6–42.

For low-pressure horizontal return tubular boilers, see p. 11–11.

SMALL WATER-TUBE BOILERS.—Fig. 12 shows a small water-tube boiler fired with a chain grate stoker. The smaller sizes of these boilers are shipped after being

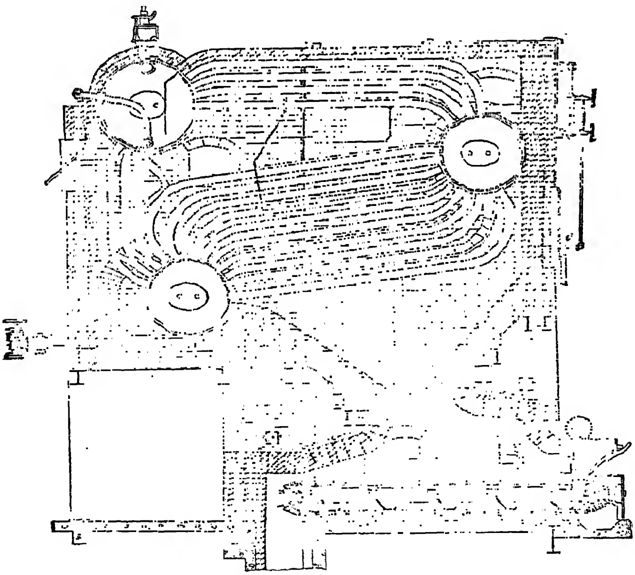


Fig. 12. Stirling Boiler with Babcock & Wilcox Stoker

completely assembled in the shop. The boiler may be provided with a superheater if desired.

BOILERS FOR MISCELLANEOUS SERVICES are discussed elsewhere. See Railroad Engineering, Heating and Ventilation, Marine Engineering.

5. BOILERS WITHOUT NATURAL CIRCULATION

TYPES OF FORCED CIRCULATION BOILERS.—These boilers are of two types: 1. Water and steam flow directly through the boiler and superheater respectively, termed a once-through boiler. 2. More water is fed to the boiler than is evaporated. The mixture of steam and water passes to a chamber where water is separated from the steam before the steam passes through the superheater. In Type 2, water may be more readily treated for preventing corrosion of the interior of boiler tubes than in Type 1. In Type 2 boilers, any ingredient added to prevent corrosion is removed with the excess water which is separated from the steam, while in Type 1 boilers, such ingredients pass from the boiler with the steam and may give trouble by being deposited in the turbine.

OPERATION OF FORCED CIRCULATION BOILERS.—In operating forced circulation boilers, it is most important that a proper control device be employed. Natural circulation boilers have an appreciable storage of water at the saturated temperature corresponding to the pressure, which assists in maintaining steady steam pressure. In a series boiler there is little contained water as compared with a natural circulation boiler. This makes it necessary to use a quickly responsive control system to regulate combustion and feedwater supply. The difficulty of operating a forced circulation boiler at varying loads has hindered its use. This difficulty no longer exists, as a successful automatic control has been developed for use with this type boiler.

A once-through series boiler installed at Purdue University for operating at pressures up to the critical pressure of about 3200 lb. and over is shown in Fig. 13. It has a Bailey automatic control which regulates fuel, feedwater and air supply.

6. PERFORMANCE OF STEAM BOILERS

The performance of a steam boiler comprises both its capacity for generating steam and its economy of fuel. According to the A.S.M.E. Power Test Code the performance of a boiler (including firing equipment) should be expressed in terms of: 1. Efficiency, *i.e.*, ratio of heat units output to high calorific value of the fuel (dry, or as-fired). 2. Fuel (dry, or as-fired), lb. per hr. per sq. ft. of grate surface, per retort, or per burner. 3. Combustion space, cu. ft. per lb. of fuel (dry, or as-fired) per hr. 4. Rate of heat absorption per sq. ft. of steam-generating unit surface per hour, units of evaporation. 5. Heat released, B.t.u. per cu. ft. of furnace volume per hr.

Efficiency usually determined is that of the steam-generating unit, and is the overall efficiency of boiler, superheater, economizer, reheater, air heater, furnace, and fuel-burning equipment. The terms superheater, economizer, reheater, and air heater are omitted if the unit does not include these appurtenances. If air heaters are installed, a comparative efficiency also is calculated. In determining this efficiency, heat absorbed by air heater is not prorated over the steam-generating surface (boiler, superheater, economizer and reheater). Hence, comparative efficiency is lower than overall efficiency by an amount equal to percentage of heat absorbed by air heater (B.t.u. absorbed by air heater per lb. of fuel $\times$ 100 $\div$ High calorific value of fuel, B.t.u. per lb.) In many instances, regardless of the various types of heat-absorbing surfaces installed, efficiency of boiler and furnace, or boiler, superheater and furnace also is determined.

OVERALL EFFICIENCY OF A STEAM-GENERATING UNIT varies principally with type of fuel, methods of firing, and various combinations of heat absorbing surfaces. Overall efficiencies of 85% or more have been achieved in several instances without economizers or air heaters, but for continuous operation such performances cannot be expected. Yearly overall efficiencies of 80% have been reported for large central stations, but the average is lower, probably about 77%. In large isolated or industrial plants, a yearly overall efficiency of 70% is a fair average. Small isolated or industrial plants, in most instances, average considerably less, although a number of such plants fired by pulverized fuel or oil, have reported yearly overall efficiencies of 70% or more.

Several installations consisting of boilers, superheaters, economizers, reheaters, water-cooled furnaces, and air heaters, or some combination thereof, have reported overall test efficiencies of 86–90% when fired by pulverized fuel, mechanically atomized oil, underfeed or chain-grate stokers, and 82–86% when fired by natural gas. Overall test efficiencies as high as 94% have been obtained, and yearly overall efficiencies of 86% are common. A pulverized fuel installation, consisting of boiler, superheater, economizer, and water-cooled furnace reports a monthly efficiency of 88%, while another pulverized fuel installa-

tion, consisting of a high-pressure boiler, superheater, economizer, reheater, water walls and air heater reports a monthly operating efficiency of 89.1%, and a 6-month combined operating efficiency of 87.6%. High monthly, or yearly operating efficiencies depend on load characteristics, type of fuel, and the most modern equipment and operating methods.

EVAPORATION RATES.—A steam-generating unit of total water heating surface of 16,763 sq. ft., consisting of boiler 5938 sq. ft., furnace water walls and floor 2460 sq. ft., and steaming economizer 8365 sq. ft., has operated at a capacity of over 290,000 lb. per hr. This corresponds to an equivalent evaporation, from and at 212° F., of over 368,000 lb. per hr., or 43.9 lb. per sq. ft. of heating surface per hr., or approximately 1275% of rated capacity, when based on heating surface of boiler and furnace water walls and floor. Based on total water heating surface, including economizer, evaporation is 22 lb. per sq. ft. of heating surface per hr., or approximately 640% of rated capacity.

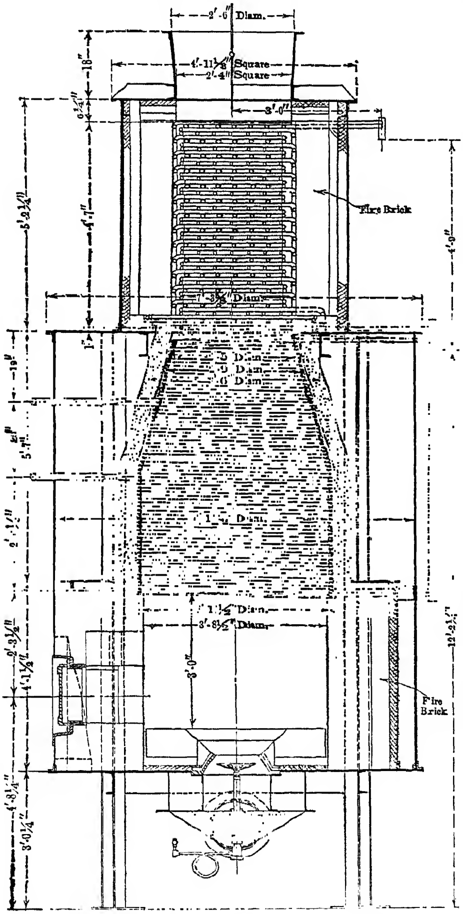


Fig. 13. Continuous Circulation Boiler

Water-tube Boilers.—Maximum rates of evaporation per sq. ft. of heating surface, from and at 212° F., of a horizontal water-tube boiler, absorbing heat mostly through convection, are from 15.1 lb. per hr. for a boiler 18 tubes high, to 22.5 lb. per hr. for a boiler 11 tubes high. Evaporation rates exceeding 20 lb. per sq. ft. of heating surface per hr., from and at 212° F., have been obtained with inclined tube multi-drum boilers.

Horizontal Return Tubular Boilers.—Evaporation rates of 3.5 to 5.5 lb. per hr. from and at 212° F. are most common for horizontal return tubular boilers used in small stationary plants. Locomotive boilers rarely exceed 16 lb. per sq. ft. per hr., but such rates are very uneconomical, because of steam required for creating draft and the attendant loss of coal up the stack.

Marine Boilers.—A standard Babcock & Wilcox marine boiler at the U. S. Naval Oil Testing Plant, Philadelphia, has operated at a maximum evaporation rate of 18.7 lb., from and at 212° F., per sq. ft. of heating surface per hr., while from a Babcock & Wilcox express type marine boiler, installed at the same plant, a maximum evaporation of 22.7 lb., from and at 212° F., per sq. ft. of heating surface per hr. was obtained. Furnace arrangement and volume were those common in marine practice.

HEAT TRANSFER in B.t.u. per hr. per sq. ft. of heating surface differs widely in different boilers and in different sections of the heating surface of any boiler. In the design of boilers actual heat transfer per sq. ft. of each type of heating surface must be known, as well as the variables affecting such transfer. Experiments have shown that the lower row of tubes between the first baffle and the front header of a Babcock & Wilcox boiler, with uncooled furnace, absorbs heat at an average rate per sq. ft. of heating surface

equivalent to the evaporation of from 50–75 lb. of water from and at 212° F. No distinction was made as to the amount of heat absorbed by radiation and by actual contact between hot gases and tubes, but a large portion of the heat absorbed was by direct radiation.

In a Babcock & Wilcox boiler with ordinary clean surfaces, if W = total weight of gases passing per hr., lb., and A = average gas passage area, sq. ft., of all passes of the boiler, average heat transfer U, B.t.u. per sq. ft. per hr. per deg. F. mean temperature difference may be closely approximated by $U = 2 + 0.0014(W/A)$.

A paper by C. F. Hirshfeld and G. U. Moran (*Trans.* A.S.M.E. FSP–52–34) gives a record of performance of 186 large steam-generating units in over 60 power stations in all sections of the U. S., under widely different conditions of operation, and with different classes of fuel. Many data relative to heat release and heat transfer are included.

BOILER CONSTRUCTION

1. SCOPE OF CODES FOR CONSTRUCTION

The Code most largely used for construction of land boilers is the Boiler Construction Code of the American Society of Mechanical Engineers. The Code of Boiler Rules of the Commonwealth of Massachusetts was developed before that of the A.S.M.E. The state of Ohio also developed its own code. These codes are coming closer and closer together, and there is a growing desire to make the fundamental features the same in all.

Rules covering construction of steam boilers for marine service have been issued by U. S. Department of Commerce, Bureau of Navigation and Steamboat Inspection, under the title Amended Rules I and II, General Rules and Regulations, 51st Supplement to General Rules and Regulations, January 1, 1935. Existing codes and regulations were considered in formulating these rules, including the A.S.M.E. Boiler Code. Rules for fusion welded drums or shells of marine boilers and pressure vessels are included, being based on those prepared by a subcommittee of the Committee on Welding in Marine Construction of the Am. Welding Soc., which, in turn, were based on those prepared by the A.S.M.E. Boiler Code Committee in co-operation with the Am. Welding Society.

The A.S.M.E. co-operated with the Am. Petroleum Inst. in preparing a Code for Construction of Unfired Pressure Vessels for Petroleum Liquids and Gases, the first edition of which, designated as A.P.I.–A.S.M.E. Code, was published in 1934.

FACTORS OF SAFETY USED IN VARIOUS CODES.—No term has been misused so widely as the so-called factor of safety. Consider the following three types of vessels, designated as types A, B and C.

Type A. Steam Boilers or Vessels Pierced with Unreinforced Holes.—In these vessels, stress at the edge of holes may be over twice average hoop stress. In applying an initial hydrostatic test of ($1\,{}^1/_2 \times$ working pressure) to a drum built with a so-called factor of safety of 5, the steel at the most highly stressed part will be so strained as to produce a permanent set and redistribution of forces in the stressed area. The factor is based on ultimate strength, and with a steel in which yield strength is $^1/_2$ the ultimate strength, yield strength would be $2\,{}^1/_2$ times average hoop stress at the working pressure. If stress at the sides of tube holes is ($2 \times$ average hoop stress), and the hydrostatic test is made at ($1\,{}^1/_2 \times$ working pressure), stresses at the sides of tube holes would be ($3 \times$ average hoop stress) at the working pressure. This would exceed yield strength of the material. If the so-called factor of safety is made less than 5, as is often done, there is a greater yielding of the material on application of the hydrostatic test and a greater redistribution of stresses.

Type B. Vessels Having No Holes or Other Stress Raisers in Their Shells.—In these vessels, yield strength would not be exceeded until hoop stress, which would be uniform from end to end of cylinder, was exceeded. Such vessels could be operated safely with a much lower so-called factor of safety than those with holes or other stress raisers in the shell. Application of a hydrostatic test of ($1\,{}^1/_2 \times$ working pressure) to a vessel of this type, built with a factor of safety of 5, would not exceed yield strength, as would occur with Type A vessels with unreinforced holes or other stress raisers in their shells.

Type C. Vessels of Type of Construction Used for Penstocks for Boulder Dam.—This construction is described by C. M. Day and Peter Bier of the U. S. Bureau of Reclamation in *Mech. Engg.*, Aug., 1934. The yield strength of the special steel used in the penstocks is 38,000 lb. per sq. in.; hoop stress was limited to 18,000 lb. per sq. in. The

article describes tests made by elastic analysis, and by subjecting 1/6 scale models of the penstocks to hydrostatic pressure. The parts were so built that maximum stress at any point would not exceed about 19,000 lb. per sq. in.

Each of the three types of vessels has its own field of application. Type A includes all pressure vessels having unreinforced holes, or in which unreinforced holes, as telltale holes for determining shell thickness, may be drilled after the vessels are in operation. This type is covered by A.S.M.E. and similar construction codes.

Type B includes vessels constructed under rules of the Interstate Commerce Commission for transporting liquids and gases under pressure. For such vessels, a so-called factor of safety of, say, 3 1/2 where there is no corrosion, will give as great a degree of safety as some Type A vessels where the factor is 5.

Type C represents the highest type of construction for pressure vessels where walls cannot be made of uniform thickness with no stress raisers. By limiting both hoop and maximum stresses, due to departing from uniform shell thickness, as well as at special branch connections and reinforced openings, no question can arise as to the effect of exceeding yield strength in any part of the structure.

THE A.S.M.E. CODE FOR LOCOMOTIVE BOILERS specifies:—

Factor of safety used in design and construction of new boilers shall not be less than 4.5.

Factor of safety used in determining maximum allowable working pressure calculated on conditions actually obtained in service shall not be less than 4.0.

Maximum allowable working pressure determined by conditions obtained in service shall not exceed that for which boiler was designed.

THE JOINT A.P.I.–A.S.M.E. CODE contains provisions for removing vessels from service if the factor of safety becomes lower than certain set figures. At a first glance it would appear that factors of safety given in the A.P.I.–A.S.M.E. Code are lower than those given in the Unfired Pressure Vessel Section of the A.S.M.E. Boiler Code in the ratio of 4 to 5, whereas the factor of 4 applies to a figure below which vessels cannot be operated, and the A.S.M.E. Code refers to initial factor of safety when boilers are built. Another feature in the two codes brings the two factors for stress-relieved unfired pressure vessels nearer together. In the A.P.I.-A.S.M.E. Code, tensile strength used in applying the formulas is that for coupons which are tested at the steel mill, after being stress-relieved in the same manner as the vessel will be stress-relieved; whereas in the A.S.M.E. Code coupons for such vessels are not stress-relieved at the steel mill. In stress-relieving the coupons, which is done by holding them at a temperature of from 1100 to 1200° F., for 1 hour per inch of thickness of plates, tensile strength may be lowered about 10%. The factor 5 in the A.S.M.E. Code was considered ample to cover this lowering of the tensile strength; whereas in the A.P.I.–A.S.M.E. Code it was embodied in the Code on account of the lower operating factor.

RULES FOR CONSTRUCTION.—The matter which follows will be limited to the rules of the A.S.M.E. Boiler Code which, as already stated, includes rules for the construction of unfired pressure vessels.

The A.S.M.E. Boiler Construction Code has received wide recognition and has been enacted into the laws of the following: Arkansas, California, Delaware, Indiana, Maryland, Michigan, Minnesota, Missouri, New Jersey, New York, Ohio, Oklahoma, Oregon, Pennsylvania, Rhode Island, Utah, Washington, Wisconsin, District of Columbia, Hawaii, Canal Zone. It has also been adopted by the cities of: Chicago, Ill.; Detroit, Mich.; Erie, Pa.; Evanston, Ill.; Houston, Tex.; Kansas City, Mo.; Los Angeles, Cal.; Memphis, Tenn.; Nashville, Tenn.; Omaha, Neb.; Parkersburg, W. Va.; Philadelphia, Pa.; Scranton, Pa.; Seattle, Wash.; St. Joseph, Mo.; St. Louis, Mo.; Tampa, Fla.

In the U. S., boilers for merchant vessels must be constructed according to rules and regulations prescribed by the Bureau of Navigation and Steamboat Inspection. Boilers for British merchant vessels must be constructed in accordance with rules of the British Board of Trade.

The following rules are greatly condensed from the A.S.M.E. Boiler Construction Code, edition of 1933, with revisions to 1935. *The Code itself should be consulted in actual design*, as a number of important features have been omitted. When fabricating boilers to be operated in states and municipalities having their own laws and ordinances, reference should be made to these as well.

The following abstract relates to power boilers only. The sections of the code relating to locomotive boilers, low-pressure heating boilers, miniature boilers and unfired pressure vessels are issued as separate publications and reference should be made to them in the design and construction of any apparatus in the foregoing classifications.

Piping systems in connection with steam boilers, especially those conveying high-temperature high-pressure steam, should be constructed in accordance with the American Standard Piping Code. An abstract of this code is given on pp. 5–23 to 5–30.

2. MATERIALS USED IN BOILERS

MATERIAL SPECIFICATIONS (A.S.M.E. Code).—Specifications for certain of the materials used in various parts of boilers are given in Table 1 and the notes appended thereto. See the Code for specifications of the following materials not listed in Table 1.

Steel Plates of Flange and Firebox Qualities for Forge Welding	A–89
Forged or Rolled Steel Pipe Flanges for High-temperature Service	A–105
Alloy Steel Bolting Material for High-temperature Service	A–96
Carbon Steel and Alloy Steel Forgings	A–18
Welded and Seamless Steel Pipe	A–53
Welded Wrought-iron Pipe	A–72
Copper Plates	B–11
Copper Bars for Staybolts	B–12
Seamless Copper Boiler Tubes	B–13
Copper Pipe	B–42
Brass Pipe	B–43
Open-hearth Iron Plate of Flange Quality	A–129

In the above list, and in Table 1, the corresponding A.S.T.M. specification is given when available.

Steel plates for any part of a boiler exposed to fire or products of combustion and under pressure shall be of the quality of steel designated as Firebox steel. Where firebox quality is not specified, when under pressure, steel shall be of firebox or flange quality, as designated in the specifications. Braces when welded shall be of wrought iron of the quality designated Extra Refined Bar Iron. Manhole and handhole covers and other parts subjected to pressure, also braces and lugs made of steel plate, shall be of firebox or flange quality. Steel bars for braces and other boiler parts, unless otherwise specified, shall be of a quality designated Steel Bars. Staybolts are to be of iron or steel, designated as Staybolt Iron or Staybolt Steel. Rivets shall be of steel or iron of the quality designated Boiler Rivet Steel or Boiler Rivet Iron.

Cross pipes connecting steam and water drums of water-tube boilers, headers, cross-boxes and all pressure parts of the boiler proper, over 2 in. pipe size or equivalent cross-sectional area, shall be of Wrought Steel or of Grade B cast steel designated in the specifications for steel castings, when maximum allowable pressure exceeds 160 lb. per sq. in. Malleable iron also may be used if maximum allowable pressure does not exceed 350 lb. per sq. in., if form and size of the internal cross-section, perpendicular to the longest dimension of the box, will fall in a 7 × 7 in. rectangle.

Mud drums of boilers used other than for heating purposes, shall be of wrought steel or Grade B steel casting. Pressure parts of superheaters, separately fired or attached to stationary boilers, unless of the locomotive type, shall be of wrought steel, Grade B cast steel, or puddled or knobbled charcoal wrought iron.

Cast iron shall not be used for nozzles or flanges attached directly to the boiler for any pressure or temperature, nor for boiler or superheater mountings such as connecting pipes, fittings, valves, etc., for steam temperatures of over 450° F., or pressures of over 250 lb. per sq. in.

Water-leg or door-frame rings of vertical fire-tube boilers and of locomotive and other type boilers, shall be of wrought iron or steel or cast steel (Class A or Class B). The O. G. or other flanged construction may be used as a substitute in any case.

Quench Bend Test.—The test specimen when heated to a light cherry red, not less than 1200° F., and quenched at once in water whose temperature is between 80° and 90° F. shall bend through 180° flat on itself without fracture on outside of bent portion. Applicable to boiler rivet steel and staybolt iron, excepting that the iron must be heated to yellow heat.

Notes to Table 1

Note 1.—If formula for minimum elongation gives values of less than 25% for firebox steel, minimum allowable elongation shall be taken as 25%. If the material is over $^3/_4$ in. thick, a deduction from specific elongation of 0.125% is made for each $^1/_{32}$ in. specified above $^3/_4$ in., to a minimum of 20%. For material $^1/_4$ in. or less thick, elongation shall be measured on a gage length equal to (24 × thickness).

Bend Test.—The specimen to bend cold through 180° without cracking on outside of bent portion as follows: Material 1 in. or less thick, around a pin of diameter equal to thickness of specimen. Over 1 in., around a pin of diameter (2 × thickness of specimen).

Permissible variation in gage under that specified, not over 0.01 in. A homogeneity test is required for firebox steel. *Process.*—Steel shall be made by open-hearth or electric furnace, or both.

Note 2.—Elongation is measured on gage length of (24 × thickness of specimen), with reduction of 2.5% for each $^1/_{16}$ in. under $^5/_{16}$ in. Specimen shall bend cold through 180° without cracking on outside of bent portion, around pin of diameter equal to thickness of specimen.

Note 3.—*Process.*—Either open-hearth or electric furnace, or both. Forgings are annealed above critical temperature.

Table 1. Specifications for Boiler Material
(Boiler Code, A.S.M.E.)

Material	A.S.T.M. Specification	Chemical Composition, percent				Physical Properties				See Note No.
		C	Mn	P max.	S max.	Tensile Strength, lb. per sq. in.	Yield Point, lb. per sq. in.	Elongation in 8 in. (min.), percent	Reduction of Area, percent	
Flange steel	A-70	...	0.30–0.60	0.04[a]	0.05	55,000–65,000	0.5 T.S.	1,500,000/T.S.	...	1
Firebox steel	A-70									
Plates 3/4 in. and under	...	0.25	0.30–0.50	0.035[b]	0.04	55,000–65,000	0.5 T.S.	1,550,000/T.S.	...	1
Plates over 3/4 in.	...	0.30	0.30–0.60	0.035[b]	0.04	55,000–65,000	0.5 T.S.	1,550,000/T.S.	...	1
Steel plate for brazing	...	0.28	0.60	0.04	0.05	70,000	28,000	1,500,000/T.S.	...	2
Seamless steel drum forgings:										
Class 1	...	0.35	0.40–0.70	0.035[a]	0.05	60,000	0.5 T.S.	26[e]	42	3
Class 2	...	0.50	0.40–0.70	0.035[a]	0.05	75,000	0.5 T.S.	24[e]	38	3
Rivet steel	A-31	...	0.30–0.50	0.04	0.045	45,000–55,000	0.5 T.S.	1,500,000/T.S.	...	4
Staybolt steel	...	...	0.30–0.60	0.04	0.045	60,000	0.5 T.S.	1,500,000/T.S.	...	5
Steel bars	A-7	...	...	0.04[c]	0.05	55,000–65,000	0.5 T.S.	1,500,000/T.S.	...	6
Steel castings:										
Class A	A-27	0.45[d]	...	0.06[g]	...	...	...	...	...	7
Class B	A-87	...	...	0.05	0.05	...	29,250	24[e]	35	7
Carbon steel castings for high-temperature service for valves, flanges and fittings	A-95	0.15–0.45	0.50[f]	0.05	0.06	70,000	36,000	22[e]	30	8
Gray iron castings:										
Light	...	...	...	...	0.10	18,000	...	...	...	9
Medium	...	...	...	...	0.10	21,000	...	...	...	9
Heavy	...	...	...	...	0.10	24,000	...	...	...	9
Malleable castings	A-47	...	...	...	...	50,000	32,500	10.0[e]	...	10
Boiler rivet iron	A-84									
Section 1 1/4 sq. in. and under	...	...	...	...	...	48,000–52,000	0.6 T.S.	28	45	11
Section over 1 1/4 to 4 sq. in. incl.	...	...	...	...	...	48,000–52,000	0.55 T.S.	28	45	11
Staybolt iron	A-84									
Section 1 1/4 sq. in. and under	...	...	...	...	...	48,000–52,000	0.6 T.S.	30	48	12
Section over 1 1/4 to 4 sq. in. incl.	...	...	...	...	...	48,000–52,000	0.55 T.S.	30	48	12
Extra refined bar iron	A-84									
Section 1 1/4 sq. in. and under	...	...	...	...	...	48,000–53,000	0.6 T.S.	25	35	13
Section over 1 1/4 to 4 sq. in. incl.	...	...	...	...	...	48,000–53,000	0.55 T.S.	25	35	13
Boiler tubes	A-83									
Grade A, Low carbon steel	...	0.08–0.18	0.30–0.60	0.04	0.045	...	...	...	...	14
Grade B, Open-hearth iron	...	0.03[d]	0.03[d]	0.02	0.045	...	...	...	...	14
Grade C, Medium carbon steel	...	0.35[d]	0.80[d]	0.04	0.045	60,000	37,000	25[e]	...	14
High tensile carbon steel plates for pressure vessels	A-149									
Flange Grade A, 3/4 in. and under	...	0.32	0.9	0.04	0.05	65,000–77,000	0.5 T.S.	1,550,000/T.S.	...	15
Flange Grade A, over 3/4 to 2 in., incl.	...	0.35	0.9	0.04	0.05	65,000–77,000	0.5 T.S.	1,550,000/T.S.	...	15

See end of table for footnotes.

(Table continued on following page.)

Table 1. Specifications for Boiler Material—*Continued*

Material		Chemical Composition, percent	Mn			Physical Properties	eld Point, lb. per sq. in	Elongation 8 in. (min per cent	ion per
Grade B, 3/4 in. and under.		0.32	0.9	0.04	0.05	70,000–82,000	0.5 T.S.	1,550,000 / T.S.	
Flange Grade B, over 3/4 to 2 in., incl.		0.35	0.9	0.04	0.05	70,000–82,000	0.5 T.S.	1,550,000 / T.S.	
Firebox Grade A, 3/4 in. and under.		0.32	0.9	0.035	0.04	65,000–77,000	0.5 T.S.	1,600,000 / T.S.	
Firebox Grade A, over 3/4 to 2 in., incl.		0.35	0.9	0.035	0.04	65,000–77,000	0.5 T.S.	1,600,000 / T.S.	
Firebox Grade B, 3/4 in. and		0.32	0.9	0.035	0.04	70,000–82,000	0.5 T.S.	1,600,000 / T.S.	
Firebox Grade B, over 3/4 to 2 in., incl.		0.35	0.9	0.035	0.04	70,000–82,000	0.5 T.S.	1,600,000 / T.S.	
High tensile carbon steel plates for fusion welded pressure vessels.	A–150								
Firebox Grade A.		0.35	0.5–0.9	0.035[c]	0.04	65,000–70,000	0.5 T.S.	1,750,000[e] / T.S.	
Firebox Grade]		0.35	0.5–0.9	0.035[c]	0.04	70,000–82,000	0.5 T.S.	1,750,000[e] / T.S.	

[a] Basic steel; acid steel, P = 0.05. [b] Basic steel; acid steel, P = 0.04. [c] Basic steel; acid steel, P = 0.06. [d] Maximum. [e] Elongation in 2 in. [f] Minimum. [g] Basic steel; acid steel, P = 0.07

Bend Test.—Specimen shall bend cold through 180° without cracking on outside of bent portion, around 1-in. pin for Class 1, and around 1 1/2-in. pin for Class 2.

Note 4.—*Process.*—Open hearth or electric furnace, or both.

Bend Test.—Specimen shall bend cold through 180° flat on itself without cracking on outside of bent portion. Quench bend test also required. (See p. 6-18). Shank of rivet shall bend cold through 180° flat on itself without cracking on outside of bent portion. Rivet head shall flatten while hot to diameter of (2 1/2 × diameter of shank) without cracking at edges.

Permissible Variation in Diameter.—±0.007 in. for bars up to 1/2 in. inclusive; ±0.01 in. for over 1/2 in. to 1 in., inclusive; ±0.012 in. for over 1 in. to 1 1/4 in., inclusive; ±0.015 in. for over 1 1/4 to 1 1/2 in., inclusive; ±0.022 in. for over 1 1/2 to 2 in., inclusive.

Note 5.—Open-hearth or electric furnace, or both. Permissible variation in gage not more than 0.005 in. under nor 0.01 in. above specified size. Bars shall be truly round to within 0.01 in. Same tests required as for rivet steel.

Note 6.—Open-hearth or electric furnace, or both. Minimum elongation in 2 in., 26%. For material over 3/4 in. thick, deduct 0.125% for each increase of 1/32 in. specified above 3/4 in.

Bend Test.—Specimen shall bend cold through 180° without cracking on outside of bent portion as follows: material of diameter or thickness 1 in. or less, flat on itself; material over 1 in. to 1 1/2 in. inclusive, around a pin of diameter equal to thickness of test specimen; material over 1 1/2 in. around a pin of diameter equal to (2 × thickness of test specimen).

Permissible Variation in Diameter.—±0.007 in. for diameters up to 1/2 in. inclusive; ±0.010 in. for over 1/2 to 1 in. inclusive; ±0.012 in. for over 1 in. to 1 1/4 in. inclusive; ±0.015 in. for over 1 1/4 to 1 1/2 in. inclusive; ±0.022 in. for over 1 1/2 to 2 in. inclusive; +3/64 to −1/64 in. for over 2 to 2 1/2 in. inclusive; +3/64 to −1/32 in. for over 2 1/2 to 3 in. inclusive.

Note 7.—Open-hearth, electric furnace, converter, or crucible. Class A castings need not be annealed unless so specified, or unless carbon content exceeds 0.3%. Class B castings shall be properly annealed, treatment depending on design and chemical composition of castings.

Note 8.—Silicon not under 0.2%. Electric furnace, open-hearth or other process approved by purchaser. All castings shall be heat treated, either annealed or normalized and never quenched. Cold bend test required when specified. Hydrostatic test required. The specification contemplates temperature up to 750° F.

Note **9.**—Air furnace, open-hearth or electric furnace. Castings with any section less than 1/2 in. thick are light castings. If no section is less than 2 in. thick, castings are heavy castings. Medium castings are those not included in foregoing classifications. Transverse tests, made on *arbitration bar* placed horizontally on supports 18 in. apart and tested under a centrally applied load, shall show following minimum breaking loads: Light castings, 1500 lb.; medium, 1750 lb.; heavy, 2000 lb. Deflection at center never less than 0.2 in. The arbitration bar is 1.20 in. diam., 23 in. long, cast in a thoroughly dry and cold sand mold, or in a core. The standard tensile test piece is 0.8 in. diam., 1 in. long between shoulders, with a fillet 7/8 in. radius, and ends to fit the holders of the testing machine.

Note 10.—Air furnace, open-hearth, or electric furnace. Castings must be sufficiently annealed.

Note 11.—Rolled from a bloom, slab pile, or box pile, made from reworked wrought or knobbled charcoal iron, free from iron scrap or steel.

Bend Test.—Specimen to bend cold through 180° flat on itself without fracture on outside of bent portion. Specimen when nicked 25% around for round bars or along one side for flat bars, with a tool having a 60° cutting edge, to a depth between 8 and 16% of diameter or thickness of specimen and broken by pressure or blows, shall show a wholly fibrous structure.

Maximum Variation in Gage.—0.01 in. for 1/4 in. nominal diameter, increasing by 0.001 for each 1/16 in. up to 5/8 in., and for each 1/8 in. from 5/8 in. to 1 7/8 in. diameter where tolerance is 0.026 in. Etch test required to show material to have been rolled from a bloom, slab pile or box pile, uniform and free from steel.

Note 12.—See Note 11; material must be twice worked, that is twice piled. Specimen to bend cold through 180° flat on itself in both directions without fracture on outside of bent portion. See Note 11 for nick bend test and etch test required. Bars must be truly round within 0.01 in and shall not be less than 0.005 in. over nor more than 0.02 in. over specified size.

Note 13.—Made from wrought iron free from iron scrap or steel.

Bend Test.—Specimen to bend cold through 180° around pin of diameter equal to diameter or thickness of specimen, without fracture on outside of bent portion. Specimen when heated between 1700 and 1800° F. to bend through 180° flat on itself without fracture on outside of bent portion. See Note 11 for nick bend test and maximum variation in gage. Etch test, structure to be uniform and free from steel.

Note 14.—Lap-welded tubes shall be made either by open-hearth or electric furnace, or both, or the knobbled, hammered charcoal iron process. Seamless tubes shall be made by either the open-hearth or electric furnace or both. Grade C tubes are seamless.

Flange Test.—For tubes not over 6 in. diameter, of thickness less than 10% of O.D., and not exceeding No. 6 B.W.G., a test specimen not less than 4 in. long shall have a flange turned over at right angles to the body of tube without cracking or showing any flaw. Flange to have a width of 1/8 to 1/2 in. measured from outside of tube. Width between these limits to be as follows:

O.D. of tube, in	Charcoal Iron	Open Hearth or Electric Furnace: Grades A and B	Open Hearth or Electric Furnace: Grade C
Up to 2 1/2 in., inclusive	12.5% of O.D.	15% of O.D.	75% of that required for Grades A and B
Over 2 1/2 in. to 3 3/4 in., inclusive	3/8 in.	3/8 in.	
Over 3 3/4 in. to 6 in., inclusive	10% of O.D.	10% of O.D.	

Flange test not required for other tubes.

Flattening Test.—For tubes except Grade C, of wall thickness not over 10% of O.D. and not exceeding No. 6 B.W.G., a section 2 1/2 in. long shall stand flattening between parallel plates until distance between plates is (3 × wall thickness) for seamless tubes, or (5 × wall thickness) for lap-welded tubes, without cracking or showing any flaw. For tubes except Grade C other than above, a 2 1/2 in. long section shall stand flattening until distance between parallel plates is (4 × wall thickness) for seamless and (6 × wall thickness) for lap-welded tubes. Lap-welded, charcoal iron tubes thicker than No. 6 B.W.G. shall be tested with weld 45° from point of maximum bend. Other charcoal iron, and all steel, lap-welded tubes, shall be tested with weld at point of maximum bend. A section 2 1/2 in. long of Grade C tubes shall stand flattening between parallel plates without cracking or showing any flaw until distance between plates is 1/2 O.D. but never less than (7 × wall thickness).

Hydrostatic Test.—Tubes under 5 in. diam. must stand an internal hydrostatic pressure of 1000 lb. per sq. in. and tubes 5 in. diam. and over, 800 lb. per sq. in., providing fiber stress corresponding to these pressures does not exceed 16,000 lb. per sq. in. for Grades A and B, and 18,000 lb. per sq. in. for Grade C. If fiber stress exceeds above values, test pressure P shall be determined by $P = (32{,}000\ t/D)$ for Grades A and B, and $P = (36{,}000\ t/D)$ for Grade C. t = tube thickness, in.; D = outside diameter, in. Lap-welded tubes shall be struck with a 2-lb. steel hammer near both ends while under test pressure. Permissible variations in wall thickness is zero under, and 2 B.W.G. over, for cold-drawn tubes or 3 B.W.G. over for hot finished tubes.

Note 15.—S = 0.25%. Open-hearth or electric furnace, or both. Specification covers plates 2 in. thick and under. Plates over 1 in. thick shall be heat treated.

Tensile Test.—Plates over 3/4 in. have elongation deduction of 0.125%, for each increase of 1/32 in., to minimum of 18% for flange, and 19% for firebox quality.

Bend Test.—Specimen shall bend cold through 180° without cracking on outside of bent portion when bent around a pin. Where t = thickness of specimen, diameter of pin to be:

	Up to 1 in. thick	1 1/2 in. thick	1 1/2 to 2 in. thick
Grade A, pin diameter	$1.5t$	$2t$	$2t$
Grade B, pin diameter	$2t$	$2t$	$2.5t$

Homogeneity test required for firebox steel only. Maximum variation in thickness = 0.01 in. under specified.

Note 16.—Si = 0.25% maximum. Open-hearth or electric furnace, or both. Specifications cover plates over 2 in. to 4 in. inclusive. Heat treatment required.

Tensile Test.—Plates over 2 1/2 in. thick have elongation deduction of 0.5% for each increase of 1/2 in., to minimum of 22% for Grade A, and 20% for Grade B.

Bend Test.—Specimen shall bend cold through 180° without cracking on outside of bent portion when bent around a pin of diameter (2 × thickness) for Grade A, and (2 1/2 × thickness) for Grade B specimens up to 3 in. inclusive, and (2 1/2 × thickness) for Grade A, and (3 × thickness) for Grade B specimens over 3 in. to 4 in. inclusive. Homogeneity test required.

3. WORKING PRESSURES

MAXIMUM ALLOWABLE WORKING PRESSURE (A.S.M.E. Code).—Maximum allowable working pressure on the shell of a boiler or drum, for temperatures not to exceed 700° F., is determined by the strength of the weakest course, computed as follows:

where P = maximum allowable working pressure, lb. per sq. in.; T = ultimate tensile strength stamped on shell plates by manufacturer, lb. per sq. in.; t = minimum thickness of shell plates in weakest course, in.; e = efficiency of longitudinal joint or ligaments between tube holes (whichever is least); R = inside radius of weakest course of shell or drum, in., if thickness of shell is less than 10% of radius, or R = outside radius if thickness is greater than 10% of radius; F = factor of safety or ratio of ultimate strength to allowed stress. For new constructions, F = 5. For temperatures exceeding 700° F., allowable stresses replacing $T \div F$ in formula [1] are given in Table 2.

Table 2.—Working Stresses of Plain Carbon Steels at Various Temperatures

Maximum Temperature, deg. F.	Minimum of specified range of tensile strength of material, lb. per sq. in.				
	45,000	50,000	55,000	60,000	75,000
	Allowable Working Stress, lb. per sq. in.				
700	9,000	10,000	11,000	12,000	15,000
750	8,220	9,110	10,000	11,200	13,000
800	6,550	7,330	8,000	9,000	10,200
850	5,440	6,050	6,750	7,400	8,300
900	4,330	4,830	5,500	5,600	6,000
950	3,200	3,600	4,000	4,000	4,000

Table 3, from Boiler Book of Hartford Steam Boiler Inspection and Insurance Co., gives the allowable pressures on boiler shells with joints of efficiencies as shown on page 6–26.

PRESSURES ALLOWED ON OLD BOILERS. (Suggested Rules Covering Existing Installations. Appendix to A.S.M.E. Code, 1933 Edition.)—Maximum allowable working pressure on shell of a boiler or drum shall be determined by formula [1], with factors of safety as given below. Boilers in service one year after these rules become effective shall be operated with a factor of safety of at least 4. Five years after these rules become effective, factor of safety shall be at least 4.5. In no case shall maximum allowable working pressure on old boilers be increased, unless they are being operated at a lesser pressure than would be allowable for new boilers, in which case the changed pressure shall not exceed that allowable for new boilers of the same construction.

The age limit of a horizontal-return tubular boiler having a longitudinal lap joint and carrying over 50 lb. per sq. in. pressure shall be 20 years, except that no lap-joint boiler shall be discontinued from service solely on account of age until 5 years after these rules become effective.

Second-hand boilers, *i.e.*, boilers where both ownership and location are changed, shall have a factor of safety of at least 5 1/2 one year after these rules become effective, unless constructed in accordance with the rules contained in the Power Boiler Section, when factor shall be at least 5.

Maximum allowable working pressure on a water-tube boiler, tubes of which are secured to cast-iron or malleable-iron headers, or which have cast-iron mud drums, shall not exceed 160 lb. per sq. in.

Maximum allowable working pressure shall not exceed 15 lb. per sq. in. on a boiler used exclusively for low-pressure steam heating.

The shell or drum of a boiler in which a typical "lap seam crack" is discovered along a longitudinal riveted joint for either butt seam or lap joint shall be permanently discontinued for use under steam pressure. By "lap seam crack" is meant the typical crack frequently found in lap seams extending parallel to the longitudinal joint and located either between or adjacent to rivet holes.

DRUM DESIGN.—The maximum diameter of boiler drums that can be built economically, particularly for bent-tube boilers with inherently low ligament efficiencies, limits increases in steam pressure. The use of steel of 70,000 lb. per sq. in. tensile strength for fusion welding, as permitted by the 1933 A.S.M.E. Code, allows pressure to be increased with a given drum. The code specification covers plate up to 4 in. thick, but no limit is placed on the thickness that may be fusion welded.

Two-plate construction is the most economical for fusion welded drums for bent-tube boilers. A thick plate is used for the tube sheet, with low ligament efficiency, and a

Diameter of Shell, Inches

Plate Thickness, In.	24 Double Riveted	30 Double Riveted	36 Double Riveted	36 Triple Riveted	42 Double Riveted	42 Triple Riveted	48 Double Riveted	48 Triple Riveted	54 Double Riveted	54 Triple Riveted	60 Double Riveted	60 Triple Riveted	60 Quadruple Riveted	66 Double Riveted	66 Triple Riveted	66 Quadruple Riveted	72 Double Riveted	72 Triple Riveted	72 Quadruple Riveted	78 Triple Riveted	78 Quadruple Riveted	84 Triple Riveted	84 Quadruple Riveted	90 Quadruple Riveted	96 Quadruple Riveted
	Pressure, lb. per sq. in.																								
1/4	189.7	151.8	126.5	133.6	108.4	114.5	94.8	100.2	84.3	89.1	75.8	80.2	85.9	68.9	72.9	78.1	63.2	66.8	71.6	61.6	66.1	57.2	61.4	57.3	53.7
9/32	213.4	170.7	142.3	150.3	121.9	128.9	106.7	112.7	94.8	100.2	85.3	90.2	96.7	77.6	82.0	87.9	71.1	75.1	80.6	69.4	74.4	64.4	69.0	64.4	60.4
5/16	234.6	187.6	156.4	167.1	134.0	143.2	117.3	125.3	104.2	111.4	93.8	100.2	107.4	85.3	91.1	97.7	78.2	83.5	89.5	77.1	82.6	71.6	76.7	71.6	67.1
11/32	258.0	206.4	172.0	183.8	147.4	157.5	129.0	137.8	114.6	122.5	103.2	110.2	118.2	93.8	100.2	107.4	86.0	91.9	98.5	84.8	90.9	78.7	84.4	78.8	73.8
3/8	281.5	225.2	187.6	202.5	160.8	173.6	140.7	151.9	125.1	135.0	112.6	121.5	129.5	102.3	110.5	117.7	93.8	101.2	107.9	93.5	99.6	86.8	92.5	86.3	80.9
13/32	304.9	243.9	203.3	219.4	174.2	188.1	152.4	164.5	135.5	146.2	121.9	131.6	140.3	110.9	119.7	127.5	101.6	109.7	116.9	101.2	107.9	94.0	100.2	93.5	87.6
7/16	326.0	260.8	217.3	235.0	186.3	201.4	163.0	176.2	144.9	156.6	130.4	141.0	150.7	118.5	128.1	137.0	108.6	117.5	125.5	108.4	115.9	100.7	107.7	100.5	94.2
15/32	349.3	279.4	232.8	251.7	199.6	215.8	174.6	188.8	155.2	167.8	139.7	151.0	161.5	127.0	137.3	146.8	116.4	125.8	134.6	116.2	124.2	107.9	115.3	107.7	100.9
1/2	372.6	298.1	248.4	269.8	212.9	231.2	186.3	202.3	165.6	179.8	149.0	161.8	172.5	135.5	147.1	156.8	124.2	134.9	143.7	124.5	132.7	115.6	123.2	115.0	107.8
17/32	....	....	....	286.6	....	245.7	....	215.0	....	191.1	....	172.0	183.2	....	156.3	166.6	....	143.3	152.7	132.2	140.9	122.8	130.9	122.1	114.5
9/16	....	....	....	298.0	....	255.4	....	223.5	....	198.6	....	178.8	192.6	....	162.5	175.1	....	149.0	160.5	137.5	148.1	127.7	137.5	128.4	120.3
19/32	....	....	....	314.5	....	269.6	....	235.9	....	209.7	....	188.7	203.3	....	171.6	184.8	....	157.2	169.4	145.1	156.4	134.8	145.2	135.6	127.1
5/8	....	....	....	331.1	....	283.8	....	248.3	....	220.7	....	198.6	214.0	....	180.6	194.5	....	165.5	178.3	152.8	164.6	141.9	152.8	142.6	133.7
21/32	....	....	....	347.7	....	298.0	....	260.7	....	231.8	....	208.6	224.7	....	189.6	204.3	....	173.8	187.2	160.4	172.8	149.0	160.5	149.8	140.4
11/16	....	....	....	359.6	....	308.2	....	269.7	....	239.7	....	215.7	233.9	....	196.1	212.6	....	179.8	194.9	165.9	179.9	154.1	167.0	155.9	146.2
23/32	....	....	....	375.9	....	322.2	....	281.9	....	250.6	....	225.5	244.5	....	205.0	222.3	....	187.9	203.8	173.4	188.1	161.1	174.6	163.0	152.8
3/4	....	....	....	391.8	....	335.8	....	293.9	....	261.2	....	235.1	254.9	....	213.7	231.7	....	195.9	212.4	180.8	196.0	167.9	182.0	169.9	159.3
25/32	....	....	....	403.9	....	346.2	....	302.9	....	269.2	....	242.3	264.3	....	220.3	240.3	....	201.9	220.3	186.4	203.3	173.1	188.8	176.2	165.2
13/16	....	....	....	420.0	....	360.0	....	315.0	....	280.0	....	252.0	274.9	....	229.1	249.9	....	210.0	229.1	193.8	211.5	180.0	196.4	183.3	171.8
27/32	....	....	....	434.1	....	372.1	....	325.5	....	289.4	....	260.4	284.0	....	236.7	258.1	....	217.0	236.6	200.3	218.4	186.0	202.8	189.3	177.5
7/8	....	....	....	449.7	....	385.4	....	337.2	....	299.8	....	269.8	292.6	....	245.2	266.0	....	224.8	243.8	207.5	225.0	192.7	209.0	195.0	182.8
29/32	....	....	....	462.9	....	396.8	....	347.2	....	308.6	....	277.7	300.7	....	252.5	273.3	....	231.4	250.6	213.6	231.3	198.4	214.8	200.4	187.9
15/16	....	....	....	479.5	....	411.0	....	359.6	....	319.6	....	287.7	309.7	....	261.5	281.5	....	239.7	258.0	221.3	238.2	205.5	221.2	206.4	193.5
31/32	....	....	....	492.5	....	422.1	....	369.4	....	328.3	....	295.5	317.9	....	268.6	289.0	....	246.2	264.9	227.3	244.5	211.0	227.0	211.9	198.6
1	....	....	....	509.6	....	436.8	....	382.2	....	339.7	....	305.8	330.7	....	278.0	300.6	....	254.8	275.6	235.2	254.4	218.4	236.2	220.4	206.7
1 1/32	....	....	....	....	....	....	....	....	....	....	....	....	338.8	....	....	308.0	....	....	282.3	....	260.6	....	242.0	225.8	211.7
1 1/16	....	....	....	....	....	....	....	....	....	....	....	....	346.7	....	....	315.2	....	....	288.9	....	266.7	....	247.6	231.1	216.7
1 3/32	....	....	....	....	....	....	....	....	....	....	....	....	354.9	....	....	322.6	....	....	295.7	....	273.0	....	253.5	236.6	221.8
1 1/8	....	....	....	....	....	....	....	....	....	....	....	....	363.0	....	....	330.0	....	....	302.5	....	279.2	....	259.2	242.0	226.8
1 5/32	....	....	....	....	....	....	....	....	....	....	....	....	370.9	....	....	337.2	....	....	309.1	....	285.3	....	264.9	247.3	231.8
1 3/16	....	....	....	....	....	....	....	....	....	....	....	....	381.8	....	....	347.1	....	....	318.2	....	293.7	....	272.7	254.5	238.6
1 7/32	....	....	....	....	....	....	....	....	....	....	....	....	389.6	....	....	354.2	....	....	324.7	....	299.7	....	278.3	259.7	243.5
1 1/4	....	....	....	....	....	....	....	....	....	....	....	....	397.8	....	....	361.6	....	....	331.5	....	306.0	....	284.1	265.2	248.6

thin plate, with high ligament efficiency is used for the wrapper. Ligament efficiency for bent-tube boilers with 3 1/2-in. tubes, of average spacing of 6 in. along the drum, is approximately 0.453; for sectional header cross-drum boilers it ranges from 0.65 to 0.75 or more, depending on the spacing of circulators. Fig. 1 gives limits of steam pressure for fusion-welded drums of 4-in. plate of 70,000 lb. tensile strength. Actual inside and outside diameters, corresponding to standard head dimensions are:

Nominal diam., in.	36	42	48	54	60	72
Actual inside diam., in.	32 5/8	38 5/8	44 3/8	50 7/8	57	69
Actual outside diam., in.	40 5/8	46 5/8	52 3/8	58 7/8	65	77

Forged drums will be of uniform thickness around the circumference, and with forged heads will weigh more than the thick-and-thin plate welded drums with welded heads, but may be made of steel of 75,000 per sq. in. tensile strength. Fig. 2 shows limiting steam pressures for forged seamless drums of 5-in. plate, 75,000-lb. tensile strength, of the following dimensions:

Nominal diam., in.	36	42	48	54	60	72
Actual inside diam., in.	31	37	43	49	55	67
Actual outside diam., in.	41	47	53	59	65	77

Fig. 3 gives longitudinal tube spacing necessary for the ligament efficiencies shown in Figs. 1 and 2.

Temperature stress on the inner surface of drums is approximately (110 × temperature

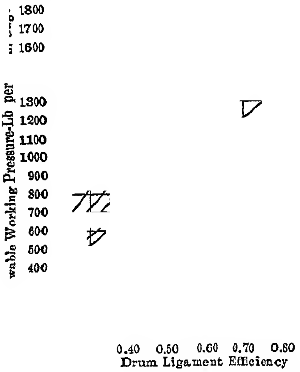


Fig. 1. Maximum Allowable Working Pressures of Fusion Welded Drums with Thick and Thin Plates

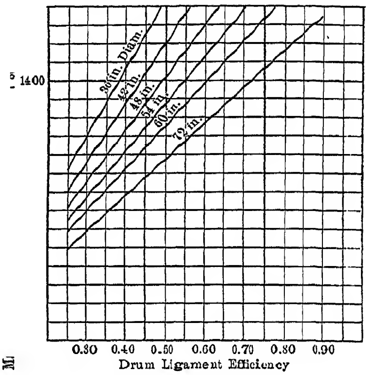


Fig. 2. Maximum Allowable Working Pressures of Forged Steel Drums

drop). Maximum allowable temperature drop T_m through the drum shell, therefore, is $T_m = (S_t - S_a)/110$, where S_t and S_a = respectively, actual and allowable stresses in the drum. Thus, in a drum of 70,000 lb. per sq. in. tensile strength, the permissible additional stress in that portion not pierced with tube holes may be 7600 lb. per sq. in., corresponding to a temperature drop of 69° F. These conditions would require that rate of heat absorption of drums over 2 in. thick be kept below 10,000 B.t.u. per sq. ft. per hr. See Fig. 4.

THICKNESS OF PLATES (A.S.M.E. Code).—The thickness of plate for any given working pressure may be computed by

$$t = (P \times R \times F) \div (T \times e), \qquad [2]$$

where e = efficiency of joint. Minimum thickness of any boiler plate under pressure shall be 1/4 in. Minimum thickness of plates in stayed surface construction shall be 5/16 in.

Minimum thickness of shell plates and dome plates after flanging to be as follows:

Diameter of shell, in.	Up to 36	36–54	54–72	Over 72
Thickness, in.	1/4	5/16	3/8	1/2

Minimum thickness of butt straps for double-strap joints to be as follows, intermediate values being found by interpolation:

Plates, in.	1/4 to 11/32	3/8 to 13/32	7/16 to 15/32	1/2 to 9/16	5/8 to 3/4	7/8	1	1 1/8	1 1/4	1 1/2
Straps, in.	1/4	5/16	3/8	7/16	1/2	5/8	11/16	3/4	7/8	1

For plate thicknesses over $1\,^1/_2$ in., butt-straps shall be at least $^2/_3$ as thick as the plate. Minimum thickness of tube sheets for horizontal return tubular boilers to be as follows:

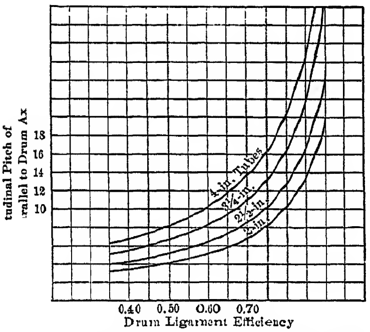
Diameter of tube sheet, in......	Up to 42	42–54	54–72	Over 72
Thickness, in...............	$^3/_8$	$^7/_{16}$	$^1/_2$	$^9/_{16}$

The resistance to crushing of steel plate is to be taken at 95,000 lb. per sq. in. of cross-sectional area.

HYDROSTATIC TESTS (A.S.M.E. Code).—Completed boilers to be subjected to hydrostatic test of $1\,^1/_2 \times$ maximum allowable pressure. Pressure to be so controlled that required test pressure will not be exceeded by more than 6%. Safety valves are to be removed, or each valve held to its seat by a clamp during test. Where it is impossible to calculate with a reasonable degree of accuracy the strength of a boiler structure, or any part thereof, a full size sample may be tested in accordance with a prescribed method. The maximum allowable working pressure of a structure by this method may be determined by

$$P = (HS/5E), \qquad [3]$$

where P = maximum allowable working pressure, lb. per sq. in.; H = hydrostatic pressure at proportional limit; S = average tensile strength of the material, lb. per sq. in.; E = average proportional limit of the material. S and E are determined from specimens cut from weakest sections, with axes parallel to direction of greatest stress. If specimens cannot be obtained, alternate method is to assume $E = 0.4\,S$ which gives $P = (H \div 2)$.



3. Longitudinal Pitch of Boiler Tubes for Various Ligament Efficiencies

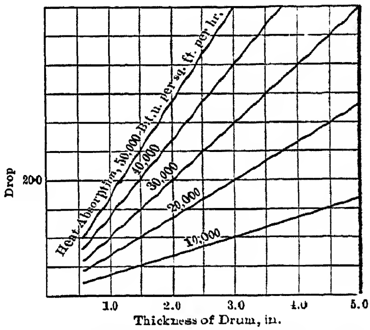


FIG. 4. Temperature Drop through Drums

4. BOILER JOINTS

LONGITUDINAL JOINTS (A.S.M.E. Code).—The distance between center lines of any two adjacent rows of rivets, or back pitch, measured at right angles to the joint should have the following minimum values: If P/D is 4 or less, minimum value is $2D$; if P/D is over 4, minimum value is $2D + 0.1\,(P - 4D)$, where P = pitch of rivets, in., in outer row where a rivet of inner row comes midway between two rivets in the outer row; or pitch of rivets in outer row less pitch of rivets in inner row, where two rivets in inner row come between two rivets in outer row (it is assumed that joints are of the usual construction, and rivets symmetrically spaced); D = diameter of rivet holes, in. On longitudinal joints, distance from centers of rivet holes to edges of plates, except rivet holes in the ends of butt straps, should be not less than $1.5D$ nor more than $1.75D$, measured from center of rivet hole to calking edge of plate before calking. Edge of plate is to be beveled to an angle not sharper than 70° to the plane of the plate, and as near thereto as practicable. Calking is to be done with a tool of such form that there is no danger of scoring or damaging the plate underneath the calking edge or of splitting the calking sheet.

Riveted longitudinal joints of a shell or drum exceeding 36 in. diameter shall be of butt and double-strap construction. This rule does not apply to that portion of boiler shell which is staybolted to the firebox sheet. Longitudinal joints of shell or drum less than 36 in. diam., may be lap-riveted, if the maximum allowable pressure does not exceed 100 lb. per sq. in. The longitudinal joints of horizontal return tubular boilers shall be

located above the fire line of the setting. In horizontal return tubular boilers with lap joints, no course shall be over 12 ft. long. In butt and double-strap construction, longitudinal joints of any length may be used. Butt straps and ends of shell plates forming longitudinal joints shall be rolled or formed by pressure, not blows, to proper curvature.

If shell plates in horizontal return tubular boilers are over $^5/_8$ in. thick, the portion of the plates forming the laps of circumferential seams exposed to fire or products of combustion shall be planed or milled as in Fig. 5, providing the requirements for strength of circumferential joints are complied with (see below).

CIRCUMFERENTIAL JOINTS.—The strength of circumferential joints in boilers whose heads are not stayed by tubes or through braces shall be at least 50% of that required for the longitudinal joints of the same structure. If tubes or stays relieve 50% or more of the load which would act on an unstayed solid head, the strength of the circumferential joints in the shell shall be at least 35% of that required for longitudinal joints. In the portion of the circumferential joints of horizontal return tubular boilers exposed to the products of combustion, shearing strength of rivets shall be at least 50% of the full strength of the plate corresponding to the thickness at the joint. Distance from center of rivet holes to edge of plate in boilers having headers supported by tubes or through stays, shall not be less than (1 $^1/_4$ × diam. of rivet hole). Back pitch between two rows of rivets shall not be less than (1 $^3/_4$ × diam. of rivet hole).

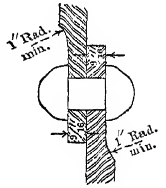


FIG. 5.

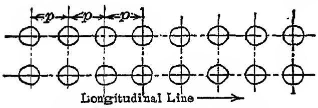
LIGAMENTS (A.S.M.E. Code).—When a shell or drum is drilled for tubes on a line parallel to its axis, the efficiency of the ligaments between tube holes is determined by formulas [4] and [5], where E = efficiency of ligament, P = unit length of ligament, in., p = pitch of holes, in., d = diameter of holes, in., n = number of holes in length P.

When pitch of tube holes on every row is equal (Fig. 6)

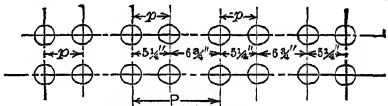
$$E = (p - d) \div p. \qquad [4]$$

When pitch of tube holes on any one row is unequal (Figs. 7 and 8)

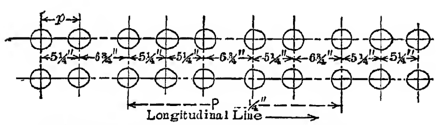
$$E = (P - nd) \div P. \qquad [5]$$



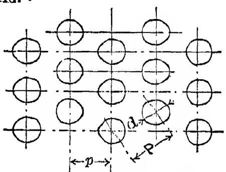
PITCH OF HOLES EQUAL IN EVERY ROW
FIG. 6



PITCH OF HOLES UNEQUAL IN EVERY SECOND ROW
FIG. 7



PITCH OF HOLES VARYING IN EVERY SECOND AND THIRD ROW
FIG. 8



Ligaments FIG. 9

Ligaments between tube holes which are subjected to a longitudinal stress are required to be at least 50% of the strength of ligaments which come between tube holes which are subjected to circumferential stress. If holes are so drilled as to form diagonal ligaments as in Fig. 9, their efficiency is given by the chart Fig. 10, in which abscissas are p/d and ordinates P/p, where p = longitudinal pitch of tube holes, in., P = diagonal pitch of tube holes, in., and d = diameter of tube holes, in. Compute values of p/d and P/p and read efficiency from the diagram. If the point so found falls above the curve of equal efficiency for diagonal and longitudinal ligaments, the longitudinal ligaments will be the weaker, and the efficiency then is computed from formula [4].

EFFICIENCY OF RIVETED JOINTS (A.S.M.E. Code).—Let X = efficiency = ratio of strength of unit length of riveted joint to the strength of the same length of a solid plate; T = tensile strength of material, lb. per sq. in.; t = thickness of plate, in.; b = thickness of butt strap, in.; P = pitch of rivets, in., on row having greatest pitch; d = diameter of rivet, after driving, in.; a = cross-section of rivet after driving, sq. in.; s = strength of rivet in single shear, lb. per sq. in.; S = strength of rivet in double shear, lb. per sq. in.;

c = crushing strength of mild steel, lb. per sq. in.; n = number of rivets in single shear in a unit length of joint; N = number of rivets in double shear in same length of joint.

Single-riveted Lap Joints:

A = strength of solid plate = PtT.
B = strength of plate between rivet holes = $(P - d)tT$.
C = shearing strength of one rivet in single shear = nsa.
D = crushing strength of plate in front of one rivet = dtc.
X = (B/A) or (C/A) or (D/A), whichever is least.

Double-riveted Lap Joints:

A and B as above, C and D to be taken for two rivets.
X = B, C, or D (whichever is least) divided by A.

Butt- and Double-strap Joint, Double-riveted:

A = strength of solid plate = PtT.
B = strength of plate between rivet holes in outer row = $(P - d)tT$.
C = shearing strength of two rivets in double shear, plus shearing strength of one rivet in single shear = $NSa + nsa$.
D = strength of plate between rivet holes in second row, plus shearing strength of one rivet in single shear in outer row = $(P - 2d)tT + nsa$.
E = strength of plate between rivet holes in second row, plus crushing strength of butt-strap in front of one rivet in outer row = $(P - 2d)tT + dbc$.
F = crushing strength of plate in front of two rivets, plus crushing strength of butt-strap in front of one rivet = $Ndtc + ndbc$.
G = crushing strength of plate in front of two rivets, plus shearing strength of one rivet in single shear = $Ndtc + nsa$.
H = strength of butt straps between rivet holes in inner row = $(P - 2d)2bT$.
Note.—This method of failure is not possible for thickness of butt straps required by A.S.M.E. Code, and the computation need be made only for old boilers in which thin butt straps have been used.
X = B, C, D, E, F, G, or H (whichever is least) divided by A.

Butt- and Double-strap Joint, Triple-riveted:

The same as for double-riveted, except that four rivets instead of two are taken for N in computing C, F, and G.

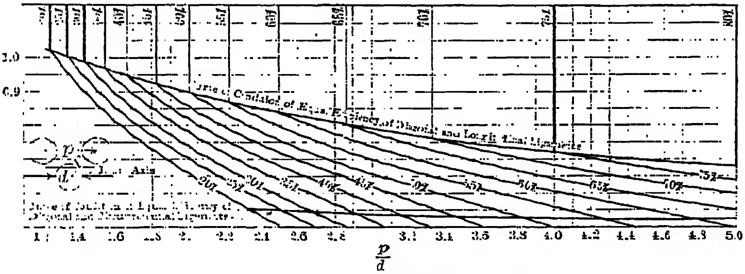


FIG. 10. Efficiency of Diagonal Ligaments

Butt- and Double-strap Joint, Quadruple-riveted:

A, B, and D the same as for double-riveted joints.
C = shearing strength of eight rivets in double shear and three rivets in single shear $= NSa + nsa$.
E = strength of plate between rivet holes in third row (the outer row being the first) plus shearing strength in single shear of two rivets in second row and one rivet in single shear in outer row = $(P - 4d)tT + nsa$.
F = strength of plate between rivet holes in second row, plus crushing strength of butt-strap in front of one rivet in outer row = $(P - 2d)tT + dbc$.
G = strength of plate between rivet holes in third row, plus crushing strength of butt-strap in front of two rivets in second row and one rivet in outer row $= (P - 4d)tT + ndbc$.
H = crushing strength of plate in front of eight rivets, plus crushing strength of butt-strap in front of three rivets = $Ndtc + ndbc$.
I = crushing strength of plate in front of eight rivets, plus shearing strength in single shear of two rivets in second row and one in outer row = $Ndtc + nsa$.
X = B, C, D, E, F, G, H, or I (whichever is least) divided by A.

RIVETING (A.S.M.E. Code).—Rivet and staybolt holes, and holes in braces and lugs to be drilled full size, or punched not to exceed $1/4$ in. less than full diameter for material over $5/16$ in. thick, and $1/8$ in. less for material not over $5/16$ in. thick, and then drilled or reamed to full diameter. Plates, butt straps, braces, heads and lugs to be bolted in position for drilling or reaming all rivet holes in boiler plates. Burrs and chips to be removed before riveting. Rivets shall be of sufficient length to completely fill rivet holes, and to form a head at least equal in strength to the bodies of the rivets. Rivets are to be machine-driven wherever possible, with sufficient pressure to fill rivet holes, and allowed to cool and shrink under pressure.

In computing ultimate strength of rivets in shear, the following values in lb. per sq. in, of cross-sectional area of the rivet shank are to be used: Iron rivets, single shear, 38,000; double shear, 76,000. Steel rivets, single shear, 44,000; double shear, 88,000.

WELDED JOINTS (A.S.M.E. Code).—The ultimate strength of a joint properly welded by the forging process shall be taken as 35,000 lb. per sq. in. when steel is manufactured according to specifications for steel plates of flange and firebox qualities for forge welding. Fusion welding may be used for butt- or lap-welded joints between door-hole flanges of furnace and exterior sheets, provided sheets are stayed around door-hole, and distance between flange and stays does not exceed staybolt pitch. Fusion welding may be used in lieu of riveted joints in fireboxes of internally-fired boilers, provided welds are between two rows of staybolts, or in case of flat surfaces, the welds are not less than one-half staybolt pitch from corner. Furnaces subjected to compression stresses may be fusion welded, provided welds are stress relieved. Ends of inner butt-straps of longitudinal joints may be fusion welded to edges of heads or circumferential butt-straps, provided carbon content of steel does not exceed 0.30%.

FUSION WELDING (A.S.M.E. Code).—Drums or shells of power boilers may be fabricated by means of fusion welding, a process of welding metals in the molten, or molten and vaporous state, without application of mechanical pressure or blows. Joint efficiency shall be taken as 90%. Tension tests are required for both joint specimen and the weld metal specimen. Tensile strength of both specimens must at least equal that of the minimum of range of the plate which is welded. Weld metal specimen must have minimum elongation of 20% in 2 in. Weld metal specimen tensile tests are not required for plate of thickness less than $5/8$ in. Bend-test specimen shall bend cold, until least elongation within or across entire weld on outside fibers is 30%. Specimens of weld metal shall have minimum specific gravity of 7.8. All longitudinal and circumferential welds shall be examined throughout their entire length by radiography.

Where plates of unequal thickness abut, the edge of the thicker shall be reduced to approximately the same thickness as the other. Welded drums shall be designed so that bending stresses are not brought directly on the welded joint. Corner welds shall be avoided, unless plates forming corners are independently supported.

Longitudinal and circumferential joints shall be of double-welded butt type, reinforced at center of weld on each side of plate by $1/16$ in. for plates up to $5/8$ in. incl., and $1/8$ in. for thicker plates. The reinforcement may be removed, but if not removed shall be reasonably free from irregularities, grooves, or depressions. Unreinforced holes shall not be located in a welded joint. Distance between edge of hole and edge of weld to be 1 in. for plates less than 1 in. thick; to be the thickness of the plates for plates 1 in. to 2 in. thick; and to be 2 in. minimum for plates over 2 in. thick.

All fusion-welded boiler drums and all connections attached by fusion welding shall be stress relieved, by uniform heating to 1100° to 1200° F. for a period of 1 hr. per 1 in. thickness, and allowed to cool slowly in still atmosphere.

All fusion-welded boiler drums shall be given a hammer test while under hydrostatic pressure of ($1\ 1/2 \times$ maximum allowable working pressure). Hammer test consists of a swinging blow with a hammer at 6 in. intervals on both sides of welded joint; weight of the hammer in pounds to approximately equal thickness of shell in tenths of an inch, but not over 10 lb. Following this, hydrostatic pressure of twice maximum allowable working pressure is maintained until inspections of all joints and connections are made.

5. BRACED AND STAYED SURFACES

FLAT SURFACES (A.S.M.E. Code).—Maximum allowable pressure for braced and stayed flat plates and surfaces which require staying is found by

$$P = C \times (T^2/p^2), \quad . \quad . \quad . \qquad [6]$$

where P = maximum allowable pressure, lb. per sq. in.; T = thickness of plate *sixteenths* of an inch; p = maximum pitch, in., measured between straight lines through the centers of the staybolts in the different rows, which lines may be horizontal, vertical or inclined; C = a constant whose values are as follows: Stays screwed through plates, with ends

riveted over, plates not over $^7/_{16}$ in. thick, 112; plates over $^7/_{16}$ in. thick, 120; stays screwed through plates, and fitted with single nuts outside of plate, or inside and outside nuts, omitting washers, 135; stays with heads not less than ($1.3 \times$ diam. of stay), screwed through plates or made a taper fit, having the heads formed on stays before installation and not riveted over, and having a true bearing on plate, 150; stays fitted with inside nuts and outside nuts and washers, diameter of washers being not less than $0.4\,p$ and thickness not less than T, 175.

If a doubling plate, of thickness t_1, covers the full stayed area of a flat boiler plate of thickness t, and at least $^3/_8$ in. thick, and is securely riveted thereto, t_1 being not less than $^2/_3\,t$, T in the formula may be taken as $^3/_4\,(t_1 + t)$, but not more than $1.5\,t$, and the value of C may be made $1.15\,C$. If two sheets are connected by stays, only one of which requires staying, the value of C is governed by the thickness of the latter.

To determine distance between centers of rivets or between edge of tube holes and centers of rivets attaching crowfeet of braces to braced surfaces, formula [6] is modified to

$$p = \sqrt{CT^2/P}, \qquad [7]$$

using $C = 135$. For other types of stays, use the value of C given for thickness of plate and type of stay used. To determine the maximum spacing between inner surface of shell and centers of rivets parallel to shell attaching crowfeet of braces to the head, use the above formula, with $C = 175$; for other types of stays take $C = 1.3$ times the value of C applying to thickness of plate and type of stay specified.

Table 4 gives maximum allowable pitch of screwed staybolts, with ends riveted over as computed by the foregoing formula. Values not given in Table 5 may be computed from the formula and used, provided pitch does not exceed $8\,^1/_2$ in. For staybolts adjacent to riveted edges bounding a stayed surface, distance from edge of staybolt hole to a straight line tangent to edges of rivet holes may be substituted for p. When edge of a flat stayed plate is flanged and riveted, distance from center of outermost stays to inside of supporting flange shall not exceed pitch of the stays, p, plus inside radius of flange. Maximum pitch p may be increased by staybolt hole diameter if staybolts are adjacent to a furnace door or other boiler fitting, tube hole, handhole, or other opening.

In water-leg boilers, staybolts may be spaced at greater distances than given in Table 4, if the portion of the sheet between staybolts has sufficient transverse strength to give a factor of safety of 5 at the maximum allowable pressure.

CURVED SURFACES (A.S.M.E. Code).—Two methods are given for finding maximum allowable pressure, the minimum value to be used:

a. Find pressure without making allowance for holding power of stays, but making allowance for weakening effect of stay-holes, or riveted longitudinal joint or other construction. To this add pressure found by formula [6] taking $C = 70$.

b. Find pressure without making allowance for holding power of stays, but making allowance for weakening effect of stay-holes or riveted longitudinal joint or other construction. To this pressure, add pressure corresponding to the strength of the stay for stresses given in Table 5, each stay being assumed to resist steam pressure on the full area of the external surface supported by the stay.

The determination of the maximum allowable working pressure for a wrapper sheet of a locomotive type boiler requires the use of a third rule, which is given in the Code.

SEGMENTS OF HEADS (A.S.M.E. Code) are to be stayed by head-to-head, through, diagonal, crowfoot or gusset stays, except that horizontal tubular boilers not over 36 in.

Table 4.—Maximum Allowable Pitch, in Inches, of Screwed Staybolts, Ends Riveted Over

Pressure, lb. per sq. in.	Thickness of Plate, in.						
	5/16	3/8	7/16	1/2	9/16	5/8	11/16
	Maximum Pitch of Staybolts, in.						
100	5 1/4	6 3/8	7 3/8	......	......	......	......
110	5	6	7	8 3/8	......	......	......
120	4 3/4	5 3/4	6 3/4	8	......	......	......
125	4 3/4	5 5/8	6 5/8	7 3/4	......	......	......
130	4 5/8	5 1/2	6 1/2	7 5/8	......	......	......
140	4 1/2	5 3/8	6 1/4	7 3/8	8 3/8	......	......
150	4 1/4	5 1/8	6	7 1/8	8	......	......
160	4 1/8	5	5 7/8	6 7/8	7 3/4	......	......
170	4	4 7/8	5 5/8	6 3/4	7 1/2	8 3/8	......
180	......	4 3/4	5 1/2	6 1/2	7 3/8	8 1/8	......
190	......	4 5/8	5 3/8	6 3/8	7 1/8	7 7/8	......
200	......	4 1/2	5 1/4	6 1/8	7	7 3/4	8 1/2
225	......	4 1/4	4 7/8	5 7/8	6 1/2	7 1/4	8
250	......	4	4 5/8	5 1/2	6 1/4	6 7/8	7 5/8
300	......	......	4 1/4	5	5 5/8	6 1/4	7

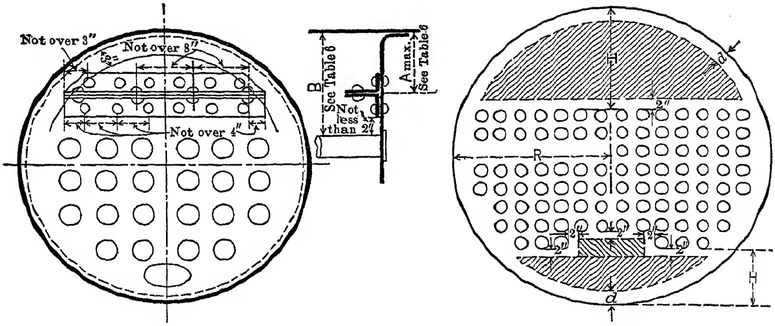


FIG. 11. FIG. 12.

Method of Staying Heads

diam. and designed for maximum pressures not over 100 lb. per sq. in., may have the segments of heads above the tubes stayed by steel angles as shown in Table 6 and Fig. 11. The areas to be stayed are indicated by the shading in Fig. 12.

$$\text{Net stayed area, sq. in.} \quad \frac{- d - 2)^2}{3} \quad \sqrt{} (H - d - 2) \; - 0.608,$$

the meaning of H and R being given in Fig. 12. The value of d may be found by

$$d = 5T/\sqrt{P}, \qquad [9]$$

where T = thickness of head in *sixteenths* of an inch, and P = maximum allowable pressure, lb. per sq. in.; or it may be taken as the outer radius of the flange of the head, not exceeding (8 × thickness of head), the larger value being used. Table 7 gives values for d, based on distance supported by flange of head; Table 8 gives net area to be stayed in shells where d = 3 in. Both tables are condensed from the Boiler Book of the Hartford Steam Boiler Inspection and Insurance Co.

If a horizontal return tubular boiler has a manhole opening below the tubes, flanged as specified under Dished Heads, p. 6–35, area to be stayed as indicated in the lower half of Fig. 12 may be reduced 100 sq. in. The surface around the manhole must be supported by through stays with nuts inside and outside at front head, and by attachments which distribute the stress at the rear head. The clear distance between bodies of braces, or of the inside braces where more than two are used, must not be less than 10 in. at any point.

In water-tube boilers with tubes connected to drum heads of diameter D, the area whose diameter is $(D - 2d)$ is to be stayed, d being found by formula [9]. No stays are required if the drum is 30 in. or less diameter, and the tube plate is stiffened by flanged ribs or gussets, providing a hydrostatic test to destruction of a similar unit shows a factor of safety of 5.

Stays are to be used in tube sheets of fire-tube boilers if distance between edge of tube holes exceeds maximum pitch of staybolts as given in Table 4.

STAYS AND STAYBOLTS (A.S.M.E. Code).—Specifications for holes for staybolts are given under Riveting. See p. 6–30.

Ends of staybolts or stays screwed through plates shall extend beyond the plate not less than two threads and must be riveted over or upset without excessive scoring of the sheets. Ends of staybolts 8 in. long or less must be drilled at least $^3/_{16}$ in. diameter,

Table 5.—Maximum Allowable Stresses for Stays and Staybolts

Description of Stays	Length between Supports	
	Not Over 120 Diameters	Over 120 Diameters
	Stress, lb. per sq. in.	
a. Unwelded or flexible stays less than 20 diameters long, screwed through plates and ends riveted over	7,500	......
b. Hollow steel stays less than 20 diameters long, screwed through plates and ends riveted over	8,000	......
c. Unwelded stays and unwelded portions of welded stays, except as specified in line *a* and line *b*	9,500	8,500
d. Steel through stays exceeding 1 1/2 in. diameter	10,400	9,000
e. Welded portions of stays	6,000	6,000

to a depth of at least 1/2 in. beyond the inside of plates, or hollow staybolts may be used. Flexible staybolts, or solid staybolts over 8 in. long need not be drilled. Staybolts in water-legs of water-tube boilers shall be hollow, or when solid must be drilled.

Riveted-over stay rods must be supported at intervals of at least 6 ft., length being measured from inside faces of stayed surfaces. Maximum allowable stress at area of least net cross-section of stays and staybolts is given in Table 5.

The least cross-sectional area of a stay must be taken in calculating the allowable stress; the strength at the weld also must be calculated with welded stays. Diameter of a screw stay is to be taken at bottom of the thread, or wherever diameter is least.

Stresses in Diagonal and Gusset Stays.—Area required for a diagonal stay is given by $A = aL/l$, where A = sectional area, sq. in., of diagonal stay; a = sectional area sq. in. of a direct stay to support the same surface; L = length of diagonal stay, in.; l = distance, in., at right angles to head, from surface supported to center of palm of diagonal stay. For staying segments in tube sheets, as in horizontal return tubular boilers, where $L < 1.15 \times l$ for any brace, diagonal stays may be calculated as direct stays, allowing 90% of the stress in Table 5.

Gusset Stays of triangular, right-angled plate, secured to angle bars along the sides at right angles, must have a cross-sectional area in the largest plane at right angles to the longest side of the plate, of not less than 10% more than required for a diagonal stay to support the same surface, assuming the diagonal stay to be at the same angle as the longest side of the gusset.

Braces and Brace Connections.—Braces shall be so designed that total load carried shall not exceed the unit stresses for unwelded stays given in Table 5. If welded, braces shall have a cross-sectional area at the weld as great as that computed for a stress of 6000 lb. per sq. in. Area of pins to resist double shear to be at least 75% of the required cross-sectional area of brace. Cross-sectional areas through blades of diagonal braces, where attached to shell, must at least equal the required rivet section, *i.e.*, 1.25 × required cross-sectional area of brace. Each branch of a crowfoot to be designed to carry 2/3 of total load on brace. Net sectional areas through sides of crowfoot or similar fastenings at rivet holes must be at least equal to the required rivet section.

Table 6.—Sizes of Angles Required for Staying Segments of Heads

Short legs of angles attached to head of boiler. All dimensions in inches

Height of Segment, Dimension B, Fig. 11	30-in. Boiler			34-in. Boiler			36-in. Boiler			Dimension, A, Fig. 11
	Angle, 3 × 2 1/2	Angle, 3 1/2 × 3	Angle, 4 × 3	Angle, 3 1/2 × 3	Angle, 4 × 3	Angle, 5 × 3	Angle, 4 × 3	Angle, 5 × 3	Angle, 6 × 3 1/2	
	Thickness			Thickness			Thickness			
10	3/8	5/16	5/16	...	...	...	...	...	...	6 1/2
11	7/16	3/8	5/16	7/16	5/16	5/16	...	...	...	7
12	9/16	7/16	3/8	1/2	7/16	5/16	7/16	5/16	...	7 1/2
13	...	9/16	7/16	11/16	1/2	5/16	9/16	3/8	...	8
14	...	...	1/2	...	5/8	3/8	5/8	7/16	3/8	8 1/2
15	...	...	...	...	...	1/2	3/4	1/2	3/8	9
16	...	...	...	...	...	...	...	5/8	7/16	9 1/2

Table 7.—Value of d (Fig. 12) or Distance Supported by Flange of Head

Working Pressure, lb. per sq. in.	Thickness of Head, Inches										
	3/8	7/16	1/2	9/16	5/8	11/16	3/4	13/16	7/8	15/16	1.0
50	4.24	4.95	5.66	6.36	7.07	7.78	8.49	9.19	9.90	10.61	11.31
60	3.87	4.52	5.16	5.81	6.46	7.10	7.75	8.39	9.04	9.68	10.33
70	3.59	4.18	4.78	5.38	5.98	6.57	7.17	7.77	8.37	8.96	9.56
80	3.35	3.91	4.47	5.03	5.59	6.15	6.71	7.27	7.83	8.39	8.94
90	3.16	3.69	4.22	4.74	5.27	5.80	6.33	6.85	7.38	7.91	8.43
100	3.00	3.50	4.00	4.50	5.00	5.50	6.00	6.50	7.00	7.50	8.00
110	2.86	3.33	3.81	4.29	4.77	5.24	5.72	6.20	6.67	7.15	7.63
120	2.75	3.21	3.66	4.12	4.58	5.04	5.50	5.96	6.42	6.88	7.33
125	2.68	3.13	3.58	4.03	4.47	4.92	5.37	5.81	6.26	6.71	7.16
130	2.63	3.07	3.50	3.95	4.38	4.82	5.26	5.70	6.14	6.57	7.00
140	2.53	2.95	3.38	3.80	4.22	4.64	5.07	5.49	5.91	6.33	6.76
150	2.45	2.85	3.26	3.67	4.08	4.49	4.90	5.31	5.71	6.12	6.53
160	2.37	2.77	3.16	3.56	3.95	4.35	4.74	5.14	5.53	5.93	6.32
170	2.30	2.68	3.06	3.44	3.83	4.22	4.60	4.98	5.37	5.75	6.13
180	2.23	2.60	2.98	3.35	3.72	4.10	4.47	4.84	5.21	5.59	5.96
190	2.17	2.54	2.90	3.26	3.63	3.99	4.35	4.72	5.08	5.44	5.80
200	2.12	2.47	2.83	3.18	3.54	3.89	4.24	4.59	4.95	5.30	5.66

Crown Bar and Girder Stays must be proportioned according to formula

$$\text{Maximum allowable pressure} = Cd^2t \div (W - P)DW, \quad . \quad . \quad . \quad . \quad [10]$$

where W = extreme distance between supports, in.; P = pitch of supporting bolts, in.; D = distance center to center of girders, in.; d and t = depth and thickness of girder, respectively, in.; C = constant, depending on number of supporting bolts as follows:

No. of supporting bolts.	1	2 or 3	4 or 5	6 or 7	8 or more
	7000	10,000	11,000	11,500	12,000

Table 8.—Net Area of Heads to be Stayed When d (Fig. 12) Equals 3 In.

Height, Tubes to Shell H, in.	Diameter of Boiler, Inches												
	24	30	36	42	48	54	60	66	72	78	84	90	96
	Area to be Stayed, Square Inches												
8	28	33	37	40	43	47	51	53	55	58	60	63	65
8 1/2	35	41	46	51	55	59	63	66	70	74	76	80	82
9	42	49	56	62	67	72	76	82	86	90	92	95	98
9 1/2	50	58	66	70	80	86	91	96	101	105	111	116	119
10	57	68	77	85	93	99	106	112	117	123	129	132	137
10 1/2	66	78	89	98	107	114	123	131	135	142	147	153	160
11	74	88	100	111	121	130	138	147	155	161	169	174	183
11 1/2	83	99	112	124	137	146	156	165	173	181	189	196	204
12	91	109	125	139	151	163	174	184	194	203	213	219	230
12 1/2	....	120	138	153	167	180	193	204	216	224	234	243	252
13	....	132	151	168	183	197	211	224	235	247	256	267	279
13 1/2	....	143	164	183	200	216	230	246	258	270	282	293	302
14	....	155	178	199	217	234	250	266	280	294	305	319	331
14 1/2	....	167	192	215	235	254	271	287	303	318	333	345	360
15	....	178	206	231	252	273	291	309	326	343	357	372	386
15 1/2	....	.....	220	247	271	291	312	332	350	368	382	400	417
16	....	.....	235	263	289	312	334	355	374	394	411	423	443
16 1/2	....	.....	249	281	308	332	357	380	399	420	436	457	475
17	....	.....	264	297	326	353	378	402	425	447	467	486	502
17 1/2	....	.....	.....	314	345	374	400	426	449	471	494	516	536
18	....	.....	.....	331	365	396	424	450	476	500	520	543	564
18 1/2	....	.....	.....	349	384	417	448	476	501	526	552	577	598
19	....	.....	.....	366	404	439	470	500	529	555	580	604	631
19 1/2	....	.....	.....	384	424	461	496	528	558	584	613	641	663
20	....	.....	.....	401	444	483	519	552	583	613	642	667	699
20 1/2	....	.....	.....	.....	464	505	543	578	613	643	675	706	729
21	....	.....	.....	.....	485	528	568	604	640	673	705	733	766
21 1/2	....	.....	.....	.....	505	551	594	632	669	703	739	766	797
22	....	.....	.....	.....	526	574	618	658	697	734	769	800	835
22 1/2	....	.....	.....	.....	.....	597	643	687	726	765	800	835	867
23	....	.....	.....	.....	.....	620	668	713	754	796	830	869	906
23 1/2	....	.....	.....	.....	.....	642	695	740	784	827	866	904	945
24	....	.....	.....	.....	.....	667	719	768	814	859	897	939	978
24 1/2	....	.....	.....	.....	.....	689	745	797	843	892	934	975	1018
25	....	.....	.....	.....	.....	714	771	825	875	922	966	1010	1051
25 1/2	....	.....	.....	.....	.....	737	798	855	907	956	1003	1047	1092
26	....	.....	.....	.....	.....	761	824	882	936	987	1035	1083	1126
26 1/2	....	.....	.....	.....	.....	.....	850	909	968	1024	1073	1120	1167
27	....	.....	.....	.....	.....	.....	877	939	998	1053	1106	1157	1202
27 1/2	....	.....	.....	.....	.....	.....	904	968	1030	1089	1145	1195	1243
28	....	.....	.....	.....	.....	.....	930	997	1060	1120	1177	1232	1279
28 1/2	....	.....	.....	.....	.....	.....	.....	1028	1092	1157	1211	1270	1321
29	....	.....	.....	.....	.....	.....	.....	1056	1123	1187	1248	1305	1360
29 1/2	....	.....	.....	.....	.....	.....	.....	1084	1155	1221	1284	1347	1400
30	....	.....	.....	.....	.....	.....	.....	1115	1187	1255	1321	1382	1442
30 1/2	....	.....	.....	.....	.....	.....	.....	.....	1218	1290	1358	1424	1480
31	....	.....	.....	.....	.....	.....	.....	.....	1252	1324	1394	1459	1523
31 1/2	....	.....	.....	.....	.....	.....	.....	.....	1286	1359	1433	1496	1561
32	....	.....	.....	.....	.....	.....	.....	.....	1317	1394	1467	1538	1605
32 1/2	....	.....	.....	.....	.....	.....	.....	.....	.....	1430	1508	1575	1650
33	....	.....	.....	.....	.....	.....	.....	.....	.....	1465	1542	1617	1687
33 1/2	....	.....	.....	.....	.....	.....	.....	.....	.....	1500	1578	1655	1733
34	....	.....	.....	.....	.....	.....	.....	.....	.....	1536	1617	1695	1770
34 1/2	....	.....	.....	.....	.....	.....	.....	.....	.....	.....	1654	1735	1816
35	....	.....	.....	.....	.....	.....	.....	.....	.....	.....	1692	1775	1856
35 1/2	....	.....	.....	.....	.....	.....	.....	.....	.....	.....	.....	1810	1900
36	....	.....	.....	.....	.....	.....	.....	.....	.....	.....	.....	1857	1941

Stay Tubes supporting tube plates in multi-tubular boilers must have a sectional area as follows:

Total section of stay tubes, sq. in. $= (A - a)P \div T$ . . . . [11]

where A = area of that portion of plate containing the tubes, sq. in.; a = aggregate area of holes in tube plate, sq. in.; P = maximum allowable pressure, lb. per sq. in.; T = working tensile stress allowed in tubes, not to exceed 7000 lb. per sq. in. Pitch to be determined by formula [6] using values for C given in Table 9. If ends of tubes are not shielded from radiant heat, reduce C by 20%. Tubes are to project about $1/4$ in. and be slightly flared at each end. Stay tubes, when threaded, to be not less than $3/16$ in. thick at bottom of thread; nuts on stay tubes are not advised. For a nest of tubes, take $C = 140$ and p = mean pitch of stay tubes. For spaces between nests of tubes, p = horizontal distance center to center of bounding rows of tubes and C as in Table 9.

Table 9.—Values of C for Determining Pitch of Stay Tubes.

Pitch of Stay Tubes in Bounding Rows	No Nuts Outside of Plates	With Nuts Outside of Plates
Where there are two plain tubes between two stay tubes.......	$C = 120$	$C = 130$
Where there is one plain tube between two stay tubes..........	$C = 140$	$C = 150$
Where every tube in the bounding rows is a stay tube and each alternate tube has a nut..................................	....	$C = 170$

6. DISHED HEADS AND STEAM DOMES

DISHED HEADS (A.S.M.E. Code). **Convex Heads.**—For a blank unstayed convex head which is a segment of a sphere, with pressure on the concave side,

$$t = (8.33\ PL/2T), \quad \text{[12]}$$

where t = thickness of plate, in.; P = maximum allowable pressure lb. per sq. in.; L = inside radius to which head is dished, in.; T = tensile strength, lb. per sq. in. When two radii are used, the longer shall be used as the value of L. If the radius is less than 80% of the diameter D of the shell to which the head is attached, t shall be at least that found by making $L = 0.8D$. L shall never exceed D.

Concave Heads.—When pressure is on the convex side, maximum allowable pressure shall not be over 60% of that for convex heads of the same dimensions with pressure on the concave side.

Manhole Openings.—For dished heads with manhole openings, t must be increased 15% or at least $1/8$ in. over that called for by the formula, unless it has a flanged opening supported by a flue. Manhole openings are to be flanged to a depth of at least $3t$, measured from the outside, for plate up to 1 $1/2$ in. thick, and 3 in. plus thickness of plate for plates exceeding 1 $1/2$ in. thick. Dished heads thinner than those called for by formula [12] must be stayed as flat surfaces with no allowance for holding power due to the spherical form. The corner radius of an unstayed dished head, measured on the concave side, must be not less than $3t$ and never less than $0.03\ L$.

STEAM DOMES, if used, should be constructed in accordance with the Code.

Inside Diameter 24 in. or Over.—Riveted longitudinal joints to be butt- and double-strap construction, or dome may be made seamless of one piece of steel pressed into shape. Its flange shall be double-riveted to the shell.

Inside Diameter Under 24 in.—If (inside diameter, in. × maximum allowable working pressure) does not exceed 4000, flange may be single-riveted to shell and may be lap-type if computed with factor of safety of not less than 8.

The longitudinal joint of a dome may be butt welded, and the dome flange may be double full-fillet lap welded to the shell, in place of riveting, without requiring X-ray examination. Flanges shall be formed with an inside corner radius of at least $2\ t$ for plates 1 in. thick or less, and at least $3\ t$ for plates over 1 in. thick; t = thickness of plate.

7. BOILER TUBES

PRESSURES ALLOWED (A.S.M.E. Code).—Maximum allowable working pressure for seamless steel tubes for pressures below 875 lb. per sq. in., and lap welded wrought-iron tubes below 358 lb. per sq. in. may be found by

$$P = [\{(t - 0.039)/D\} \times 18{,}000] - 250, \quad \text{[13]}$$

and for lap-welded steel tubes for pressures of 875 lb. per sq. in. and above by

$$P = \{(t - 0.039)/D\} \times 14{,}000, \quad \text{[14]}$$

and for lap-welded wrought-iron tubes for pressures of 358 lb. per sq. in. and above, by

$$P = \{(t - 0.039)/D\} \times 10{,}600, \qquad [15]$$

where P = maximum allowable pressure, lb. per sq. in.; t = thickness of tube wall, in.; D = outside diameter of tube, in. See Table 10. Maximum working pressures for superheater tubes are the same as for boiler tubes.

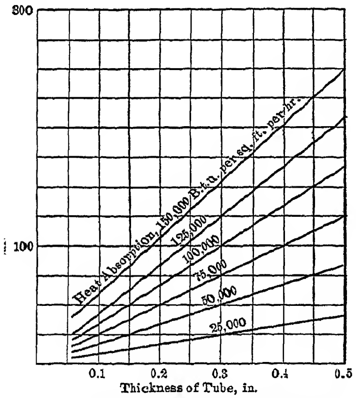
The minimum thickness of tubes for fire-tube boilers, for maximum allowable pressures not over 175 lb. per sq. in. is as follows:

Diam., in.	From.............	1	2 1/2	3 1/4	4	5
	To less than.......	2 1/2	3 1/4	4	5	6
Thickness, B. W. G.............		13	12	11	10	9

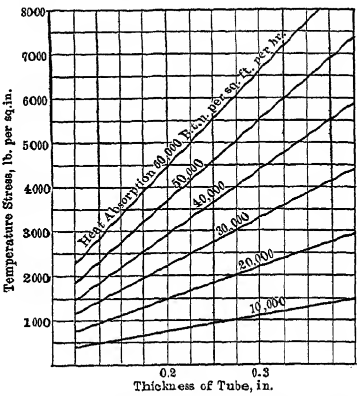
For each increase of one gage in thickness over the above, the maximum allowable pressure will be increased by (200 lb. ÷ tube diam. in.). For cross-sectional area of stay-tubes, see Stays and Staybolts, p. 6–30.

For copper tubes for water-tube boilers formula [16] has been recommended for maximum working pressure, notation being as before. Copper tubes should not be used for pressures over 250 lb. or temperatures over 406° F. See Table 11.

$$P = [\{t - 0.039)/D\}12{,}000] - 250 \qquad [16]$$



13. Temperature Drop through Carbon Steel Tube Walls



. 14. Temperature Stress in Carbon Steel Superheater Tubes

Temperature Stresses.—Figs. 13 and 14 from a paper by Perry Cassidy (Joint meeting Engrs. Soc. Western Penna. and Pittsburgh Section, A.S.M.E., Oct. 15, 1935) give data from which temperature stress in tubes may be determined. For higher rates of heat absorption, as in tubes exposed to furnace radiation, multiply temperature drop given in Fig. 13 by 1.10.

The temperature difference between boiling water and inside scale-free tube surfaces generally is small. Film conductance or transfer rates will be from 2000 to 3000 B.t.u. per hr. per sq. ft. per deg. F. So long as the inside tube surface is in contact with boiling water, temperature drop generally does not exceed 50 deg. F. in the first row of boiler tubes for high radiating temperatures, and 25 to 30 deg. for moderate furnace temperatures. The temperature of the outside surface of front-row scale-free boiler tubes, may be obtained by adding from 15 to 50 deg. to temperature corresponding to drum pressure and then adding temperature drop through tube wall.

For 1400 lb. per sq. in. drum pressure, maximum temperatures of outer surfaces of front-row boiler tubes may be as follows, based on 588° F. water temperature;

Heat absorption, B.t.u. per sq. ft. inside surface per hr.	50,000	75,000	100,000	125,000	150,000
Max. temperature, 3 1/4-in. tubes, 0.375 in. thick....	683	715	748	780	813
Max. temperature, 4-in. tube, 0.45 in. thick.........	694	732	770	808	846

Fig. 15 shows why wall and boiler tubes exposed to furnace radiation must be absolutely free from scale in high-pressure high-capacity boilers. Dense hard sulphate scale will have a thermal conductivity of from 12 to 15 B.t.u. per sq. ft. per hr. per deg. F. per in. thickness. Soft, less dense scale, such as carbonate may have lower conductivity, and it is possible for thin deposits of scale to have conductivity of 2 to 6.

Boiler tubes shielded from furnace radiation, with corresponding low rates of heat absorption may not be damaged by overheating due to reasonable scale deposits, although there are other objections to scale deposits besides overheating. Scale in any part of the boiler may imply scale in the front bank tubes and wall tubes.

TUBE HOLES AND ENDS (A.S.M.E. Code).—Tube holes to be drilled full size from solid plate, or punched at least 1/2 in. smaller in diameter than full size and then drilled, reamed or finished with a rotating cutter to full size. Sharp edges of holes on both sides of plate are to be removed.

Fire-tube boilers are to have both ends of tubes rolled and beaded, or rolled and welded at firebox or combustion chamber end. Ends of all tubes, suspension tubes, and nipples must be flared not less than 1/8 in. over the diameter of tube hole on all water-tube boilers and superheaters, or flared not less than 1/8 in., rolled and beaded, or flared, rolled and welded. Projection through tube sheets or headers to be not less than 1/4 in., nor more than 1/2 in. before flaring.

SIZE OF BOILER TUBES.—Table 12 gives dimensions of boiler tubes sanctioned by the A.S.M.E. Code for stationary fire-tube power boilers, together with their calculated surface per foot of length, and the length per square foot of surface, both external and internal, for allowable pressures up to 175 lb. per sq. in.

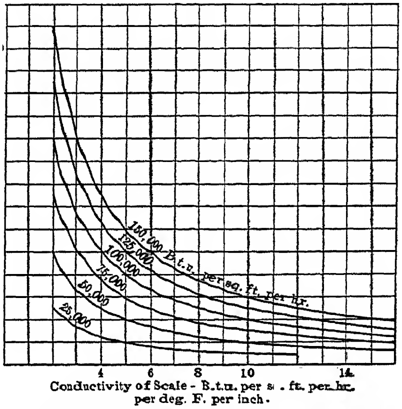


Fig. 15. Temperature Drop through Boiler Scale 0.01 in. Thick

HOLDING POWER OF EXPANDED TUBES.—The safe holding power of expanded tubes with smooth tube seats often is a factor in boiler design. The holding power of such seats varies, depending on the type of finish. Fig. 16 gives the safe holding power of drilled and reamed tube seats properly expanded and flared. It will be noted that up to a tube thickness of 0.2 in., holding power of the 2-in. tube seat 1 in. wide is greater than allowable stress in the tube. The 3 1/4-in. and 4-in. tube seats are stronger than the tubes below a thickness of 0.12 in. Holding power of the seat is based on an approximate factor

Table 10.—Maximum Allowable Working Pressures for Seamless Steel Tubes for Water-tube Boilers

B. W. G. =	17	16	15	14	13	12	11	10	9	8	7	6	5
t =	.058	.065	.072	.083	.095	.109	.120	.134	.148	.165	.180	.203	.220
D, inches	Lb. per sq. in.												
1/2	434	686	938	1334	1766	2270	.....	.....	.....	.....	.....	.....	.....
3/4	206	374	542	806	1094	1430	1694	2030	.....	.....	.....	.....	.....
1	.....	218	344	542	758	1010	1208	1460	1712	2018	.....	.....	.....
1 1/8	.....	166	278	454	646	870	1046	1270	1494	1766	2006	.....	.....
1 1/4	.....	124	225	383	557	758	916	1118	1319	1564	1780	2111	.....
1 1/2	.....	.....	146	278	422	590	722	890	1058	1262	1442	1718	1922
1 3/4	.....	.....	.....	203	326	470	583	727	871	1046	1200	1437	1612
2	.....	.....	.....	146	254	380	479	605	731	884	1019	1226	1379
2 1/4	.....	.....	.....	.....	198	310	398	510	622	758	878	1062	1198
2 1/2	.....	.....	.....	.....	153	254	333	434	535	657	765	931	1053
2 3/4	.....	.....	.....	.....	117	208	280	372	464	575	673	824	935
3	.....	.....	.....	.....	.....	170	236	320	404	506	596	734	836
3 1/4	.....	.....	.....	.....	.....	.....	199	276	354	448	531	658	752
3 1/2	.....	.....	.....	.....	.....	.....	167	238	310	398	475	594	681
3 3/4	.....	.....	.....	.....	.....	.....	139	206	273	355	427	537	619
4	.....	.....	.....	.....	.....	.....	.....	178	240	317	385	488	565
4 1/2	.....	.....	.....	.....	.....	.....	.....	.....	186	254	314	406	474
5	.....	.....	.....	.....	.....	.....	.....	.....	142	204	258	340	402

Table 11.—Maximum Allowable Working Pressure for Copper Tubes for Water-tube or Fire-tube Boilers

B. W. G. =	12	11	10	9	8	7	6	5	4
	Lb. per sq. in.								
D = 2 in.	170	231	250	250	250	250	250	250	250
3 1/4	....	....	101	142	215	250	250	250	250
4	....	....	....	....	128	173	242	250	250
5	....	....	....	....	....	....	143	184	218

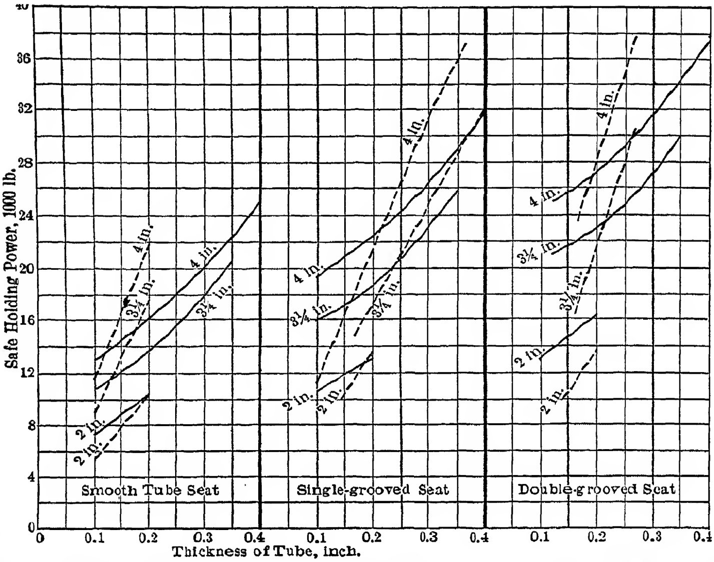


Fig. 16. Holding Power of Expanded Tubes. Tube seat 1 in. wide. Temperatures below 750° F.

of safety of 2, governed by initial slip of the tube resulting in leakage. The ultimate factor of safety is correspondingly higher. Fig. 16 shows also holding power of tube seats with single and double grooves. A change in the width of these tube seats will only affect holding power of the smooth portion.

Eccentric loading on tube seats, due to beam action or other causes, may result in high moments at the tube seat which require careful analysis to determine the safety.

8. BOILER FITTINGS

FLANGES AND FLANGED FITTINGS should conform to the standards of the American Standards Association. Dimensions of flanges of all valves and pipe fittings of cast iron or steel are given in Section 5. See pp. 5-42 to 5-68.

CAST-IRON AND MALLEABLE HEADERS FOR WATER-TUBE BOILERS.—Maximum pressure allowed 160 lb. per sq. in. for cast iron; 350 lb. per sq. in. for malleable. Cast-iron headers tested to destruction must stand hydrostatic pressure of at least 1200 lb. per sq. in.; malleable, 2250 lb. per sq. in. Hydrostatic test required on all new headers

Table 12.—Dimensions of Standard Boiler Tubes for Fire-tube Boilers

Outside Diameter, in.	Standard Thickness, in.	Birmingham Wire Gage No.	Inside Diameter, in.	Inside Circumference, in.	Outside Circumference, in.	Length of Tube per sq. ft. of Inside Surface, ft.	Length of Tube per sq. ft. of Outside Surface, ft.	Cross-Sectional Area, Inside, sq. in.	Cross-Sectional Area, Outside, sq. in.	Nominal Weight per ft., lb.
1	0.095	13	0.81	2.54	3.14	4.73	3.82	0.52	0.78	0.87
1 1/4	.095	13	1.06	3.33	3.92	3.60	3.06	0.88	1.22	1.14
1 1/2	.095	13	1.31	4.11	4.71	2.92	2.55	1.35	1.76	1.38
2	.095	13	1.80	5.66	6.28	2.11	1.90	2.55	3.14	1.91
2 1/2	.109	12	2.28	7.17	7.85	1.67	1.52	4.09	4.90	2.75
3	.109	12	2.78	8.74	9.42	1.37	1.27	6.08	7.06	3.33
3 1/2	.120	11	3.26	10.24	10.99	1.17	1.09	8.35	9.62	4.28
4	.134	10	3.74	11.75	12.56	1.02	0.95	10.99	12.56	5.47
4 1/2	.134	10	4.24	13.32	14.13	0.90	0.84	14.12	15.90	6.17
5	.148	9	4.72	14.81	15.70	0.80	0.76	17.49	19.63	7.58
6	.165	8	5.69	17.90	18.84	0.67	0.63	25.50	28.27	10.16

with tubes attached, as follows: Cast iron, 500 lb. per sq. in.; malleable, ($2\,^1/_2 \times$ working pressure), but never less than 500 lb. per sq. in.

MANHOLES. WASH-OUT HOLES (A.S.M.E. Code).—Minimum size of elliptical manholes 11×15 in. or 10×16 in. Minimum diameter, circular manholes, 15 in. Handholes, whose greatest dimension exceeds 6 in. to be reinforced as manholes. Reinforced rings to be of rolled, forged or cast steel, and of minimum thickness according to Table 13. Manhole frames on shells or drums shall have proper curvature, and

Table 13.—Minimum Thickness of Independent Riveted Reinforcing Rings or Flanges

Thickness of shell plate, in..			1/4–11/32	3/8–13/32	7/16–15				1 1/8–2	over 2
Thickness of reinforcing ring or flange, in.	1/8	3/16	1/4	5/16	3/8	7/16	1/2	5/8	3/4	

on boilers over 48 in. diam. shall be riveted to shell with two rows of rivets. Their strength, and also that of rivets in shear, must at least equal tensile strength of a cross-sectional area determined by product $t \times (d - 2)$, where t = shell thickness (using 90% efficiency), and d = diameter of opening in shell in the finished construction. Manhole plates are to be of rolled, forged or cast steel. Minimum width of bearing surface for gasket, $^{11}/_{16}$ in. Maximum thickness of gasket, $^1/_4$ in. when compressed.

A manhole must be located in front head below the tubes of horizontal return tubular boilers of 48 in. or more diameter. Smaller boilers to have either manhole or handhole below tubes. A manhole is required in the upper part of shell or head of fire-tube boilers over 40 in. diameter, except vertical fire-tube boilers or internally-fired boilers not over 48 in. diameter. Smaller boilers must have handholes or manholes above the tubes.

Wash-out plugs or handholes are to be provided as follows: In traction, portable or locomotive-type stationary boilers: Rear head below tubes, 1; front head at crown sheet, 1; lower part of water leg, 4; near throat sheet, 1. In vertical fire-tube boilers, except those of steam fire engines or boilers 24 in. diam. or less: shell at line of crown sheet, 3; shell at water line or opposite fusible plug when used, 1; shell at lower part of water leg, 3. In vertical fire-tube boilers, submerged tube type; shell, in line with upper tube sheet, 2 or more. All boilers 24 in. diam. or less shall have at least one opening for inspection, and two openings in addition to blow-off for washing out boiler. Openings shall not be less than 1 in. pipe size and shall be fitted with brass plugs.

THREADED OPENINGS.—Minimum number of threads for all pipe connections of 1 in. diam. or over is given in Table 14. Threaded joints shall not be used, either at the shell or at terminating end of connections over 3 in. pipe size, where maximum allowable working pressure exceeds 100 lb. per sq. in.

Table 14.—Minimum Number of Threads for Pipe Connections

Size of connection, in.	1 & 1 1/4	1 1/2 & 2	2 1/2 to 4	4 1/2 to 6	7 & 8	9 & 10	12
Threads per in.	11 1/2	11 1/2	8	8	8	8	8
Min. number threads required.	4	5	7	8	10	12	13
Min. thickness of material, in.	0.348	0.435	7/8	1	1 1/4	1 1/2	1 5/8

FUSIBLE PLUGS (A.S.M.E. Code).—Fusible plugs, if used, shall be filled with tin with a melting point between 400 and 500° F. and shall be renewed once each year. Least diameter of fusible metal shall be not less than $^1/_2$ in., except for maximum allowable working pressures of over 175 lb. per sq. in. or when it is necessary to place a fusible plug in a tube, in which case least diameter of fusible metal shall be not less than $^3/_8$ in., the tube wall being not less than 0.22 in. thick or sufficient to give four threads. Fusible plugs, if used, shall be located at lowest permissible water level of the boiler. Fusible plugs are not advisable when boilers are to operate at working pressures over 225 lb. per sq. in.

9. SAFETY VALVES

Safety valves are designed to open automatically to relieve excess pressure in boiler or pressure vessel to which they are attached. They may be of the direct, spring-loaded pop type. Dead weight or weighted-lever safety valves shall not be used.

DISCHARGE CAPACITY (A.S.M.E. Code).—The capacity of all safety valves of $^1/_2$ in. size or larger shall be the manufacturer's guarantee, and shall be plainly marked by him in such a way as not to be obliterated in service. Markings shall give: *a*, name of manufacturer; *b*, size, in.; *c*, pressure, lb.; *d*, blow-down, lb.; *e*, capacity, lb. per hour; *f*, A.S.M.E. Standard. Capacity is to be figured at a pressure 3% higher than setting pressure, and with valve adjusted for blow-down given under *d*.

SAFETY VALVE REQUIREMENTS (A.S.M.E. Code).—Each boiler shall have at least one safety valve. If water heating surface exceeds 500 sq. ft., or steam generating capacity exceeds 2000 lb. per hour, two or more safety valves are required. Safety valve

capacity for each boiler shall be such that the safety valve or valves will discharge all the steam that can be generated by the boiler without allowing pressure to rise more than 6% above maximum allowable working pressure, or more than 6% above highest pressure to which any valve is set, if this pressure is less than maximum allowable working pressure.

One or more safety valves on every boiler shall be set at or below maximum allowable working pressure. The remaining valves may be set within a range of 3% above such pressure, but the range of setting of all of the valves on a boiler shall not exceed 10% of the highest pressure to which any valve is set.

Safety valves shall be of such a type that no failure of any part can obstruct free and full discharge of steam from the valve. They may be of the direct spring-loaded pop type, with seat and bearing surface of the disc inclined at an angle between 45° and 90° to the center line of the spindle. The valve shall be rated at a pressure 3% in excess of that at which it is set to blow, and a blow-down of not more than 4% of said pressure, but never less than 2 lb. Safety valves may be used which give any opening up to full discharge capacity of area of opening at inlet of the valve. Weighted-lever safety valves are prohibited.

Safety valves shall operate without chattering and shall be set and adjusted as follows: To close after blowing down not more than 4% of the set pressure, but not less than 2 lb. in any case. For spring-loaded pop safety valves, operating at pressures up to and including 300 lb. per sq. in., blow-down shall not be less than 2% of set pressure. Blow-down as marked on valve shall not be reduced.

Seats and discs must be of corrosion-resistant metal, the seat being so fastened to the body that there is no possibility of its lifting.

Springs shall not show a permanent set exceeding 1/16 in., 10 minutes after being released from a cold compression test which closes the springs solid. The valve must be able to lift from its seat at least 1/10 diameter of the seat before the coils of the springs are closed or before there is any other interference. No springs shall be used for any pressure greater than 10% above or below that for which it was designed.

Safety valves over 3 in. size and used for gage pressures greater than 15 lb. per sq. in., shall have flanged inlet connection.

Safety Valves on Superheaters.—Every attached superheater shall have one or more safety valves near the outlet, whose discharge capacities may be included in determining

Table 15.—Safety Valves for Fire-tube Boilers

Pressure Range from 15 lb. to 100 lb., Inclusive

Nominal Horsepower Rating	Minimum Capacity, lb. per hr.	45-deg. Bevel Seat								Flat Seat							
		25 lb.		50 lb.		75 lb.		100 lb.		25 lb.		50 lb.		75 lb.		100 lb.	
		No.	Diam., in.	No.	Diam., in.	No.	Diam., in.	No.	Diam., in.	No.	Diam., in.	No.	Diam., in.	No.	Diam., in.	No.	Diam., in.
50	1,500	1	4	1	3	1	3	1	2	1	3 1/2	1	3	1	2	1	2
75	2,250	2	3 1/2	2	3	2	2	2	2	2	3	2	2	2	2	2	1 1/2
100	3,000	2	4	2	3	2	3	2	2	2	3 1/2	2	3	2	2	2	2
120	3,600	2	4 1/2	2	3 1/2	2	3	2	2 1/2	2	3 1/2	2	3	2	2 1/2	2	2
140	4,200	2	4 1/2	2	3 1/2	2	3	2	3	2	4	2	3	2	3	2	2
160	4,800	3	4	2	4	2	3 1/2	2	3	2	4	2	3 1/2	2	3	2	2 1/2
180	5,400	3	4 1/2	2	4	2	3 1/2	2	3	2	4 1/2	2	3 1/2	2	3	2	2 1/2
200	6,000	3	4 1/2	2	4 1/2	2	3 1/2	2	3	2	4 1/2	2	3 1/2	2	3	2	3
220	6,600	4	4	2	4 1/2	2	4	2	3 1/2	3	4	2	4	2	3	2	3
240	7,200	4	4 1/2	3	4	2	4	2	3 1/2	3	4	2	4	2	3 1/2	2	3

For Pressures in Excess of 100 lb.

Nominal Horsepower Rating	Minimum Capacity, lb. per hr.	45-deg. Bevel Seat												Flat Seat							
		125 lb.		150 lb.		175 lb.		200 lb.		225 lb.		250 lb.		150 lb.		175 lb.		200 lb.		225 lb.	
		No.	Diam., in.	No.	Diam., in.	No.	Diam., in.	No.	Diam., in.	No.	Diam., in.	No.	Diam., in.	No.	Diam., in.	No.	Diam., in.	No.	Diam., in.	No.	Diam., in.
50	2,500	2	2	2	1 1/2	2	1 1/2	2	1 1/2	2	1 1/4	2	1 1/4	2	1 1/4	2	1 1/4	2	1	2	1
75	3,750	2	2 1/2	2	2	2	2	2	2	2	1 1/2	2	1 1/2	2	1 1/2	2	1 1/2	2	1 1/2	2	1 1/2
100	5,000	2	3	2	2 1/2	2	2 1/2	2	2	2	2	2	2	2	2	2	2	2	2	2	1 1/2
120	6,000	2	3	2	3	2	2 1/2	2	2 1/2	2	2	2	2	2	2	2	2	2	2	2	2
140	7,000	2	3	2	3	2	3	2	2 1/2	2	2 1/2	2	2 1/2	2	2 1/2	2	2 1/2	2	2	2	2
160	8,000	2	3 1/2	2	3	2	3	2	3	2	3	2	2 1/2	2	3	2	2 1/2	2	2 1/2	2	2
180	9,000	2	3 1/2	2	3 1/2	2	3	2	3	2	3	2	3	2	3	2	3	2	2 1/2	2	2 1/2
200	10,000	2	4	2	3 1/2	2	3 1/2	2	3	2	3	2	3	2	3	2	3	2	3	2	2 1/2
220	11,000	2	4	2	3 1/2	2	3 1/2	2	3	2	3	2	3	2	3	2	3	2	3	2	2 1/2
240	12,000	2	4	2	4	2	3 1/2	2	3 1/2	2	3	2	3	2	3	2	3	2	3	2	3

number and size of safety valves for the boiler if there are no intervening valves between the superheater safety valve and the boiler, and if discharge capacity of safety valves on the boiler as distinct from the superheater, is at least 75% of the total valve capacity required. Every independently-fired superheater which may be shut off from the boiler, permitting superheater to become a fired pressure vessel, shall have one or more safety valves with a discharge capacity equal to 6 lb. of steam per sq. ft. of superheating surface, measured on side exposed to hot gases. The number of safety valves installed shall be such that the total capacity is at least equal to that required. A soot blower connection may be attached to the same outlet from the superheater as used for safety valve connection. Safety valves on superheaters discharging superheated steam at temperatures over 450° F. shall have a casing, including base, body, bonnet and spindle, of steel, steel alloy or equivalent heat-resisting material, a seat and a disc of nickel composition or equivalent, a flanged inlet connection, and a fully-exposed spring outside the casing, protected from escaping steam.

SAFETY VALVES FOR LOCOMOTIVES.—Requirements are same as those given for power boilers with following exceptions: Every locomotive boiler shall have at least two safety valves of capacity sufficient to prevent a rise of more than 5% above specified boiler pressure. Safety valves shall be set to pop at pressures not over 6 lb. above working steam pressure.

SAFETY VALVE CAPACITY REQUIRED (A.S.M.E. Boiler Code).—Minimum aggregate relieving capacity of the safety valve or valves required on a boiler is determined on the basis of 6 lb. of steam per hr. per sq. ft. of boiler heating-surface for water-tube boilers. For all other types of power boilers, minimum allowable relieving capacity is determined on the basis of 5 lb. of steam per hr. per sq. ft. of heating-surface, where maximum allowable working pressure is above 100 lb. per sq. in. For maximum allow-

Table 16.—Safety Valves for Water-tube Boilers

Nominal Rated Horse-power	Minimum Capacity, lb. per hr.	45-deg. Bevel Seat											
		100 lb.		125 lb.		150 lb.		200 lb.		250 lb.		300 lb.	
		No.	Diam., in.	No.	Diam., in.	No.	Diam., in.	No.	Diam., in.	No.	Diam., in.	No.	Diam., in.
100	6,000	2	3	2	3	2	3	2	2 1/2	2	2	2	2
125	7,500	2	3 1/2	2	3 1/2	2	3	2	3	2	2 1/2	2	2
150	9,000	2	4	2	3 1/2	2	3 1/2	2	3	2	3	2	2 1/2
200	12,000	2	4 1/2	2	4	2	4	2	3 1/2	2	3	2	3
250	15,000	3	4	2	4 1/2	2	4 1/2	2	4	2	3 1/2	2	3
300	18,000	3	4 1/2	3	4	3	4	2	4	2	3 1/2	2	3 1/2
350	21,000	4	4 1/2	3	4 1/2	3	4	2	4 1/2	2	4	2	3 1/2
400	24,000	4	4 1/2	4	4	3	4 1/2	3	4	2	4 1/2	2	4
450	27,000	....	......	4	4 1/2	4	4	3	4	2	4 1/2	2	4
500	30,000	....	......	4	4 1/2	4	4 1/2	3	4 1/2	3	4	2	4 1/2
550	33,000	....	......	....	......	4	4 1/2	3	4 1/2	3	4	2	4 1/2
600	36,000	....	......	....	......	....	......	4	4	3	4 1/2	3	4
700	42,000	....	......	....	......	....	......	4	4 1/2	3	4 1/2	3	4 1/2
800	48,000	....	......	....	......	....	......	....	......	4	4 1/2	3	4 1/2
900	54,000	....	......	....	......	....	......	....	......	4	4 1/2	4	4 1/2
1000	60,000	....	......	....	......	....	......	....	......	....	......	4	4 1/2

Nominal Rated Horse-power	Minimum Capacity, lb. per hr.	Flat Seat											
		100 lb.		125 lb.		150 lb.		200 lb.		250 lb.		300 lb.	
		No.	Diam., in.	No.	Diam., in.	No.	Diam., in.	No.	Diam., in.	No.	Diam., in.	No.	Diam., in.
100	6,000	2	3	2	2 1/2	2	2	2	2	2	1 1/2	2	1 1/2
125	7,500	2	3	2	3	2	2 1/2	2	2	2	2	2	2
150	9,000	2	3 1/2	2	3	2	3	2	2 1/2	2	2 1/2	2	2
200	12,000	2	4	2	3 1/2	2	3	2	3	2	2 1/2	2	2 1/2
250	15,000	2	4 1/2	2	4	2	3 1/2	2	3	2	3	2	3
300	18,000	3	4	2	4 1/2	2	4	2	3 1/2	2	3	2	3
350	21,000	3	4	2	4 1/2	2	4 1/2	2	4	2	3 1/2	2	3
400	24,000	3	4 1/2	3	4	2	4 1/2	2	4	2	3 1/2	2	3 1/2
450	27,000	4	4	3	4 1/2	3	4	2	4 1/2	2	4	2	3 1/2
500	30,000	4	4 1/2	3	4 1/2	3	4	2	4 1/2	2	4	2	3 1/2
550	33,000	4	4 1/2	4	4	3	4 1/2	3	4	2	4 1/2	2	4
600	36,000	....	......	4	4 1/2	3	4 1/2	3	4	2	4 1/2	2	4
700	42,000	....	......	4	4 1/2	4	4 1/2	3	4 1/2	3	4	2	4 1/2
800	48,000	....	......	....	......	4	4 1/2	3	4 1/2	3	4	3	4
900	54,000	....	......	....	......	....	......	4	4 1/2	3	4 1/2	3	4
1000	60,000	....	......	....	......	....	......	4	4 1/2	3	4 1/2	3	4 1/2

able working pressures of 100 lb. per sq. in. or less, the basis is 3 lb. of steam per hr. per sq. ft. of boiler heating-surface. Tables 15 and 16, condensed from the Boiler Book of the Hartford Steam Boiler Inspection and Insurance Co., give the number and dimensions of valves required on various sizes of boilers at different pressures, the tables being computed in accordance with the foregoing rules, and average lifts for the valves being assumed as follows:

Diam. of valve, in.	1	1 1/4	1 1/2	2	2 1/2	3	3 1/2	4	4 1/2
Assumed lift, in.	0.04	0.04	0.05	0.06	0.06	0.08	0.09	0.10	0.11

In all cases valves shall conform to the requirement that they discharge all steam that can be generated by the boiler without allowing pressure to rise more than 6% above highest pressure to which any valve is set. If two or more safety valves are used on a boiler, they may be either separate or twin valves, made by mounting individual valves on Y-bases, duplex, triplex or multiplex valves having two or more valves in the same body casing. Safety valves shall be connected to the boiler independently of any other steam connection and attached as closely as possible to the boiler. They shall be connected, so as to stand in an upright position, with spindle vertical whenever possible. Each safety valve connection shall have a cross-sectional area not less than the combined inlet areas of all valves mounted thereon. In fire-tube boilers, openings in the boilers for safety valves shall not be less than given in Table 17. No valve of any description shall be placed between the safety valve or valves and boiler, nor on discharge pipe between the valves and atmosphere. Discharge pipes must be at least the full size of the valve and must be fitted with an open gravity drain at or near each safety valve where water of condensation may collect. Each valve must have an open gravity drain through the casing below the level of valve seat. If a muffler is used, it shall have sufficient outlet area to prevent back-pressure from interfering with proper operation and discharge capacity of valve. The discharge capacities given in Table 17 may be interpolated to determine the values for intermediate pressures.

Table 17.—Minimum Size of Boiler Outlets for Safety Valves from Fire-tube Boilers for Various Discharge Capacities

Gage Pressure, lb. per sq. in.	Nominal Pipe Size and Actual Diameters of Pipe Sizes, in.													Special Sizes (Use not recommended)	
	1/2 0.622	3/4 0.824	1 1.049	1 1/4 1.380	1 1/2 1.610	2 2.067	2 1/2 2.469	3 3.068	3 1/2 3.548	4 4.026	5 5.047	6 6.065	8 7.981	4 1/2 4.506	7 7.023
	Lb. of Steam per Outlet per Hour														
15			131	163	245	391	486	782	1026	1,303	2,052	2,916	5,212	1,613	3,909
25			174	218	326	523	653	1046	1372	1,742	2,744	3,898	6,968	2,156	5,226
50			284	354	532	851	1064	1703	2235	2,839	4,470	6,352	11,356	3,513	8,517
75			393	492	738	1181	1475	2361	3099	3,935	6,198	8,805		4,870	11,805
100			503	629	944	1510	1877	3019	3963	5,032	7,926	11,259		6,227	
125			613	767	1149	1836	2299	3677	4826	6,128	9,652	13,711		7,583	
150			723	904	1355	2158	2710	4335	5690	7,226	11,380			8,940	
175			835	1040	1561	2497	3121	4993	6553	8,320	13,106			10,298	
200			941	1178	1766	2826	3532	5651	7418	9,420	14,836			11,655	
225			1052	1315	1972	3154	3944	6310	8280	10,514				13,013	
250			1161	1451	2177	3484	4355	6968	9143	11,614				14,366	

SUPERHEATERS, ECONOMIZERS AND AIR HEATERS

1. SUPERHEATERS *

When heat is added to water in a closed vessel, and the water is transformed into steam, the temperature of the liquid and the steam is equal and substantially constant, so long as any water remains as liquid. The quantity of heat that can be added to a unit weight of water also is a constant, provided that pressure remains constant throughout the process. When all the water has been transformed into steam, further addition of heat raises the temperature and heat content of the steam, which then is *superheated*. If the steam be withdrawn from presence of the water as it is generated, and passed through heated tubes it will become superheated. This is the principle of the superheater used in commercial practice. The superheater is that portion of the steam generating

* The curves in this section and the notes accompanying them are from The Trend of Boiler Design, by Perry Cassidy, presented at a joint meeting of Engrs. Soc. of Western Penn. and Pittsburgh Section, A.S.M.E., Oct. 15, 1935.

unit which receives saturated steam from the boiler proper, and in which further transfer of heat occurs from the hot gases of combustion to the steam.

Superheaters are installed, almost invariably, in large and modern power plants throughout the world. The efficiency of a power-generating unit, including boiler and turbine, will be higher with a superheater than without one. Also, further economies are realized in the plant as a whole because of reduced steam consumption of prime movers and auxiliaries. On the other hand, superheaters would not be advisable in certain cases, for example, where process or heating steam is used and constant temperatures are desired.

INTEGRAL SUPERHEATERS.—Integral superheaters, convection or radiantly-heated, are placed within the boiler setting, and receive heat from the same source as the boiler proper. Choice of location of the superheater, relative to water-tube heating surface, depends on several factors. For small amounts of superheat, a conventional type convection superheater, in boilers of the Babcock & Wilcox type usually is located above the first and second boiler passes (overdeck superheater). See Fig. 1. If higher steam temperatures are required, the superheater may be placed between tube rows, where it is subjected to higher temperature gases (interdeck superheater). Although close to the furnace, the boiler surface retained between it and the furnace shields the superheater tubes from direct furnace radiation. See Fig. 2.

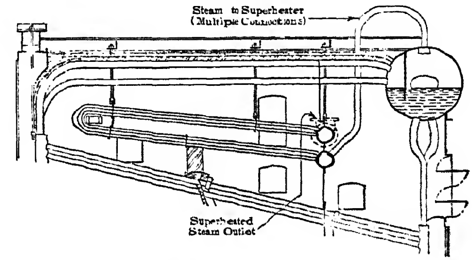


Fig. 1. Babcock & Wilcox Overdeck Superheater

Convection Superheater.—Early types of convection superheaters comprised a multiplicity of U-shaped tubes, connected in parallel between inlet and outlet headers. Saturated steam was supplied to one end of the inlet header through a single pipe connection from the boiler drum. Uniform distribution to individual tubes was insured by baffles in the header and ferrules in the tubes. Modern (1935) superheaters also have parallel circuits, but each tube is longer and is formed into an element comprising several return bends, giving a longer path between inlet and outlet connections. Several tube connections, spaced at intervals, from boiler steam drum to superheater inlet header, effect uniform distribution of saturated steam.

Inasmuch as superheater elements, in any conventional location in the setting, are exposed to relatively high temperature gases, tube metal may be overheated to the point of failure unless tubes are properly cooled by the flow of steam through them. Hence, uniform distribution of steam to all tubes is desirable, to maintain sufficient velocity of flow with a minimum pressure loss between drum and superheat outlet. Maximum steam temperature requirements are met either by radiant-heat superheaters combined with convection superheaters, or by separately fired superheaters.

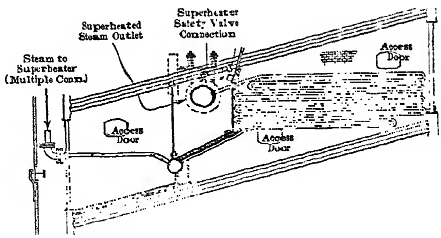


Fig. 2. Babcock & Wilcox Interdeck Superheater

In Stirling-type boilers, the convection superheater is between the first and second tube banks. The degree of superheat attained in normal operation is governed by extent of superheating surface, by effect of gas baffles in tube banks, and by amount of boiler surface preceding superheater. See Fig. 3.

Radiant Superheaters are exposed to the furnace, and the transfer of heat is by direct radiation. They usually are placed in one or more walls of the furnace, and in some cases replace a water wall. Operating conditions are more severe than with the convection type because of the high temperatures to which they are exposed. Greater attention must be given to design, to avoid tube failure due to overheating. This necessitates high steam flow, which is accomplished at the expense of pressure drop. In this type of super-

heater, steam temperature decreases with increased load. It, therefore, seldom is used except in combination with the convection superheater. Properly proportioned, such a combination can maintain constant steam temperature over a wide range of load.

COMBINATION OF CONVECTION AND RADIANT HEAT SUPERHEATERS.—In the convection type superheater, increase in superheat usually accompanies increase in load. This may be 20 to 30% higher at maximum load than when boiler is operated at lowest desired capacity. With radiant-heat superheaters, increase in load tends to decrease the amount of superheat. Within certain limits, the closer the superheater is to the furnace the more uniform will be superheat throughout the operating range. This is because the superheater becomes, in effect, a combination convection and radiant-heat superheater. A comparatively constant degree of superheat can be obtained by a properly arranged combination of convection and radiant-heat superheaters, connected in series. The steam flows first through the convection superheater. This combination is subject to limitations, as steam velocity must be high enough, especially in the radiant-heat superheater, to prevent overheating of metal, and under some conditions an excessive pressure drop through the two superheaters would result.

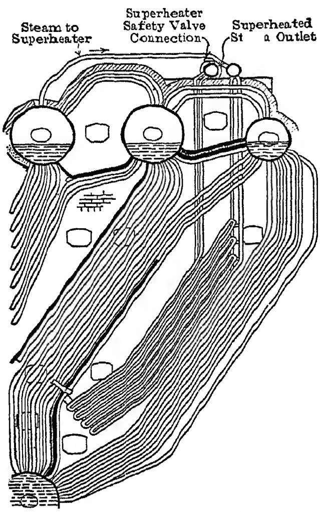


Fig. 3. Convection Superheater in Stirling Boiler

SEPARATELY FIRED SUPERHEATERS are advantageous where a high degree of superheat is required. Certain processes call for steam temperatures that would cause operating difficulties if attempted with an integral superheater. A change in prime movers or industrial process, may require a higher steam temperature from a plant already in operation. The most economical solution may be a separately fired superheater supplied from the main steam line. One superheater can serve several boilers.

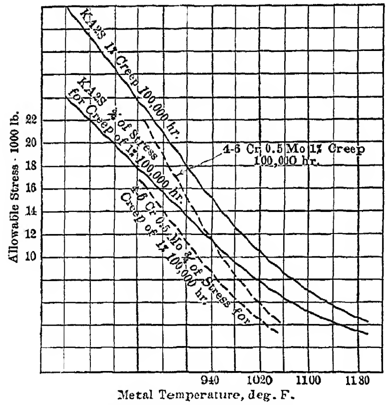


Fig. 4. Creep Stress in Alloy Superheater Tubes

Separately fired superheaters require carefully supervised operation. Interrupted steam flow, occasioned by decreased demand for steam, will lead to excessive superheat, unless firing conditions are carefully controlled.

Various arrangements of heating surface are used. Best economies result with a counterflow of steam and gas, the gases usually making three or four passes over the heating surface. Provision may be made against danger of excessive tube temperatures, or tube burning, by a combination of parallel flow in the high-temperature portion, and counterflow in the lower-temperature portion of the unit. Temperatures also may be controlled by rate of firing and variation in excess air supply.

A Babcock & Wilcox separately fired superheater using oil fuel, at the Delray Station of Detroit Edison Co., is designed to heat 90,000 lb. of steam per hr. from 700 to 1100° F., at 400 lb. per sq. in. pressure. Superheater consists of three sections in series. The upper section is a convection section of carbon steel tubes and has a counterflow of steam and gases. The intermediate section also is a convection section, with special alloy tubes and parallel flow of steam and gases. The lower section is a radiant-heat section. It consists of refractory-covered special alloy steel tubes, which form sidewalls and floor of the furnace.

REHEATER SUPERHEATERS may be placed in the main boiler setting or may be separately fired. ' They must offer low frictional resistance to flow of steam and provide uniform flow through all tubes. Counterflow of steam and gases make possible a higher degree of superheat with a given amount of heating surface. Required steam temperatures may be obtained by control of firing rates in separately fired types, by desuperheater equipment, by by-passing gases, or by using a live steam reheater in series with the main flue gas reheater, in integral type units.

TUBE STRESSES. USE OF ALLOY TUBES.—In determining thickness of superheater tubes, stresses due to temperature gradient through tube walls, and creep stress also must be considered. In superheater tubes, used at high temperatures, creep stress should govern. Ordinary carbon steel tubes should not be used at metal temperatures exceeding 900–950° F., depending on the pressure, as the actual working stresses in such tubes may exceed limiting creep stress.

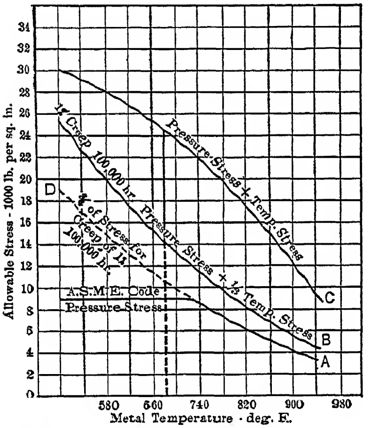


Fig. 5. Allowable Stresses in Seamless Steel Superheater Tubes. Low-carbon Tubes, A.S.M.E. Code, Grade A

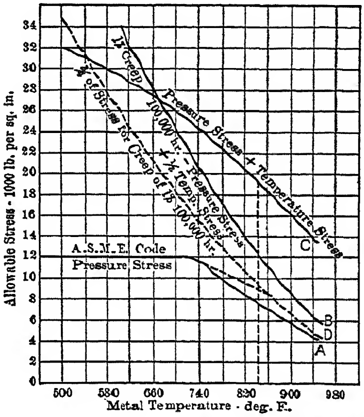


Fig. 6. Allowable Stresses in Seamless Steel Superheater Tubes. Medium-carbon Tubes, A.S.M.E. Code, Grade C

A number of superheaters now (1935) supply steam at temperatures of 850° F. and more. In these, special alloy steel tubes are used in the superheater where metal temperatures are highest. Satisfactory alloys for superheater tubes, where temperature on inside of tube exceeds 950° F. are:

Alloy	C	Mn	Si	Cr	Mo	
4–6 Cr, 0.5 Mo....	0.15	0.35	0.25	4.0–6.0	0.4–0.6	
KA2S...........	0.07 max.	0.60 max.	0.75 max.	16.5–20.0	....	7.0–10.5

The 4–6 Cr, 0.5 Mo tube is satisfactory for metal temperatures at the inside surface up to 1050° F, and the KA2S for temperatures up to 1150–1200° F. Thermal conductivity of KA2S alloy is about 0.4 that of carbon steel, and its thermal expansion is about 30% greater at 900° F. and 40% greater at 500° F. Temperature stress for a given rate of heat absorption and tube thickness is about 3.5 times that of carbon steel. Increase of creep stress permissible at 900° F. is in about the same ratio as that of carbon steel. KA2S tubes should not be expanded into carbon steel headers or drums, but should be resistance-welded to short lengths of carbon steel tubes, which then are expanded into tube seats. Thermal conductivity of 4–6 Cr, 0.5 Mo alloy is slightly less than that of carbon steel, and its thermal expansion is about 0.9 that of carbon steel at 900° F. and 0.94 at 500° F. Higher creep stresses are permissible than with carbon steel tubes, but the resistance to oxidation is no better. Fig. 4 gives creep stresses of KA2S and 4–6 Cr, 0.5 Mo tubes.

Figs. 5 and 6 give limiting stresses for superheater tubes of grade A low-carbon steel, specification S-17 of A.S.M.E. Boiler Code, and of grade C medium carbon steel, 60,000 lb. per sq. in. minimum tensile strength. The lower curves correspond to values allowed by the

Code (see Table 10, p. 6–37). Required thickness for pressure stress only may be calculated on the basis of stress values in the lower curves by the formula $t = Pd/2S$, where t = wall thickness, in., P = pressure, lb. per sq. in.; d = inside diam., in.; S = permissible stress at temperature of inner surface of tube (curves A, Figs. 5 and 6). (Pressure stress + $^1/_3$ temperature stress for elastic range) should not exceed stress corresponding to 1% creep in 100,000 hours (Curve B, Figs. 5 and 6); and (pressure stress + full temperature stress) should not exceed limit of curve corresponding to 0.9 yield point. In Figs. 5 and 6, curve C governs for temperatures to the left of the heavy vertical line, and curve B governs for temperatures to the right of it. In Fig. 6, the Code values (Table 10) are lower than curve D, and the dotted line joining curves A and D should be used in preference to curve A for temperatures of 750° to 950° F. In no case should thickness of tube be less than the values in Table 10. Fig. 7 shows required thickness of superheater tubes for various pressures and temperatures. It indicates that Code thickness (Table 10) should be increased for pressures of 720 lb. with temperature above 900° F., and for pressures of 400 lb. with temperature above 950° F.

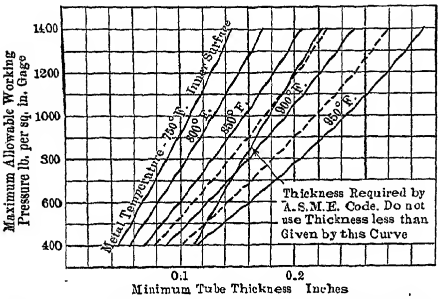


FIG. 7. Required Thickness for Superheater Tubes for Pressure Stress

EFFECT OF FEEDWATER TEMPERATURE.—For the same fuel burning rate, superheat increases with decrease in temperature of feedwater. Gas weights and temperatures entering superheater will not change, but steam weight through superheater will be less, since more heat is required to evaporate each pound of water. For the same capacity, degrees of superheat will vary, approximately, in direct proportion to B.t.u. absorbed per pound of steam in boiler and superheater (and economizer, if any), for a change of not more than 150° F. in feed temperature. Excessive superheat may result if it is necessary to cut out the feedwater heater and supply the boiler with cold feedwater.

SETTING OF SUPERHEATER SAFETY VALVES.—In all superheaters, a certain steam velocity through the tubes is necessary for protection against burning. Superheater safety valves, therefore, should be set to operate at a pressure below that of the boiler (saturated) safety valves, to insure that superheater valves blow first. Should safety valves on boiler and superheater be set for the same pressure, boiler safety valves would blow first, and the superheater would have little or no flow of steam, with consequent danger of tube burning.

REGULATION OF AMOUNT OF SUPERHEAT.—Higher steam temperatures demand some means of automatically regulating amount of superheat. Four methods are available: 1. Temperature of superheated steam is reduced to required temperature after it leaves superheater. This has been accomplished by: *a*, a controlled water spray in inlet or discharge pipe; *b*, placing tubes in the water space of steam and water drum, through which the part of the superheated steam passes, and after being cooled, is mixed with the rest of the superheated steam; *c*, by-passing part of the total superheated steam generated through cooling tubes, submerged in water from the steam and water drum. 2. Only part of the heated gases pass over the superheater; the quantity by-passed around it is regulated by dampers, or by using a divided furnace. 3. Interstage desuperheating, where two superheaters are placed in series; temperature of superheated steam entering second superheater is regulated to maintain a constant final temperature. 4. Some superheated steam is recirculated back to the steam and water drums through a desuperheating zone; velocity of the steam through the superheater thus is increased without increasing delivered steam; this method requires a compressor to recirculate the steam. Present (1935) practice is limited to desuperheating and to by-passing of the gases in conventional boiler designs. In the latter case, design should be such that the damper is not exposed to excessive temperature. In method 1 the spray pipe is simply a nozzle in a steam main or in a mingling chamber. Fig. 8 shows a submerged type desuperheater, which can be used either interstage or at superheater outlet. It is connected directly to the boiler and designed for natural circulation on the water side, steam generated being discharged behind the separator in the steam and water drum. A butterfly valve regulates the quantity of superheated steam passing through the submerged tubes.

AMOUNT OF HEATING SURFACE in superheaters varies widely for each installation. It depends on several variables, including temperature, pressure, load, location of superheating surface relative to boiler heating surface, amount of superheat required, temperature of gases, and heat transfer rate. High temperatures and pressures require more heating surface than do low temperatures and pressures, because of the greater ratio of heat in the superheat to the latent heat per pound of steam at high pressures. Quantity of steam and amount of superheat required are determined both by process and power requirements. Location of superheater may be selected arbitrarily in accordance with good engineering practice, and should not involve mechanical or operating difficulties (cleaning, excessive tube temperatures, etc.). Temperatures of gases will depend on such factors as type of fuel, combustion conditions, amount of water-tube surface or water screen swept by the gases before they strike superheater, etc. Conductance will vary as temperature and velocity of gases. The gases should be limited to a temperature that will not produce excessive superheater tube temperatures. This may be accomplished by the addition of water tube surface ahead of the superheater. Velocity of gases is limited by the allowable draft loss. The nearer the superheater is to the furnace, the more important is the effect of radiant heat as a factor in determining conductance.

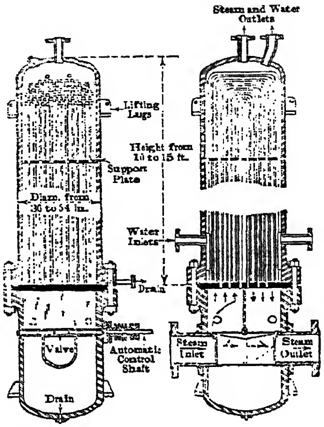


FIG. 8.

SUPERHEATER SURFACE REQUIRED.—The relation between heat absorbed by steam in the superheater and that given up by gases of combustion, radiation included, is

$$AUT = \quad - t_2); \; AUT = W_1 c_1(T_1$$

where A = area of superheater surface, sq. ft., U = conductance, B.t.u. per hr. per sq. ft. per deg. mean temperature difference; T = logarithmic mean temperature difference between steam and gases of combustion; W, W_1 = respectively, weight of gases of combustion and weight of steam passing through superheater per hr.; c, c_1 = respectively, mean specific heat of gases of combustion and of steam; t_1, t_2 = respectively, gas temperature entering and leaving superheater, deg. F.; T_1, T_2 = respectively, temperature of superheated steam and saturated steam.

Surface required is $A = H/UT$, where H = heat to be absorbed by superheater, B.t.u. per hr. = $(h_2 - h_1)$; h_1, h_2 = respectively, total heat of superheated and saturated steam, B.t.u. (see Steam Tables).

$$T = \frac{\text{(Maximum temperature difference)} - \text{(Minimum temperature difference)}}{\log_e \{\text{(Maximum temperature difference)}/\text{(Minimum temperature difference)}\}}$$

Conductance U varies with temperature of gases, gas velocity, steam velocity, tube size and spacing, surface cleanliness, and other variables of minor importance. Approximate values for 2-in. tube elements, varying with position of superheater are

Mass flow.................	2000	3000	4000
U, interdeck superheater....	7–7.5	8.5–9.5	10–11.5
U, overdeck superheater.....	6–6.5	7.5–8.5	9–10

Mass flow is defined as lb. of gas or steam per hr. per sq. ft. of minimum free flow area.

Temperature drop through the steam film in superheater tubes for various rates of heat absorption and steam mass flows is given in Fig. 9. Temperature drop $t_d = H/U_f$; $U_f = 0.95\,G/1000$, where U_f = steam film conductance, B.t.u. per hr. per sq. ft. inside tube surface; G = mass steam flow; other notation as above. The curves are based on a specific heat of steam of 0.55 and a friction factor of 0.004. For any other specific heat or friction factor, temperature drop will vary inversely as the square root. Fig. 10 shows the relation between mass flow, tube thickness and rate of steam generation for 2-in. tubes on 3-in. centers.

Table 1 gives superheater surface required for various degrees of superheat under the

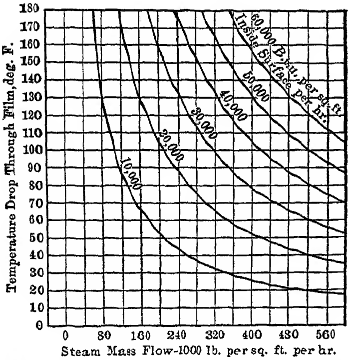


Fig. 9. Effect of Steam Mass Flow and Heat Absorption on Temperature Drop through Steam Film in Superheater Tubes

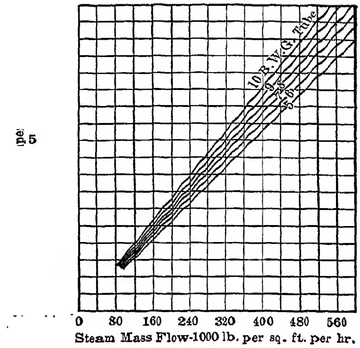


Fig. 10. Steam Mass Flow in 2-in. Superheater Tubes, Spaced 3-in. Centers

conditions specified, the superheater having 2-in. continuous tube elements on 3-in. centers. The assumed section is 94 tubes wide with elements approximately 11 ft. long.

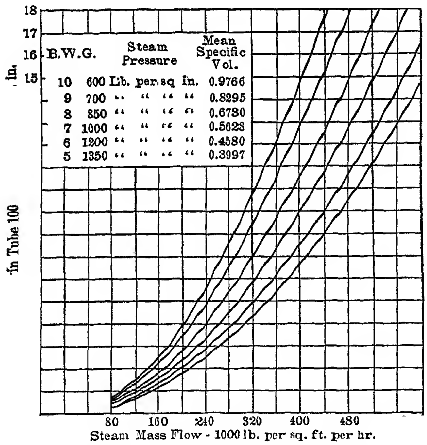


Fig. 11. Pressure Drop in 2-in. Straight Superheater Tubes. Steam Temperature 800° F.

In the overdeck position the superheater is parallel flow above the first pass of the boiler, and counterflow above the second pass, or *vice versa.* Hence, the average of the two logarithmic mean temperature differences is used. Surface required overdeck is much greater than interdeck for the same number of water tubes in the bank, due both to lower conductance, and to lower mean temperature difference, the latter having the greater significance. A unit with overdeck superheater may cost less than one with interdeck superheater, as the latter usually requires a double-deck boiler and increased head room. Also, it requires more time and care to bring a boiler on the line from cold with an interdeck than with an overdeck superheater for the same number of tubes in the boiler bank. The position of the interdeck superheater necessitates lower rates of heat input, so that the gas temperature in this location will not cause excessive superheater tube temperatures during the period of zero steam flow.

FUEL REQUIRED FOR SUPERHEATING.—When combined efficiency of boiler and superheater equals that of boiler alone, fuel required for superheating the same

Table 1.—Superheater Surface Required for Various Superheats

Conditions: Capacity, 300,000 lb. steam per hr.; pressure, 410 lb. per sq. in. at superheat outlet; gas weight, 350,000 lb. per hr.; gas mass flow, 4000 lb. per hr. per sq. ft. of minimum free flow area through superheater.

Interdeck. Entering Gas, 1950° F.			Overdeck. Entering Gas, 1300° F.	
Superheat, deg. F.	Sq. ft., Parallel Flow	Sq. ft., Counterflow	Superheat, deg. F.	Sq. ft.
100	1400	1396	100	3220
200	3032	2962	200	8200
300	5275	4940	....	....
400	8835	7550	....	....

weight of steam is in direct proportion to the additional B.t.u. absorbed per pound of steam. The increase in fuel required for various degrees of superheat at 410 lb. per sq. in. superheated steam pressure, and 200° F. initial feedwater temperature is

Superheat, deg. F.	25	50	75	100	150	200
Additional fuel, percent	1.82	3.56	5.17	6.70	9.54	12.37

The potential energy of each pound of steam is increased by the same percentage, and for a given amount of work the steam weight required is proportionately less. From the standpoint of heat units absorbed, which is the measure of capacity, no additional fuel is required for a given absorption whether superheated or saturated, provided combined efficiency of boiler and superheater is the same as for boiler alone, but steam weight is reduced, due to the greater heat absorption per pound of superheated steam.

PRESSURE DROP IN SUPERHEATER TUBES can be expressed by

$$p = (400\, fV/D) \times (G/100{,}000)^2$$

where p = pressure drop, lb. per sq. in., f = friction factor; V = specific volume of steam, cu. ft. per lb.; D = inside diam. of tube, in.; G = steam mass flow, lb. per hr. per sq. ft. of free flow area. Bends will increase p from 50 to 100%, depending on their number per 100 ft. of tube and their radius. Additional pressure drop for each 90-deg. bend, expressed in equivalent feet of straight pipe is, approximately,

Radius of bend, tube diameters	1	2	3
Equivalent straight pipe, ft.	6.2	4.3	3.3

Pressure drops of 20–25 lb. per sq. in. per 100 ft. of tube are not excessive. Fig. 11 shows pressures drops in 2-in. tubes of various thicknesses.

2. ECONOMIZERS

The primary function of an economizer is to utilize part of the heat in flue gas to heat feedwater. There is always a most economical temperature of flue gases, above and below which steam costs are greater. Excessive reductions in flue gas temperature so increase capital investment and fixed charges on the increased heating surface necessary as to offset any gain in efficiency, and corresponding fuel economy.

GENERAL TYPES OF ECONOMIZERS.—Economizers are either of wrought steel or cast iron. Present-day practice with wrought steel economizers, favors using an economizer with each boiler. The individual economizer operates when the boiler operates, has no by-pass flues, and air leakage is a minimum. Cast-iron economizers generally utilize flue gas from two or more boilers, and have by-pass flues.

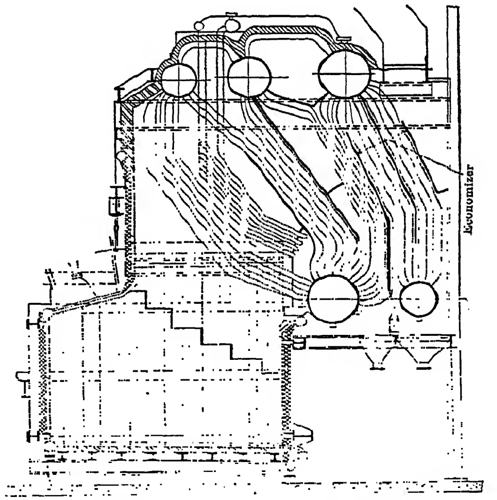


FIG. 12. Stirling Boiler Equipped with Integral Economizer

Economizers are either integral or independent. Integral economizers are located in the boiler setting, and connected directly to the boiler. See Fig. 12. Independent economizers usually are located above, or behind, the boiler. Connection to boiler drum may be through a single connection and valve, or directly through tubes, with no collecting header.

TREND OF ECONOMIZER DESIGN AND PRACTICE.—Early economizers were cast iron, including both tubes and headers. The tubes were set vertically in a flue between boiler and stack, and by-passes were common. Water and gas velocities were low, with corresponding low conductances. In smaller sizes, water flowed in parallel through all tubes. Larger sizes were in three or more sections, water flowing in parallel through tubes of each section, but in series through the sections. Later, steel tubes ex-

panded into headers ("boiler section type") tended toward large weight and space reduction. See Fig. 13.

Modern economizer practice dates from 1920, when means were developed for removing air and contained gases from water before it entered the economizer. Effective counterflow of gas and water, and long flow-paths for water are provided. Tubes usually are of smaller diameter than the boiler tubes. Gas velocities are high, and limited only by draft loss and power expense for draft fans. Water velocities, high enough to insure good heat transfer rates, and long flow paths have eliminated local convection circulation of water. Flow is forced one way, non-circulating, limited only by allowable excess pump power. Horizontal tubes, connected by a series of hair-pin bends, forged return bends, or combinations of both, eliminate stresses due to variation in length between hot tubes at the top and cooler tubes at the bottom. The upflow of water causes steam formed due to lack of water flow, especially in stand-by service, to rise to the top. Hot water will displace the steam, which tends to eliminate possibility of collapse of steam in pockets by condensation, with resultant water hammer. Wrought steel return bends with integral flanges, joining straight, and single- or multi-looped tubes, result in a flexible arrangement, without restraint on metal expansion, and obviate the possibility of cross connection between the several series flows of water. The latest (1935) horizontal wrought steel return bend economizer eliminates collecting headers by leading each series tube directly to the boiler drum. See Fig. 14.

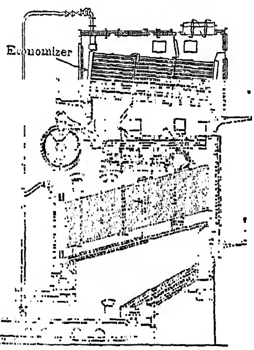


FIG. 13. Babcock & Wilcox Section Type Economizer

STEAMING ECONOMIZER.—A steel tube economizer designed to steam at higher rates of output, is known as a steaming economizer. In this economizer, each series tube is led directly to the boiler drum. Provision also is made to recirculate water when starting up. The steaming economizer is most useful in high capacity installations, where the boiler practically amounts to a water screen protecting the superheater, and boiler exit gas temperatures are high.

FACTORS AFFECTING EFFICIENCY.—The larger the economizer surface, the higher the gas velocity and temperature, and the lower the feedwater temperature, the greater will be the increase in efficiency effected by the addition of an economizer to a given boiler. Counterflow of gas and water gives greater heat absorption with less surface than does parallel flow.

With the exception of steaming economizers, economizer heating surface should be designed to limit maximum feedwater temperature, under any operating condition, to at least 40° F. below saturation temperature in the boiler. More surface than this probably would cause water in the economizer to reach the steaming point should boiler feed be cut off. Resumption of feed to the boiler would project water and steam from some forms of economizer into the boiler, partially emptying the economizer. On refilling economizer, temperature strains would set up, resulting in leakage, or water hammer might occur, causing serious trouble.

Economizer surface installed will vary with load conditions, feed temperature, fuel, extent to which gases are cooled, ratio of gas weight to water weight, etc. In non-steaming economizers, and where no air heaters are used, the surface usually is 40 to 60% of boiler heating surface. If air heaters are included, the surface usually approximates 25 to 30%. The surface of integral economizers ranges from 25 to 50% of boiler heating surface. The surface in steaming economizers ranges from 140 to 280% of boiler heating surface.

Temperature of gases entering economizer will vary with different types of boilers, operating loads, combustion conditions, etc. The temperature to which the gases can be cooled, in passing through the economizer will vary with amount of heat to be absorbed by feedwater, entering feed temperature, dew point of the gas (with low inlet feed temperture this is a limiting factor because of external corrosion), and economical exit temperature below which any gain in efficiency would be offset by increased cost.

For a definite amount of economizer surface and set temperature ranges, increase in gas velocity increases conductance. Gas velocity for a fixed amount and arrangement of surface is limited by allowable draft loss. With counterflow of gas and water, the mean temperature difference between gases and water is greater for economizer surface than for boiler surface beyond the superheater. Thus, less surface is necessary for a given degree of gas cooling, even if the conductance per degree difference is the same for both, and it is

practicable to design the economizer for higher conductance to further reduce the amount of economizer heating surface.

Gas velocities through cast-iron economizers usually are low, resulting in low conductance. Lack of facilities to properly keep cast-iron tubes externally clean also lowers conductance. In the typical cast-iron economizer, with a gas mass flow approximating 1500 lb per hr. per sq. ft. of gas passage area, a conductance of 2.5 to 3.5 B.t.u. per hr. per sq. ft. per deg. F. mean temperature difference may be expected. Increasing gas mass flow to 3000 lb. increases conductance to 5 to 6 B.t.u. Steel tube counterflow economizers utilize high gas velocities. At higher boiler steaming rates, conductance in a steel tube economizer will be from 12 to 13.5 B.t.u. per hr. per sq. ft. per deg. F. mean temperature difference for a gas mass flow of approximately 6500 lb. per hr. Draft loss through steel tube economizers under these conditions depends primarily on the gas mass flow and the height of the economizer. Feedwater temperatures will vary widely, depending on the source, *i.e.*, raw water, condenser hotwell, or feedwater heater.

DETERMINATION OF SURFACE REQUIRED.—The relation between heat absorbed by the water, and heat given up by the gases is:

$$AUT = Wc(t_1 - t_2);\ AUT = W_1 c_1(T_1 - T_2)$$

where A = area of economizer surface, sq. ft.; U= conductance, B.t.u. per hr. per sq. ft. per deg. mean temperature difference; T =logarithmic mean temperature difference between water and gases of combustion (see p. 6–47); W =weight of gases passing through economizer per hr; W_1=weight of water passing through economizer per hr.; t_1, t_2 = temperatures of entering and leaving gases, respectively, deg. F.; c and c_1 = mean specific heats of gas and water, respectively; T_1, T_2 = temperatures of leaving and entering water, respectively, deg. F. Surface required is either

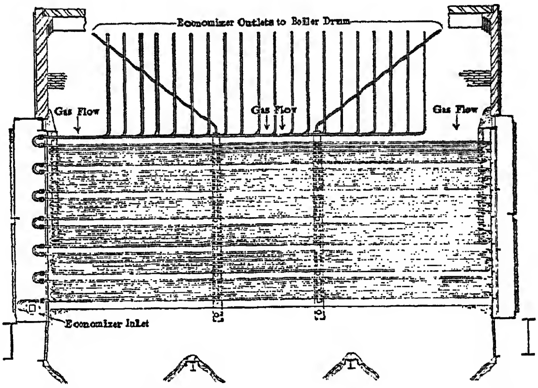


FIG. 14. Babcock & Wilcox Wrought Steel Continuous-tube Return Bend Economizer

$$A = H_w/UT \quad \text{or} \quad A$$

where H_w, H_g = heat absorbed by water or released by gas, respectively, B.t.u. per hr.; other notation as above.

Economizer surface required for various temperature drops in a 2-in. steel tube economizer is as follows, for conditions specified below. The advantage of counterflow over parallel flow is apparent.

Gas temperature drop, deg. F.	100	200	300	400	500
Sq. ft. surface, counterflow	1,143	2,592	4,440	7,020	10,920
Approximate rows high	5	12	20	31	48
Sq. ft. surface, parallel flow	1,145	2,621	4,630	7,820	15,350
Approximate rows high	5	12	21	35	68

Conditions: Capacity, 300,000 lb. water entering at 200° F.; gas mass flow, 6200 lb. per hr. per sq. ft. of minimum free flow area; entering temperature of gas, 900° F.; gas weight, 350,000 lb. per hr.; assumed economizer section 18 tubes wide, 24 ft. long; tube spacing, 3 1/2 in. centers.

Approximate draft loss for the 48 high counterflow economizer under above conditions is 3.25 in. of water. Increasing economizer section from 18 to 22 rows wide, increases required surface by 12.6% to 12,300 ft., but draft loss is reduced 35% to approximately 2.1 in. of water. Water pressure drop through economizer also is reduced about 35%. The increase in fixed charges and decrease in fan and pump power must be considered in order to determine which unit is more economical.

CORROSION AND CLEANING.—**External Corrosion** is due to condensation of the moisture content of the gases. Condensation begins when gases reach the dew point. Gases in contact with tubes reach the dew point sooner than gases in the main stream.

Corrosion, particularly with high-sulphur fuels, usually is most noticeable at the cold end of the economizer. Good practice dictates tube temperatures not lower than 180° F. for steel tube economizers, and higher if sulphur is excessive. Cast-iron economizers were first used because of their low corrosion, but the greater safety and higher conductance of the steel tube economizer more than offsets its greater liability to corrosion.

Internal Corrosion will be negligible if oxygen or air in solution in feedwater be eliminated. In steel tube economizers, maximum allowable oxygen content is 0.025 c.c. per liter for 180° F. feed or above. This feed temperature, with oxygen limited as above, prevents internal corrosion, and tube temperatures also are high enough to prevent external corrosion, unless sulphur in the flue gas is excessive.

Cleaning.—Dust or soot deposits on cast-iron tubes usually are removed by mechanically-operated scrapers. Soot blowers are used extensively for cleaning the gas side of steel tube economizers. A method, sometimes used consists in washing the tubes by water, delivered over the top of the tube bank, running down by gravity, flushing all outside deposits.

DEAERATION AND DEACTIVATION OF FEEDWATER.—To prevent internal corrosion, dissolved gases in feedwater must be removed before it is fed to economizer or boiler. Deaeration, or liberation of gases by boiling the water, and deactivation, or absorption of gases by a chemical reagent, such as iron turnings, are processes effecting this result.

In a typical deaerator, the water is heated to saturation temperature, thus removing gases from solution. Further agitation, which may be effected by steam jets or by ebullition of the liquid itself, separates the entrained gases, which then are vented from the apparatus. The Elliott deaerator, operating with an open heater temperature of 210° F., and a separator temperature of 188° F., will deliver water with a gas content not exceeding 0.02 c.c. per liter.

In the usual type of deactivator, the water, heated by external means, passes to a closed chamber filled with iron turnings, or thin perforated plates, of any rapid deactivating material. Oxygen and free gases are absorbed, the rate depending primarily on feedwater temperature. A combination deaerator and deactivator often is used where feed temperature is below 200° F. and water free of gases is desired.

An open feedwater heater, properly vented, with feed temperature of 210–212° F. will reduce the oxygen content to approximately 0.2 c.c. per liter.

3. AIR HEATERS

The purpose of the air heater is to extract additional heat from the gases of combustion leaving the economizer, and to raise the temperature of air used for combustion, for drying or other purposes. Gas exit temperatures from the economizer may be excessive if inlet feedwater temperatures are high, and the use of an air heater is necessary for the most economical production of steam. The fact that considerable duct length for gases and air is required, and that air leakage or gas infiltration, if excessive, is harmful, impose problems of design and arrangement that require considerable study.

TYPES OF AIR HEATERS.—Air heaters may be classed as tubular, plate and regenerative.

Tubular Air Heater.—Gas usually flows through the tubes and air passes across and in the opposite direction outside of them. Fig. 15 shows a tubular air heater designed for use with boilers fired by any fuel. A nest of steel tubes is expanded into upper and lower tube plates, and enclosed in a rectangular steel casing, which is the air flue. Provision is made for the expansion of tubes with reference to the casing. Inlet and outlet openings for air and gas are provided. Deflecting baffles are arranged to distribute air evenly over the outside of the tubes with minimum resistance. This type of air heater is inherently tight and maintains separation of air and gas streams.

Plate Type Air Heaters consist essentially of a casing, divided into a series of thin narrow compartments by metal partitions. Gas and air pass through alternate spaces in opposite directions. The large area of joints imposes leakage problems of considerable magnitude.

Regenerative Type Air Heaters consist of rotating plate elements that alternately absorb heat from the gas and transfer it to the air. The elements are not conducting partitions, but are mediums for temporary heat storage, and must move from the gas duct to the air passage.

ADVANTAGES OF USING HEATED AIR.—The heat abstracted from the gases and returned to the system is not absorbed by the working fluid (water or steam) but by air for the furnace. The economies effected are due to the release of this heat in the

furnace and its absorption by the heating surface. The increased temperature of combustion air accelerates combustion, and a smaller furnace volume may be used, or for a given volume, greater capacity for heat release is available. The heat released per pound of fuel by combustion, and the resulting temperature rise being the same under identical conditions, furnace temperatures will be higher with hot air than with cold. Raising furnace temperature will increase the difference between the temperature of hot solids and gases in the furnace and that of surfaces exposed to the furnace. The absorption of furnace heat and the temperature of furnace exit gases are increased over what they would be without preheated air, resulting in increased heat-absorbing capacity per square foot of gas-swept surface between furnace and air heater. Efficiency also increases because of lower unconsumed combustible losses.

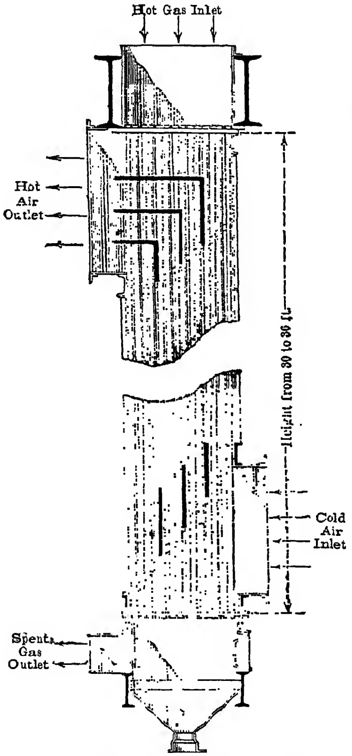


Fig. 15. Babcock & Wilcox Tubular Air Heater

HEATING SURFACE.—Air heater surface costs much less per square foot than boiler or economizer surface. The air heater is not subject to steam pressure, and can be of lighter construction, with less regard to trouble from leaky joints. The maximum difference in pressure between air and gas passages, resulting from induced draft on the gas side and forced draft pressure on the air side, normally does not exceed 0.75 lb. per sq. in. Air heater surface installed, in connection with boilers and economizers, usually is from 80 to 150% of boiler heating surface. Without economizers it may be as high as 300%. With steaming economizers, the maximum amount of air heater surface may approximate 900% of boiler heating surface.

TRANSFER RATE.—The rate of heat transfer of air heater surface is less than that of economizer or boiler surface, because air offers greater thermal resistance, on the receiving side of the surface, than does the boiler. While form of gas and air passages has some effect on the rate of heat transfer, it depends primarily on gas and air velocities. Draft loss increases with velocity. Hence velocity, and also transfer rate, is limited by the allowable draft loss through the heater. Heat transfer rates for air heaters vary between 2.5 and 4 B.t.u. per hr. per sq. ft. per deg. mean temperature difference between gas and air. For every installation there is some velocity beyond which the power necessary to provide draft offsets any gain in efficiency effected by the heater.

ALLOWABLE AIR TEMPERATURES.—In air heaters, built of ordinary carbon steel, metal temperature should not exceed 900° F. This limit can be raised by using alloy steels, as KA2, or by calorizing the carbon steels. Temperatures of preheated air for combustion vary widely. For fuels burned in suspension (pulverized coal, oil, gas) maximum air temperatures used are about 600° F., but the extreme temperature, with a properly cooled furnace, is that which the metal of the air heater can withstand. In pulverized coal firing, secondary air may approach 600° F. The temperature of primary air may be limited by mills or burners, and usually approximates 225° F., depending primarily on the moisture content of the coal. In stoker firing, maximum allowable air temperature varies from 350° F. with chain-grate stokers to 450–500° F. with underfeed stokers.

HEAT ABSTRACTED BY AIR HEATER.—The exit gas temperature from the air heater usually chosen is that which is most economical for operation of the particular unit. The exit air temperature is based on the requirements or limitations of the furnace and burner or stoker. With constant inlet air temperature, final temperature limits for exit air and exit gas determine the amount of heat that can be abstracted and returned.

and the temperature at which gases must enter the air heater. The position of the air heater, between furnace and stack, is fixed by inlet gas temperature, which in turn depends on the heating surface in advance of the air heater. This limitation of gas temperature at the air heater entrance, often renders advisable a small economizer between air heater and boiler as a means of adjusting gas temperature, to avoid additional heating surface in the boiler. The weight of the gas passing through an air heater is greater than the weight of the air. This causes a loss which increases with the temperature to which the air is heated. The loss is reduced by the use of an economizer, and there is a certain ratio of economizer and air heater surface that will give the best results.

OVERALL EFFICIENCY VS. LOAD VARIATIONS.—Load variations cause variations in exit gas temperature. The more gas temperature is lowered by the air heater the less it will vary with load. Overall efficiency of boilers with air heaters will vary less with load than boilers without them. With widely variable plant load this tends to keep average efficiency high. Air temperature necessarily increases with load.

CORROSION AND CLEANING.—Corrosion occurs only on the gas side of air heaters, and may be limited by keeping gas temperature above the dew point. With high-sulphur fuels, the temperature limit must be raised, to avoid all possibility of the condensation of moisture containing SO_2 on the heater surfaces.

Air leakage into the stream of hot gases cools the gases locally, impairs heating capacity, and necessitates a compensating increase in fan power to handle the excess flue gas. Leakage is minimized by good design, and is least for tubular types.

Air heaters in which gases pass over the tubes may be cleaned by mechanical soot blowers, or washed with water. A turbine cleaner is used with those heaters in which gas passes through the tubes.

CALCULATION OF SURFACE REQUIRED.—The relation between heat absorbed by the air and heat given up by gases of combustion is

$$AUT = Wc(t_1 - t_2) \qquad \text{or} \qquad AUT = W_1 c_1(T_1 - T_2)$$
$$A = H_a/UT \qquad \text{or} \qquad A = H_g/UT$$

where A = area of air heater surface, sq. ft.; U = conductance, B.t.u. per hr. per sq. ft. per degree mean temperature difference; T = logarithmic mean temperature difference between air and gases of combustion; W, W_1 = respectively, weight of gases and air passing through heater, lb. per hr.; H_a, H_g = respectively, heat absorbed by air and released by gases, B.t.u. per hr.; c, c_1 = respectively, mean specific heats of gases and air; t_1, t_2 = respectively, temperatures of entering and leaving gases, deg. F.; T_1, T_2 = respectively, temperatures of leaving and entering air, deg. F.

$$T = \frac{(\text{Maximum temperature difference}) - (\text{Minimum temperature difference})}{\log_e\{(\text{Maximum temperature difference})/(\text{Minimum temperature difference})\}}$$

Since the gas outlet from a boiler or economizer usually is the full width of the setting, and floor space is at a premium, air heaters generally are equal in width to the setting, and narrow in direction of the depth. An additional advantage is the ease of fluid entrance, and in a tubular air heater, as Fig. 15, air resistance is low by reason of fewer number of tube rows crossed. The relation of width to length determines the proportions of the surface in effective counterflow relations.

Air heater surface required for various temperature drops for a 2 1/2-in. O.D. tubular counterflow air heater containing 1500 tubes, under the operating conditions stated are:

Gas temperature drop, deg. F.	100	200	300	400
Sq. ft. external tube surface	6,010	15,280	31,750	70,500
Approximate length of tubes, ft.	6	16	32	72
Exit air temperature, deg. F.	205	329	451	573

Conditions: Gas, 350,000 lb. per hr., entering at 700° F.; gas mass flow, 7840 lb. per hr. per sq. ft. of free flow area. Air, 300,000 lb. per hr., entering at 80° F.; air mass flow, 6730 lb. per hr. per sq. ft. of free flow area.

Because of the excessive tube length of 72 ft. required for a 400° F. drop in gas temperature, the heater would be made in two sections in series, each with tubes 36 ft. long, with common upper and lower tube sheets. Gas flows down through one section and up through the other. A central partition extending from the upper tube sheet to the common passage between sections separates the air in the two sections. The counterflow principle is preserved. Gas draft loss for the two sections under the above conditions, is approximately 4.8 in. of water, and air resistance approximately 4.4 in. Increasing the section of the heater will reduce draft loss almost directly as the square of the number of tubes, since the length of the heater will be decreased only slightly by reduced air and gas conductances.

UTILIZATION OF WASTE HEAT

Waste gases from many types of industrial furnaces are discharged at high temperature and may profitably be utilized for generating steam in waste heat boilers. Such installations may be classified generally as: 1. Those operating with practically constant gas temperatures and weights. 2. Those in which, due to furnace operation cycles, gas temperatures and weights vary over a considerable range. See Table 1 for types of industrial furnaces supplying gas to waste heat boilers, and the gas temperatures in each class of work, according to former standard practice. Many furnaces of recent design for these and other purposes operate with lower exit gas temperatures, due to improvements in design, including air preheating by recuperators and regenerators for high temperatures. For these special cases, reference should be made to other sources. Temperatures shown represent average values over a complete cycle of furnace operation.

Exhaust gases from internal combustion engines also may be sources of heat for steam boilers. No generalization as to gas temperatures is feasible, since it depends on mean pressures in the engine cylinder. Mean pressure varies widely, and is lower for 2-cycle engines with scavenging air than for 4-cycle engines.

TREND IN WASTE HEAT BOILER DESIGN.—Installations prior to 1915 were for use with high temperature gases, and depended on natural draft. Non-interference with operation of the furnace was the prime requisite in design. As ample draft at the industrial furnace was the only indication of non-interference, waste heat boilers receiving gases from the furnace were installed to operate with low gas velocities, and consequent minimum draft resistance. Baffles usually were omitted, and high exit gas temperatures from the boiler were common and considered desirable for increasing draft produced by the chimney. Operation of waste heat boilers with any but high temperature waste gases was not economically practicable. See p. 6-09 for typical designs of waste heat boilers.

Since 1915, the use of mechanical draft has made available gas velocities much higher than were common in former practice, exceeding those usually found in direct-fired boilers. High gas velocities result in high rates of heat transfer by convection. This is desirable in waste heat boilers to offset lack of heat transferred by radiation. That part of the boiler exposed to a furnace temperature of 2800° F. (direct-fired) will absorb approximately 14 times more heat by radiation than will a boiler exposed to a furnace temperature of 1200° F. (waste heat practice). With entering waste gas temperatures of 1200–1300° F., and the boiler developing rated capacity, average velocity is about 2500 ft. per min. through the setting, or almost four times the velocity needed to develop rated capacity in a similar direct-fired boiler. If initial gas temperature is lowered, gas velocity must be increased to maintain the same heat transfer rate. With a definite initial gas temperature, boiler capacity is limited by the maximum gas velocity that can be obtained with available gas producing equipment.

High draft loss through modern (1935) waste heat boilers, together with reduction in exit gas temperatures from the boiler, preclude dependence on natural draft stacks alone, and require installation of induced draft fans. The adoption of fans, not used by furnace operators before 1915, has given better control of draft at the primary furnace. With excess draft usually available, furnace operations have been improved, and, in many cases, industrial output increased. Turbine-driven fans, primarily used because most of the power required to drive them was recovered from heat in the exhaust steam, are

Table 1.—Temperatures of Waste Gas from Industrial Furnaces

Type of Primary Furnace	Temperature, deg. F.
Nickel refining furnace	2500–3000
Beehive coke ovens	1950–2300
Black liquor furnace	1800–2000
Zinc refining furnace	1700–2000
Heating furnace	1700–1900*
Copper reverberatory furnace	1650–2000
Copper refining furnace	1450†
Cement kiln (Dry process)	1150–1350
Cement kiln (Wet process)	800–1100
Open-hearth steel furnace (Producer-gas-fired)	1200–1300
Open-hearth steel furnace (Oil, tar, or natural gas)	800–1100
Gas benches	1050–1150
Oil stills	900–1000
Glass tanks	800–1000

* During operating periods. With furnace kept hot but heating no material, average temperature 1000–1100° F.

† Average over 36-hr. cycle; range 500–2100° F.

susceptible to changes in steam pressure, which affects the flow of gases through the boiler. The tendency in recent years has been to use electric motors for the fan drive.

Waste heat boilers may be of the water-tube or fire-tube type. Consideration must be given to the avoidance of dirt accumulations, and for this reason, vertical tubes predominate in fire-tube boilers because of their greater ability to shed dirt. High gas velocities are utilized to scour the tubes, this being at the expense of draft loss. Waste gases at pressures of several atmospheres sometimes are used. Casing then must be designed to be gas tight, to avoid the hazards of inflammable or toxic gases.

USE OF ECONOMIZERS WITH WASTE HEAT BOILERS.—Economizer heating surface in waste heat work usually is 35 to 40% of the boiler heating surface, which is much less than in direct-fired practice. The high ratio of gas weight to water weight in waste heat economizer practice makes this reduction possible. In direct-fired practice, rise in feedwater temperature through the economizer is, roughly, 1° F. for each 2° F. drop in gas temperature; in waste heat practice 1° F. rise in feedwater temperature results in approximately 1° F. drop in gas temperature. This makes possible, with economizers in waste heat work, an increase in feedwater temperature greatly in excess of that which may be obtained with an equal amount of economizer surface in direct-fired practice, despite low gas temperatures from the waste heat boiler.

CLEANING.—Gases used in waste heat boilers are invariably dirty. With proper arrangement of dusting apparatus, soot pockets, and in some installations, dust conveying systems, boilers, economizers and flues can be kept as clean as in direct-fired work. Dust-conveying systems are used when dust has a reclaimable value.

TYPICAL TEST RESULTS.—Table 2 shows results which have been obtained with waste heat boilers in connection with various industrial furnaces.

Table 2.—Tests of Waste Heat Boilers *

	Type of Industrial Furnace				
	Beehive Coke Ovens	Heating Furnace	Cement Kiln	Open-hearth Steel Furnace	Glass Tanks
Boiler heating surface, sq. ft.	10,200	5,480	14,800†	5,830	2,860
Weight of gas, lb. per hr.	155,100	87,571	194,735	61,000	43,660
Temperature of entering gas, deg. F.	2,158	1,745	1,325	1,436	808
Temperature of exit gas, deg. F.	477	436	506	464	401
Draft at boiler damper, in.	4.40	1.87	3.90	3.60	3.15
Draft at boiler inlet, in.	2.00	0.68	0.73	1.60	0.96
Draft loss through boiler, in.	2.40	1.19	3.17	2.00	2.19
Boiler horsepower developed	1,956	784	1,280	461	133
Percent of rated horsepower	192	134	87	79	46

* All tests with Babcock & Wilcox water-tube boilers.

† Two boilers of 740 Hp. each. Each boiler equipped with 2500 sq. ft. of economizer surface. Temperature of exit gas from economizer, 405° F.; power developed by economizer 111 Hp. Total developed Hp., boilers and economizers, 1391.

MOISTURE IN STEAM

1. STEAM CALORIMETERS

Moisture in steam is caused principally by high concentration of total solids, and other impurities (see p. 6–65), in the boiler water, resulting in foaming and priming. In boiler tests it is necessary to determine whether the steam as generated is wet, saturated, or superheated. To ascertain the quality of wet steam, a throttling or a separating calorimeter, or a combination of the two generally is used. The barrel calorimeter, one of the early types, has been abandoned as inaccurate, except at very high pressure or low moisture. Under these conditions the conductivity method should be used.

THROTTLING CALORIMETER.—In the simplest throttling calorimeter, steam is drawn from the main by a sampling pipe and throttled through a small orifice into a chamber, fitted with a thermometer well, a cock for pressure manometer attachment, and a comparatively large exhaust opening, closed by a valve. Some calorimeters are open to atmosphere on the exhaust side, the exhaust valve and manometer attachment being eliminated. The instrument, and all pipes and fittings leading to it, should be thoroughly insulated to diminish radiation losses. The only observations required are temperature and pressure of steam in the calorimeter chamber, and pressure of steam in the main. The range of moisture determination of the throttling calorimeter is limited, but it increases with higher pressures up to 400 lb. per sq. in. abs., where the limiting moisture content

of the steam is about 7%. At very low pressures, steam must be practically dry if the calorimeter is to operate satisfactorily. If the chamber of the calorimeter is connected to a condenser the range may be extended.

A simple form of throttling calorimeter is shown in Fig. 1. Two concentric metal cylinders are screwed to a cap containing a thermometer well. Steam pressure is measured by a gage connected to the supply pipe. Steam passes through orifice *A* and expands to atmospheric pressure, its temperature at this pressure being measured by thermometer *C*. Radiation losses are reduced by using the annular space *D* as a jacket to which steam is supplied through hole *B*.

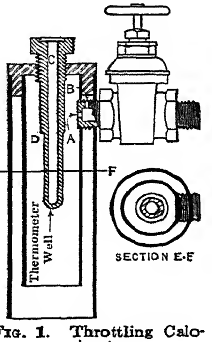


Fig. 1. Throttling Calorimeter

The principal source of error in steam calorimeter determinations is failure to obtain an average sample of steam. The type of steam sampling nozzle and its location is, therefore, extremely important. The A.S.M.E. Power Test Code recommends a 1/4- or 3/8-in. brass pipe, preferably the smaller, the portion projecting into the main being drilled in a straight line with 1/8-in. holes, as shown in Fig. 2 and Table 1. The end projecting within the main should extend to within 1/2 in. of opposite side of the steam main. The nozzle is installed in the line, with the holes directly facing the steam flow, preferably in a pipe where steam flow is downward, and as far removed from any disturbing element, as a valve or elbow, as possible. The next best location is in a pipe wherein flow is vertically upwards, other conditions being as before. A pipe in which steam ascends usually will show greater moisture, with the same sampling nozzle, than one in which steam descends. Pipe bends and horizontal pipes should be avoided as sampling nozzle locations.

EQUATION FOR DETERMINING MOISTURE IN STEAM.—Let p_1, p_2 = respectively, pressure in main and in calorimeter, lb. per sq. in., abs.; h_1, H_1 = respectively, heat of vaporization and total heat of steam corresponding to pressure p_1, B.t.u. per lb.; H_2 = total heat of saturated steam corresponding to p_2, B.t.u. per lb.; t_2 = temperature of saturated steam corresponding to p_2, deg. F.; t_c = temperature in calorimeter, deg. F.; c_p = specific heat of superheated steam in temperature range t_2 to t_c (assume 0.47 for approximately atmospheric pressures existing in calorimeter); Y_1 = percentage moisture in steam, by weight, at pressure p_1. Then

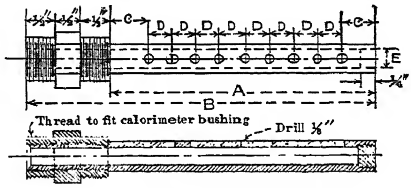


Fig. 2. Sampling Nozzle

$$- H_2 - 0.47(t_c - t_2)\}/h_1$$

Table 1.—Steam Sampling Nozzles for Pressures Above Atmospheric Pressure

Nominal Size of Steam Main, in.	Internal Diameter, in.	Holes in Nozzle		Dimensions, in. See Fig. 2				
		No.	Diam., in.	*A*	*B*	*C*	*D*	Pipe Size of Nozzle, in. *E*
2	1.939	3	1/8	1 3/4	3 1/4	5/8	1/2	1/4
2 1/2	2.323	3	1/8	2	3 1/2	1/2	1/2	1/4
3	2.9	5	1/8	2 3/4	4 1/4	3/8	1/2	1/4
3 1/2	3.364	5	1/8	3	4 1/2	1/2	1/2	1/4
4	3.826	6	1/8	3 5/8	5 1/8	9/16	1/2	1/4 or 3/8
4 1/2	4.290	6	1/8	4	5 1/2	7/16	5/8	1/4 or 3/8
5	4.813	6	1/8	4 5/8	6 1/8	7/16	3/4	1/4 or 3/8
6	5.761	6	1/8	5 1/2	7	7/8	3/4	1/4 or 3/8
8	7.625	6	1/8	7 1/2	9	5/8	1 1/4	1/4 or 3/8
10	9.75	6	1/8	9 1/2	11	3/8	1 3/4	3/8
12	11.75	7	1/8	11 1/2	13	1/2	1 3/4	3/8
14	13	7	1/8	12 1/2	14	5/8	1 7/8	3/8
16	14.75	8	1/8	14 1/4	15 3/4	1 1/2	1 7/8	3/8
18	16.5	8	1/8	16	17 1/2	1 7/16	1 7/8	3/8
20	18.5	9	1/8	18	19 1/2	1	2	3/8

Almost invariably the calorimeter exhausts to atmosphere. Then $H_2 = 1150.2$ and $t_2 = 212$, and

$$Y_1 = 100\{H_1 - 1150.2 - 0.47(t_c - 212)\}/h_1.$$

Moisture content may be determined by the total heat-entropy chart. Since in throttling, total heat remains the same it is only necessary to locate the point on the chart corresponding to conditions in the calorimeter (p_2 and t_c), and follow a constant total heat line until it intersects a constant pressure line corresponding to pressure in the main (p_1). Quality then can be read directly from the chart, and percentage of moisture by weight is (1 − quality reading from chart) × 100.

Correction for Radiation.—Loss from radiation will result in the temperature in calorimeter chamber being low. This temperature reading is corrected as follows:

The boiler is run at low load with low concentration, in order to obtain dry steam, at the pressure expected during the test. When observed temperature in the calorimeter becomes constant, the difference between this temperature and that of dry saturated steam, throttled to calorimeter pressure is the correction to be added to the all subsequent calorimeter temperature readings. No change then should be made in calorimeter installation or thermometer, until actual boiler test is completed.

EXAMPLE: At 160 lb. gage pressure, theoretical temperature of dry saturated steam when throttled to atmospheric pressure is 308° F. Assume normal reading to be 304° F. $(308 - 304) = 4°$ = radiation correction factor (including thermometer correction), to be added to all calorimeter temperature readings taken during test.

SEPARATING CALORIMETER.—Steam enters a separation device, consisting essentially of a perforated cup. Moisture is thrown out and deposited in a separating chamber, while the dry steam passes up and out of the separating device, through an annular steam jacket surrounding the separating chamber. The dry steam discharges through an orifice. Its total amount may be determined by condensing and weighing, or by calculating the flow by Napier's formula (see p. 16–53), providing size of orifice is known. The amount of moisture deposited in the separating chamber can be read directly from a water glass graduated in $^1/_{100}$ lb. While the accuracy of this calorimeter is less than that of the throttling type, it has a much wider range. The percentage of moisture by weight is $Y = 100\,w/(W + w)$, where Y = percentage of moisture, by weight; w = weight of moisture collected in separating chamber, lb.; W = weight of dry steam condensed after passing through calorimeter, or weight as calculated from flow formula, lb. If well insulated with hair felt, the radiation loss is known to be less than 0.05%.

THE UNIVERSAL CALORIMETER consists of a separating and throttling calorimeter, of high and low range, respectively, in series. Let Y_1 = percentage of moisture, by weight, in steam as determined by combination calorimeter; w_1 = weight of moisture collected in separating calorimeter in a given time, lb.; w_2 = weight of dry steam condensed after passing through the throttling calorimeter, lb.; Y_2 = proportion, by weight, of moisture in steam discharged from separating portion as determined by throttling calorimeter. Then, without radiation losses, $Y_1 = \{(w_1 + w_2 Y_2)/(w_1 + w_2)\} \times 100$.

THE SUPERHEATING CALORIMETER consists of a device where steam flows through a pipe which is jacketed with superheated steam, of sufficiently high temperature to superheat the steam sample. Moisture in sample taken from steam main may be determined by a heat balance between heat absorbed by sample and that lost by steam in the jacket.

THE ELECTRIC CALORIMETER is a form of superheating calorimeter. Steam from sampling nozzle enters the bottom of the calorimeter, passes upwards over heating coils, and thence to atmosphere. To determine moisture content, it is necessary to know the electrical input, temperature of exhaust steam, and the formulas and constants determined by the manufacturer of the apparatus.

CONDUCTIVITY METHOD of determining moisture and solids concentration in the steam consists of obtaining simultaneous samples of steam condensate (having a minimum of CO_2) and boiler water, both being cooled to approximately 77° F. from which the following data are obtained:

Conductivity of steam condensate; conductivity of 1% solution of boiler water in steam condensate; temperature of these samples; solids in the boiler water. From these data the parts per million (P.p.m.) of solids in the steam may be calculated as follows:

$$\text{P.p.m} = \{(L_s - W_c)/S \times 100)\}\text{P.p.m. in boiler water.}$$

$$\text{Equivalent moisture in steam} = (L_s - W_c)/S,$$

where L_s = conductivity of steam condensate at standard temperature of 77° F.; W_c = total conductivity correction due to CO_2 in pure water; S = conductivity increase due to adding 1% boiler water to steam condensate, corrected to standard temperature. The apparatus required is rather elaborate and the procedure somewhat involved. For

details, see Estimation of Solids in Steam Conductivity, J. K. Rummel, Analytical Ed. *Ind. and Engg. Chem.*, July 15, 1931.

2. STEAM SEPARATORS

STEAM SEPARATORS may be classified as gravity and inertia types. In the gravity type the velocity of steam is low enough to allow particles of moisture to fall through the flowing stream. In the inertia type, the velocity is high enough to project the particles of moisture on to collecting surfaces, from which they may drain. Inertia separators may be classed as single stream and subdivided stream separators, depending on their construction.

Boiler Drum Separators.—Separation of moisture from steam delivered from boilers may be done either in the boiler drum or in external separators. The oldest and commonest types of steam drum separator are the vertical baffle plate, which separates the steam space into two sections, the dry-pipe, and the dry-pan. The vertical baffle plate is sealed at the bottom by the water in the drum. Steam and moisture discharged from the boiler tubes is received in one section of the drum. The baffle restricts turbulence

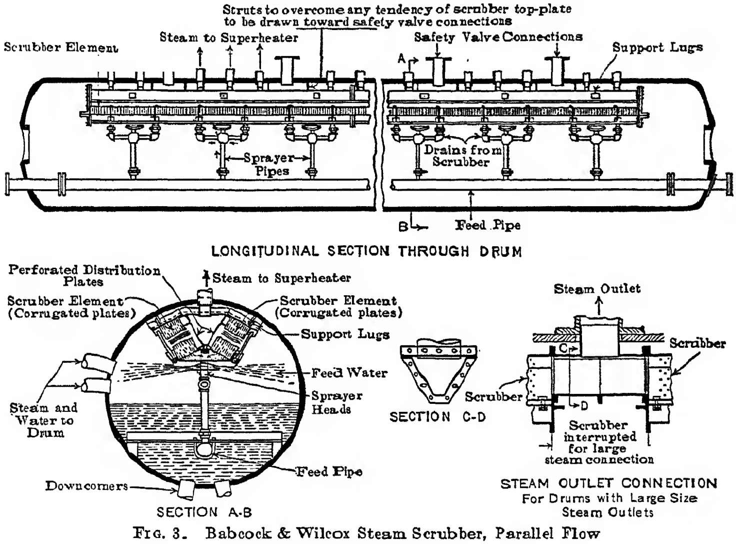


Fig. 3. Babcock & Wilcox Steam Scrubber, Parallel Flow

caused by the entrance of steam and moisture to the first section. Steam leaves the other section in a relatively quiescent state. Steam discharged from the turbulent section, after most of the water has been separated in the upper section, flows endwise to the extreme ends of the drum, around the baffle, and continues at relatively slow velocity in the quiet section to the steam outlet. This arrangement is reasonably satisfactory, providing the velocity of flow is low enough to permit the drops to fall by gravity against the current.

The latest arrangement (1935) of low-velocity steam separator uses a multiplicity of corrugated plates, offering relatively large surface for collecting moisture as the steam changes directions through the corrugated passages. Combined with this steam scrubber arrangement, steam sometimes is washed with the relatively clean boiler feedwater, delivered between sections of the series type, or before the sections of the parallel type of the steam scrubber, thus diluting impurities in the water comprising moisture in the steam, without adding to its total moisture content.

PERFORMANCE OF STEAM SEPARATORS.—The quality of steam which may be produced will vary over a considerable range, depending on type of boiler, boiler water concentrations, service conditions and type of separator used. In general, modern boilers

will deliver steam containing not more than 0.5% moisture, and often not more than 0.25% moisture. With proper control of boiler water concentrations and more efficient steam separators, moisture content may be zero as measured by steam calorimeter and it is necessary to resort to the conductivity method (see J. K. Rummel, Anal. Ed. *Indust. & Engg. Chem.*, pp. 317–320, July 15, 1931) which gives a measure of solids and equivalent moisture in the steam. It has been shown that with suitable equipment the steam may contain not more than one part per million of solids and not more than 0.1% equivalent moisture.

Fig. 3 shows the Babcock & Wilcox patented parallel-flow steam scrubber. Steam from the circulating tubes passes through a feedwater spray from sprayer heads and flows in parallel through both banks of corrugated elements, then through perforated distribution plates. It flows from the drum to the superheater through connecting tubes or a steam outlet connection. This scrubber is particularly suitable for high capacity with small size drums.

Fig. 4 shows a patented series flow steam scrubber. Steam from circulating tubes *a* passes through the first bank of corrugated elements *b* which removes the bulk of the entrained moisture, thence through the sheet of feedwater from the washer *d* and through

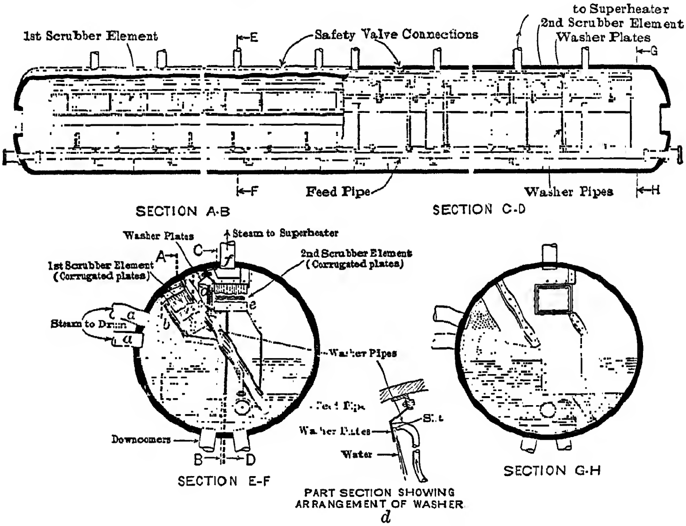


Fig. 4. Babcock & Wilcox Steam Scrubber, Series Flow

the second bank of corrugated elements to the steam space *e*, passing out of the drum to the superheater through tubes *f*. Closure plates at the ends of the scrubbers prevent steam from by-passing the corrugated elements. The following are results obtained with Babcock & Wilcox boilers with steam scrubbers installed.

	Plant A	Plant B
Boiler capacity, lb. per hr.	273,000	300,000
Steam pressure, lb. per sq. in.	1,400	625
Feed, type	100% condensate	100% make-up
Boiler water, concentration, parts per million	2,350	3,000
Solids in steam, parts per million	0.2	0.5
Equivalent moisture in steam less than	0.1%	0.1%

At plant A, with the original steam baffles in the boiler drums, the solids carry-over was at times sufficient to cause back pressure in the first stage of the high-pressure turbine to build up to 150 lb. per sq. in. after three or four days of operation. After installing the steam scrubber, operation for several months showed no increase in pressure, realizing a considerable saving in outage for cleaning.

BOILER FEEDING

1. THE INJECTOR

Principle of Operation of the Injector.—The simplest form of single-tube injector is shown in Fig. 1. Entering steam, in passing through the nozzle, acquires high velocity and is condensed by water in the combining tube. This creates a vacuum into which water flows through the water supply pipe. The high-velocity steam entering through *the nozzle, comprising a mixture of condensed steam and water, flows into the delivery* tube. There, the energy of steam expanding from boiler pressure to a partial vacuum produced by condensation, is sufficient to create pressure in the water as much as 50 to 80 lb. per sq. in. in excess of the boiler pressure, for the range of pressures in which the injector is used. This excess pressure forces the water into the boiler.

Equation of the Injector.—Let S = lb. of steam used; W = lb. of water lifted and forced into boiler; h = height, ft., of a column of water, equivalent to absolute pressure in boiler; h_0 = height, ft., that water is lifted to injector; t_1, t_2 = respectively temperature of water entering and leaving injector, deg. F.; H = heat content of steam supplied to injector, B.t.u. per lb., above 32° F.; L = work lost in friction and the equivalent lost work due to radiation and lost heat; 778 = mechanical equivalent of heat. Then

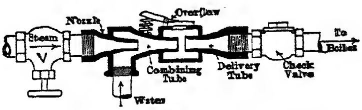


FIG. 1. Diagram of the Injector

$$S\{H - (t_2 - 32)\} = W(t_2 - t_1) + [\{(W + S)h + Wh_0 + L\} \div 778] \quad . \; . \; . \quad [1]$$

Neglecting $(Wh_0 + L)$,

$$S = [W(t_2 - t_1) + \{(W + S)/d\}\,p \times (144/778)]\,[1/\{H - (t_2 - 32)\}] \quad . \; . \quad [2]$$

or

$$S = W[(t_2 - t_1)d + 0.1851\,p] \div [\{H - (t_2 - 32)\}d - 0.1851\,p] \quad . \; . \; . \; . \quad [3]$$

where d = weight of 1 cu. ft. of water at temperature t_2; p = absolute pressure of steam, lb. per sq. in.

If in equation [1] the quantity $[\{(W + S)h + Wh_0 + L\} \div 778]$, which is the work of pumping, is assumed equal to zero, the equation takes the form

$$W/S = (H - t_2 + 32) \div (t_2 - t_1),$$

which is approximate and is analogous to that which Strickland Kneass (Theory of the Injector, p. 83) gives for the performance of an injector.

To find proper sectional area for narrowest part of the nozzle Rankine (Steam Engine, p. 477), gives

Area, sq. in. = (cu. ft. per hr. gross feedwater) ÷ ($800\sqrt{\text{pressure in atmospheres}}$).

Positive and Automatic Injectors.—Positive-type injectors have hand-controlled overflow valves, which are closed after operation has started and water appears in the overflow. The advantages of this type of injector are its ability to lift water to a greater height, to start with a lower steam temperature, and to discharge against a higher back pressure. In automatic injectors, opening and closing of the overflow is entirely automatic. This type is preferred for stationary work because of its restarting features.

THE INJECTOR AS A BOILER FEEDER is efficient and convenient. It has no moving parts, is compact, delivers hot water to the boiler without preheating, and has no exhaust steam to be disposed of. When used to feed water to a boiler, its thermal efficiency is 100%, less the trifling loss due to radiation, since all heat rejected passes into the water and is carried into the boiler. The loss of work in the injector, due to friction, reappears as heat which is carried into the boiler. The heat converted into useful work in the injector appears in the boiler as stored-up energy. Although the injector has perfect efficiency as a boiler feeder, it is not the most economical means of feeding because of its inability to handle hot water, thereby excluding the utilization of other sources of waste heat for boiler-feed heating. It also is difficult to maintain continuous flow with the injector at low capacity, because of the necessity of starting and stopping under such conditions. Furthermore, it cannot be operated with very high pressure steam due to the difficulty of efficiently operating a small nozzle under high pressure.

The injector has been widely used on locomotives but has been displaced in certain cases by direct-acting feed pumps, especially when feedwater heaters are used. It is limited in stationary work to small or single boilers, or as a reserve feeder. The injector used as a pump has an efficiency of approximately 1 to 2 percent. The weight of feedwater handled per pound of steam usually decreases as steam pressure increases, and varies between approximately 21 lb. at 20 lb. per sq. in. gage pressure to 10 lb. at 100 lb. pressure.

Maximum temperature of feedwater which can be handled will not vary widely with steam pressure, and averages from 120 to 140° F. at sea level, and lower at higher altitudes.

Table 1.—Test of Sellers Injector

From Practice and Theory of the Injector, by S. L. Kneass

Mean steam pressure, lb. per sq. in.	30	60	121	150	200
Temperature of supply water, deg. F.	67	67	54	54	50
Maximum capacity:					
Gallons water handled per hr.	1912	2535	3517	3765	4005
Temperature of delivered water, deg. F.	113	125	134	135	154
Weight of delivered water per lb. of steam used, lb.	25.9	19.1	13.6	12.6	10.3
Minimum capacity:					
Gallons water handled per hr.	765	937	1290	1432	1732
Temperature of delivered water, deg. F.	171	212	238	250	263
Ratio of minimum to maximum capacity	0.4	0.37	0.37	0.38	0.43

2. BOILER-FEED PUMPS

Pumps for boiler feeding are of two general classes: Direct-acting displacement pumps and centrifugal pumps.

RECIPROCATING PUMPS used as boiler-feed pumps usually are direct-acting duplex pumps. This type is essentially the same as two single pumps (direct-acting simplex) mounted side by side, with the piston rod of one actuating the variable gear of the other, and gives a practically continuous flow of water. Although the direct-acting simplex pump has lower first cost and maintenance, it delivers feedwater with a certain irregularity which may lead to strains in the feed piping.

Plungers are preferable to water pistons because of greater ease in adjustment, minimum slippage, and because all leakage is visible. The double plunger pump is equivalent to a piston. Steam consumption may be decreased by compound or triple-expansion steam ends, but this gain may be offset by increased initial and maintenance costs. Where exhaust is used for heating the feed there is little difference in plant efficiency.

The average direct-acting simplex or duplex boiler feed pump uses approximately 5% of the boiler steam, but if the pump exhaust is used to heat the feedwater, the net heat consumption is less than 0.1%. Direct-acting or duplex pumps for boiler feeding are, in general, used in the smaller plants because of their simplicity and reliability, it being possible to operate them even under water. See also p. 2–69.

CENTRIFUGAL PUMPS are almost universally used in medium and large size plants. They are built in two or more stages, depending on the pressure at which feedwater is to be delivered. Single-stage centrifugal pumps rarely are used for heads of over 250 ft.

Centrifugal pumps seldom are used in small plants because of the difficulty of building pumps of this type that will be efficient in small sizes, especially for high pressure work. As pressure increases, the minimum size of the pump increases, as does also back leakage.

Centrifugal pumps are continuous in action and do not set up pulsating strains in feed piping. Motor-driven multi-stage boiler-feed pumps may be operated at 3600 r.p.m., but 1750 r.p.m. is the usual speed. For larger pumps 1200 r.p.m. is used. Turbine-driven pumps may be operated at any desired speed, which may be varied in accordance with load conditions. This feature is not available in motor-driven pumps unless operated under direct current, or in connection with variable speed gear sets. Efficiencies as high as 87 to 88% have been realized under very favorable conditions, but for average size pumps handling about 1000 gal. per minute against 600 lb. pressure, an efficiency of 75 to 80% is considered good performance. See p. 2–83.

3. FEEDWATER HEATERS *

TYPES OF HEATERS.—While any device used to transmit heat to feedwater, prior to admitting this water to the boiler, may be called a feedwater heater, the term generally is applied to equipment using steam for heating. This equipment comprises two general classes: 1. Open, or direct contact, heaters, in which the steam comes directly in contact with the water. Tray type heaters and jet heaters form the two main sub-divisions of this class. 2. Closed heaters, in which the heat from the steam is transmitted through tubular metallic walls to the feedwater. Either open or closed heaters can utilize the exhaust from engines or pumps, or be used as stage heaters supplied with steam extracted from bleeder turbines.

SAVINGS OF FEEDWATER HEATERS.—Feedwater heaters, either open or closed, are useful in conserving the heat in pump or engine exhaust, high-pressure trap discharges, etc., which otherwise would be wasted. There is, roughly, a saving of 1 percent for every

* Contributed by J. S. Daugherty.

10° F. that the feedwater is heated. The saving effected by a heater may be determined from the formula $(h_2 - h_1)/(H - h_1)$, where h_1, h_2 = B.t.u. per pound of feedwater entering and leaving heater, respectively; H = B.t.u. per pound of steam at boiler pressure.

ELIMINATION OF STRAINS CAUSED BY COLD FEEDWATER.—A calculation in *The Locomotive*, March, 1893, shows that the injection of cold feedwater into a boiler will impose a stress of nearly 38,000 lb. per sq. in., on a strip of steel 10 in. long adjacent to the entrance of the feedwater pipe if no allowance is made for elasticity of adjoining sections of the boiler. Making allowance for such elasticity, however, it is quite probable that a stress of from 8000 to 10,000 lb. per sq. in. may be imposed by cold feedwater striking directly upon the plates. This stress, in addition to the normal stress due to steam pressure, will easily tax the girth seams beyond their elastic limit if the feed pipe discharges near them.

REMOVAL OF GASES FROM FEEDWATER.—Possibly of even greater importance than the fuel saving effected by the open heater is its ability to liberate and remove dissolved gases from feedwater. Gases cannot stay in solution when the water is heated to the boiling point. Consequently, water temperatures should be maintained as close as is practical to saturated steam temperatures, even to the extent of supplementing exhaust steam with reduced pressure live steam. The extent to which oxygen can be expelled by heating water in standard open heaters at atmospheric pressure is shown by the curve, Fig. 2, supplied by the Cochrane Corporation.

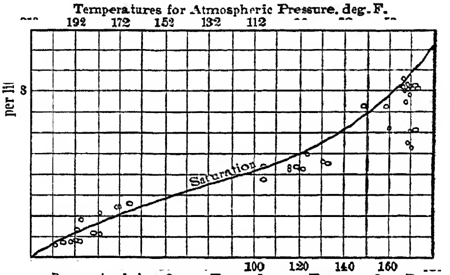


Fig. 2. Oxygen Content of Water from Standard Open Heaters

The deaerating feedwater heater, Fig. 2, has been developed from the older open heater. It is designed to accomplish practically complete removal of dissolved gases, of which oxygen is naturally the most objectionable. This is effected by heating the water exactly to the saturated steam temperature, spreading it in thin sheets over successive layers of air-separating trays, agitating it thoroughly so the gases may be brought to the surface and liberated, and sweeping the liberated gases

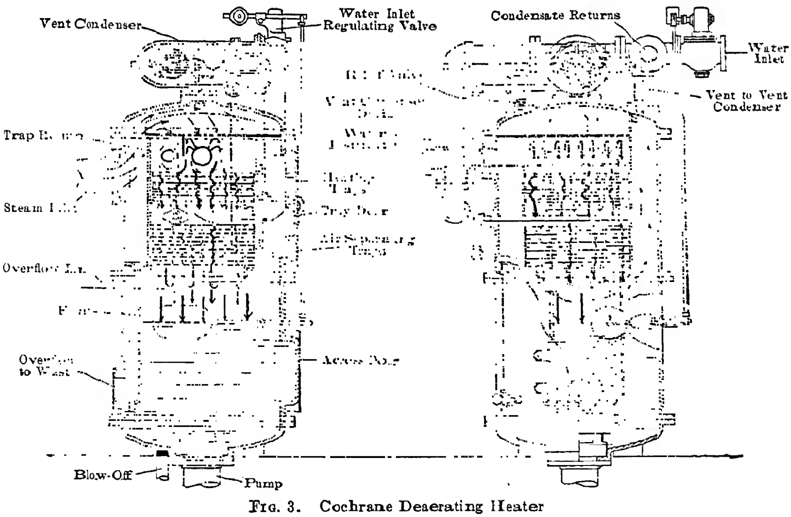


Fig. 3. Cochrane Deaerating Heater

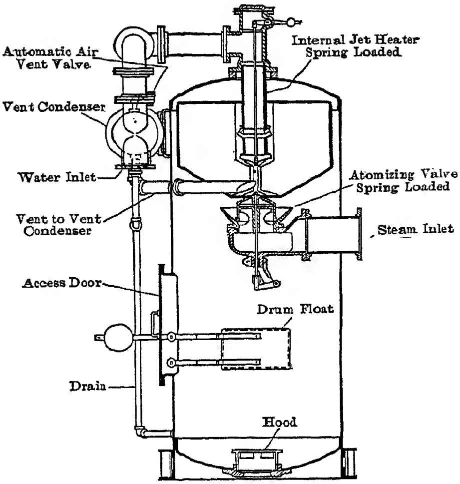


FIG. 4. Cochrane Deaerating Heater for Marine Service, Using Jets Instead of Trays

away with the steam vented to the vent condenser. Fig. 4 shows a deaerating heater for marine service, using jets instead of trays to give the same performance as the stationary deaerating heater.

CONSTRUCTION OF OPEN HEATERS.—When used for low pressures open heaters usually are constructed of cast iron. When supplied with superheated steam or used with steam extracted from bleeder turbines, the shells of tray type or jet type heaters are rolled plate, either riveted or welded.

PROPORTIONS OF OPEN TYPE FEEDWATER HEATERS.—The jet type open feedwater heater is of importance due to its ability to heat large quantities of water in a relatively small space. One large manufacturer offers a line of standard jet heaters with outlet capacities ranging from 100,000 lb. per hr. to 1,000,000 lb. per hr. in which the internal volume may be approximated from the formula,

$$V = W/10{,}000,$$

where V = internal volume, cu. ft.; W = outlet capacity, lb. per hr. The proportions of tray type open heaters are governed primarily by the particular conditions of operation and no general rule for these proportions is available. An approximation of the size of the heater may be made by having at least 1 sq. ft., in plan, of tray stack for each 15,000 lb. per hour capacity. Vertical units vary in height from about 4 ft. for small capacities to 10 ft. for larger capacities. About half the height is used for water distribution and the tray stack.

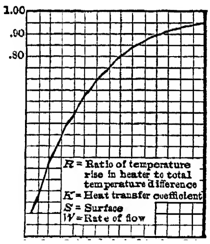


FIG. 5. Curve for Determining Surfaces of a Closed Heater

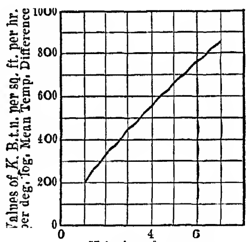


FIG. 6. Values of K

Water storage may be combined with open heaters. Where the feedwater is primarily all make-up and the load fluctuations are not severe, approximately two minutes boiler supply has been found sufficient. When the feedwater is condensate, with but a small amount of make-up, it often is the practice to incorporate condensate surge space in its storage compartment. The capacity for condensate surge varies from 5 minutes to 30 minutes supply for the boiler.

The location of an open feedwater heater in relation to the boiler feed pump is important. It must be at such an elevation above the pump inlet that the pump will receive only vapor-free liquid. The elevation will depend on temperature and pressure of water leaving the heater. Table 3, p. 2–67 gives data concerning the elevation of pumps handling hot water.

ADVANTAGES OF CLOSED FEEDWATER HEATERS.—Closed heaters in which steam does not come in contact with water but transmits its heat through tubular walls are quite important devices for heating service water for buildings and institutions, and for heating water for industrial processes, due to their ability to prevent the odor, taste, or impurities of the steam from being transmitted to the water being heated. The practical application of closed heaters to boiler feedwater heating service is limited to their inclusion as stage heaters in the regenerative feed heating cycle, which cycle should contain at least one direct contact feedwater heater. In the multi-stage regenerative feed heating cycle a multiplicity of feedwater pumps can be avoided by the use of closed heaters.

CALCULATIONS OF SURFACE OF CLOSED HEATERS.—The surface of a closed heater may be determined from the formula

$$S = (W \times T)/(K \times t_d)$$

where S = heat transfer surface, sq. ft.; W = lb. per hr. to be heated; T = temperature rise, deg. F.; K = heat transfer coefficient, B.t.u. per deg. F. logarithmic mean temperature difference per sq. ft. per hr.; t_d = logarithmic mean temperature difference, deg. F.

By using curve Fig. 5, developed by A. E. Kittredge of Cochrane Corporation, the surface of a closed heater also may be determined when the specific heat of the fluid is unity. In Fig. 5, R = Heating Ratio = (Temperature Rise)/(Max. Temp. Difference); K, S and W are as above. Values of K are influenced by velocity of fluid, fluid density, material of tube walls, etc. Conservative values are shown in Fig. 6.

FEEDWATER FOR STEAM BOILERS

1. COMPOSITION AND ANALYSIS OF FEEDWATER

IMPURITIES IN FEEDWATER.—Natural feedwater supplies contain solids and dissolved gases which may promote the following conditions in boilers: 1. Incrustation or scale. 2. Foaming, priming and solids in steam. 3. Corrosion. 4. Caustic embrittlement. To avoid these troubles, it generally is necessary to study each water supply individually and determine its individual characteristics and how it best may be treated.

Because of the high solids content, sea water and certain other bodies of water are unfit for use in boilers. Rain water becomes contaminated in falling through the atmosphere, and always contains dissolved gases, including oxygen and carbon dioxide. The latter forms a mild acid which greatly increases the solvent action of the water. Thus, with carbonic acid present, it can dissolve considerable amounts of such materials as calcium and magnesium salts from the ground through which the water passes.

It follows that waters from rivers, wells, and lakes will contain varying amounts of dissolved and suspended solids, depending on geologic formation, climate, vegetation, and pollution from various sources. Waters from mines where sulphates are present tend to be acid. Rivers and streams may become acid from industrial pollution, and sewage, sometimes present, produces objectionable decomposition products. Waters in rivers and streams often undergo rapid changes in dissolved and suspended solids, requiring close control of feedwater purification processes.

CLASSES OF IMPURITIES.—Table 1 is a partial list of impurities found in boiler feedwater, their effect in the boiler and the usual method of treatment. The solubilities are listed to show constituents which may be present in water, and whether they can be expected to precipitate under boiler operating conditions. Increase of temperature increases the solubility of some solids and precipitates others. Regardless of whether solubility increases or decreases with temperature, concentration of solids in the boiler water increases with continued evaporation. Table 2 gives the solubility of substances listed in Table 1. As solubilities vary with temperature and authorities differ on the values for some constituents, it is not possible to estimate all solubilities under boiler conditions. The important feature is the probable effect of these constituents in the boiler. See next to last column, Table 1.

The impurities may be roughly classified under the following headings:

Dissolved Gases.—Inert gases, as nitrogen and the hydrocarbons. Corrosive or active gases, as oxygen, carbon dioxide and hydrogen sulphide.

Dissolved Solids.—*Slightly Soluble Solids:* Includes most calcium and magnesium compounds. Also oil and silica.

Highly Soluble Solids: Includes all soluble salts, as sodium chloride, sodium sulphate, sodium carbonate, sodium nitrate and certain sodium silicates. Also sodium hydroxide, sodium phosphate, the acids and certain organic compounds.

Suspended Solids include the common clays and silts, organic and inorganic matter, found principally in rivers and streams, and all other insoluble matter.

Insoluble Liquids, oils, greases, soaps, etc., have a deleterious effect on boiler water.

Table 1.—Usual Impurities of Boiler Feedwater

Impurity	Formula	Molecular Weight	Equivalent Weigh	Solubility*	Probable Effect in Boiler	Methods of Treatment and Removal
Calcium Bicarbonate	$Ca(HCO_3)_2$	162.10	81.05	Moderate	Scale and sludge. Liberates CO_2	In external treatment of calcium and magnesium compounds, lime and soda softeners plus coagulation and filtration give partial removal. Zeolite softeners and evaporators give more complete removal, the former replacing calcium and magnesium with sodium. Corrosive compounds require alkali treatment. In internal treatment, calcium and magnesium are precipitated as hydroxide and carbonates by sodium hydroxide and sodium carbonate. Calcium, and sometimes part of the magnesium are changed to calcium and magnesium phosphates by treatment with sodium phosphates. Sodium hydroxide is preferred reagent for internal treatment of magnesium compounds. Calcium hydroxide is preferred for external treatment.
Calcium Carbonate	$CaCO_3$	100.08	50.04	Slight	Scale and sludge. Liberates CO_2	
Calcium Hydroxide	$Ca(OH)_2$	74.10	37.05		Scale and sludge	
Calcium Sulphate	$CaSO_4$	136.14	68.07	Moderate	Hard Scale	
Calcium Silicate	Variable	....	...	Slight	Hard Scale	
Calcium Chloride	$CaCl_2$	110.99	55.50	Very soluble	Corrosive. Scale and sludge	
Calcium Nitrate	$Ca(NO_3)_2$	164.10	82.05	Very soluble	Corrosive. Scale and sludge	
Magnesium Bicarbonate	$Mg(HCO_3)_2$	146.34	73.17	Moderate	Deposits. Liberates CO_2	
Magnesium Carbonate	$MgCO_3$	84.32	42.16	Slight	Deposits. Liberates CO_2	
Magnesium Hydroxide	$Mg(OH)_2$	58.34	29.17	Very slight	Deposits	
Magnesium Sulphate	$MgSO_4$	120.38	60.17	Very soluble	Corrosive, deposits	
Magnesium Silicate	Variable	....	...	Slight	Hard Scale	
Magnesium Chloride	$MgCl_2$	95.23	47.62	Very soluble	Corrosive, deposits	
Magnesium Nitrate	$Mg(NO_3)_2$	148.34	74.17	Very soluble	Corrosive, deposits	
Sodium Bicarbonate	$NaHCO_3$	84.00	42.00	Very soluble	Increases alkalinity and soluble solids. Liberates CO_2	
Sodium Carbonate		106.00	53.00	Very soluble	Increases alkalinity and soluble solids. Liberates CO_2	Excess sodium alkalinity may be reduced by boiler blowdown. It sometimes is neutralized with sulphuric acid externally. Phosphoric acid and acid phosphates also are used. Evaporation is best practical means of removing sodium compounds from feedwater. Boiler blow-down used for internal reduction of soluble solids.
Sodium Hydroxide	NaOH	40.00	40.00	Very soluble	Increases alkalinity and soluble solids.	
Sodium Sulphate		142.05	71.03	Very soluble	Inhibitor for caustic embrittlement. Increases soluble solids.	
Sodium Silicate	Variable			Very soluble	Increases alkalinity. May form silica scale	
Sodium Chloride		58.45	58.45	Very soluble	Increases soluble solids. Encourages corrosion.	
Sodium Nitrate	$NaNO_3$	85.01	85.01	Very soluble	Increases soluble solids.	
Iron Oxide	Fe_2O_3	159.68	26.61	Slight	Deposits. Encourages corrosion	Coagulation and filtration, evaporation, blowdown.
Alumina	Al_2O_3	101.94	16.94		May add to deposits	Coagulation and filtration, evaporation, blowdown.
Silica	SiO_2	60.06	30.03		Hard scale, acts as binder for deposits	Precipitation with aluminates, coagulation and filtration, evaporation, blowdown.
Dissolved Oxygen	O_2	32.00	16.00		Corrosive	Deaeration preferred.
Carbonic Acid or Dissolved CO_2	H_2CO_3	62.02	31.01	Very soluble	Retards hydrolysis of carbonates. Reduces alkalinity	Deaeration and alkali treatment.
Hydrogen Sulphide	H_2S	34.08	17.04	Very soluble	Corrosive	Deaeration and alkali treatment.
Acids, Organic & Mineral				Very soluble		Neutralization by alkali treatment.
Oil and Grease				Slight	Corrosive, deposits, foaming and priming	Coagulation and filtration, skimming.
Organic Matter				Very soluble	Corrosive, deposits, foaming and priming	Coagulation and filtration, evaporation.

*See Table 2.

Table 2.—Solubility of Impurities in Boiler Feedwater
Solubility in grams of substance in 100 grams of water

Substance	Formula	0° C	100° C	Authority
Calcium Bicarbonate...	$Ca(HCO_3)_2$	Soluble	Decomposes	
Calcium Carbonate....	$CaCO_3$	0.0013 (16° C)	0.002	Landolt Börnstein
Calcium Hydroxide....	$Ca(OH)_2$	0.1771	0.0667	Inter. Crit. Tables
Calcium Sulphate.....	$CaSO_4$	0.1759	0.1688	" " "
Calcium Silicate.......	$CaSiO_3$	0.0095 (17° C)*†	....	Seidell
Calcium Chloride......	$CaCl_2$	59.378	157.600	Inter. Crit. Tables
Calcium Nitrate.......	$Ca(NO_3)_2$	102.061	362.630	
Magnesium Bicarbonate	$Mg(HCO_3)_2$	Soluble	Decomposes	
Magnesium Carbonate.	$MgCO_3$	0.0106 (Cold)		Handbook of Chem. and Physics—Hodgman–Lange
Magnesium Hydroxide.	$Mg(OH)_2$	0.0008		Seidell
Magnesium Sulphate...	$MgSO_4$	26.725 (1.8° C)	71.027	Inter. Crit. Tables
Magnesium Silicate....	$MgSiO_3$	*		
Magnesium Chloride...	$MgCl_2$	52.380	72.284	Inter. Crit. Table
Magnesium Nitrate....	$Mg(NO_3)_2$	66.455	137.211 (90° C)	
Sodium Bicarbonate...	$NaHCO_3$	6.888	16.465 (60° C)	
Sodium Carbonate.....	Na_2CO_3	6.996	45.153	
Sodium Hydroxide.....	$NaOH$	42.005	338.642	
Sodium Sulphate......		4.858	42.192	
Sodium Silicate.... .	2	*		
Sodium Chloride......	$NaCl$	35.658	39.165	Inter. Crit. Tables
Sodium Nitrate.......	$NaNO_3$	73.274	175.450	" " "
Iron Oxide...........	Fe_2O_3	Insoluble		
Alumina..............	Al_2O_3			Handbook of Chem. and Physics—Hodgman-Lange
Silica................	SiO_2			
Oxygen..............	O_2	0.0069 ‡	0.0	
Carbon Dioxide.......	CO_2	0.3346 ‡	0.0576 (60° C)	
Hydrogen Sulphide....	H_2S	0.7066 ‡	0.0	

* The formulas and solubilities of the silicates are extremely variable, ranging from very high solubilities (especially sodium) to slight solubility. † Per 100 cc. solution. ‡ Pressure, 760 mm.

SPECIFICATIONS FOR BOILER WATER AND FEEDWATER.—Perry Cassidy (Joint meeting of Engrs. Soc. Western Penna. and Pittsburgh Section A.S.M.E., Oct. 15, 1935) gives the following specifications as complying with what is practicable with present feedwater treatment methods and equipment:

Feedwater.—*Dissolved Oxygen.* Preferably zero and not over 0.05 cc. per liter for boilers; zero where steel tube economizers are used. *pH Value.* Not less than 7. Excess alkalinity other than required for treatment or protection of feed lines, or to neutralize acids, should be reduced to a minimum. *Hardness.* Preferably zero. Not over 26 parts per million in terms of calcium carbonate. *Chloride.* Lowest practical minimum is desired. When due to condenser or other leakage not over 6 parts per million in terms of chlorine. *Oil.* None. *Total Solids.* Reduce to minimum. *Suspended Solids.* None. *Organic Matter.* Not more than 5 parts per million.

Boiler Water.—*Sodium Phosphate.* With residual hardness in the make-up, 50 to 100 parts per million expressed as disodium phosphate. *Alkalinity.* Between 100 to 250 parts per million depending upon silicates which also are present. The higher alkalinity is preferred when silicate concentration is 100 to 200 parts per million. *Chlorides.* Not over 500 parts per million expressed as chlorine. Preferably as low as possible. *pH Value.* Not less than 10.5, preferably 11.0. *Sulphate-Carbonate Ratio.* See p. 6–75. *Oil.* None. *Total Solids.* Not over 1700 parts per million.

ANALYSES.—In examining raw water supplies to determine their suitability for feedwater and proper methods for purification, a complete analysis is preferred. When the purification plant has been standardized, control tests may be applied to feed and boiler water to maintain desired conditions. These usually consist of tests for: Alkalinity or acidity; pH value; hardness; chloride; sodium sulphate; dissolved oxygen; dissolved solids; turbidity.

Alkalinity or Acidity is measured quantitatively by a titration method, using a standard acid or alkali in a burette and flask containing sample and color indicator. If indicator shows an alkaline reaction, sample is titrated with the standard acid until a certain color end point is reached. If indicator shows an acid reaction, it is titrated with the standard alkali to a predetermined end point. Alkalinity or acidity is then calculated in parts per million or grains per gallon of the predominating alkali or acid.

***p*H Value (Hydrogen Ion Concentration)** is determined to measure the degree of acidity or alkalinity of a sample. The colorimetric method generally used consists of adding a measured amount of a chosen indicator to a measured volume of sample in a

test tube or small cell. The color of the tube is compared to sets of color standards which represent the result for different pH values. A useful universal indicator which can be used for both titration work and approximate pH value, can be obtained from laboratory supply houses, as Palo-Myers, Inc., New York.

Explanation of pH Value.—All aqueous solutions contain hydrogen (H) and hydroxyl (OH) ions. The product of their concentrations is equal to a constant value which at room temperature is approximately 1×10^{-14}. Neutral water contains an equal number of hydrogen and hydroxyl ions. The hydrogen ion concentration is, therefore, 1×10^{-7} grams of ionized hydrogen per liter.

When acid is added the hydrogen ion concentration increases with corresponding decrease in hydroxyl ion concentration. When an alkali is added the hydroxyl ion concentration increases, and the hydrogen ion concentration decreases. Since all acids and alkalies do not ionize alike, the quantity of acid or alkali does not give a direct measure of hydrogen ion concentration. Strong acids, as hydrochloric, and strong alkalies, as sodium hydroxide, are much more effective in changing hydrogen ion concentration than relatively weak materials, as carbonic acid and sodium carbonate.

For convenience, only the hydrogen ion concentration is recorded, whether the solution be acid or alkaline. Hydroxyl ion concentration may be found by dividing hydrogen ion concentration into 1×10^{-14}. Thus, if hydrogen ion concentration is 1×10^{-8} the hydroxyl ion is 1×10^{-6}. Hydrogen ion concentration is expressed in terms of pH value, equivalent to log (1/H ion concentration), that is, to the negative exponent. Thus if hydrogen ion concentration is 1×10^{-9}, pH value is 9. The lower the hydrogen ion concentration, the higher is pH value. In neutral water $p\text{H} = 7$; in water that is relatively ten times as alkaline $p\text{H} = 8$. If $p\text{H} = 6$ the water is relatively ten times as acid as at $p\text{H} = 7$. Table 3 shows hydrogen ion concentration, its equivalent pH value, and corresponding color of the universal indicator. Table 4 lists several indicators, their solution concentrations, and the color change for the pH range to which they apply.

This method of measuring acidity or alkalinity is useful in controlling corrosion and certain chemical reactions in treatment of feed and boiler water. For feedwater pH should be at least 7, and for boiler water at least 10.5. Fig. 1 shows relation between pH and solubility of iron in deaerated water.

Hardness.—For control purposes, total hardness is determined by adding standard soap solution to a bottle containing a measured amount of sample, shaking the bottle vigorously between additions of soap solution, the bottle lying on its side, until an unbroken lather is maintained for five minutes on the water surface. Volume of soap solution used is referred to a chart or multiplied by a factor. The result is expressed in parts per million, grains per gallon, or equivalent calcium carbonate.

Table 3.—Hydrogen Ion Concentration as Shown by Color Indicators

Hydrogen Ion Concentration, Gram-Mols per Liter *		pH		Color of Universal Indicator
1.0	10^{-0}	0		
0.1	10^{-1}	1	Acid Range	
0.01	10^{-2}	2		
0.001	10^{-3}	3		
0.000,1	10^{-4}	4		Red
0.000,01	10^{-5}	5		Pink
0.000,001	10^{-6}	6		Yellow
0.000,000,1	10^{-7}	7	Neutral	Greenish Yellow
				Green
0.000,000,01	10^{-8}	8	Alkaline Range	Blue
0.000,000,001	10^{-9}	9		Blue—Violet
0.000,000,000,1	10^{-10}	10		Purple
0.000,000,000,01	10^{-11}	11		
0.000,000,000,001	10^{-12}	12		
0.000,000,000,000,1	10^{-13}	13		
0.000,000,000,000,01	10^{-14}	14		

* For hydrogen ion 1 gram-mol = 1 gram, but for hydroxyl ion 1 gram-mol = 17 grams.

Table 4.—Colorimetric Indicator Solutions

Indicator	Concentration	pH Range	Color Change
Meta Cresol Purple	0.04%	1.2—2.8	Red—Yellow
Bromphenol Blue	.04	3.0—4.6	Yellow—Blue
Methyl Red	.02	4.4—6.0	Red—Yellow
Bromcresol Green	.04	4.0—5.6	Yellow—Blue
Bromcresol Purple	.04	5.2—6.8	Yellow—Purple
Bromthymol Blue	.04	6.0—7.6	Yellow—Blue
Phenol Red	.02	6.8—8.4	Yellow—Red
Cresol Red	.02	7.2—8.8	Yellow—Red
Thymol Blue	.04	8.0—9.6	Yellow—Blue
Phthalein Red	....	8.6—10.2	Yellow—Red
Tolyl Red	....	10.0—11.6	Red—Yellow
Parazo Orange	....	11.0—12.6	Yellow—Orange
Acyl Blue	....	12.0—13.6	Red—Blue

Actually, hardness consists of such materials as calcium and magnesium carbonate and bicarbonates, calcium and magnesium sulphates, and calcium and magnesium chlorides. These materials can be precipitated by boiling, and are known as temporary hardness. For example, the bicarbonates of calcium and magnesium are changed to carbonates, which are much less soluble. The remaining hardness is known as permanent hardness.

Chloride concentration is determined by titrating a measured volume of sample with standard silver nitrate solution, using potassium chromate as an indicator. The end point is indicated by a red coloration. The result is expressed in parts per million, grains per gallon of chlorine, or equivalent sodium chloride.

Equivalent Sodium Sulphate determination is useful in boiler water analyses. In control work, benzidine sulphate titration, or the turbidity method is used. The titration consists of adding an excess of benzidine sulphate to a measured sample of water. After standing, to allow complete precipitation of sulphate as benzidine sulphate, filter and wash precipitate. Titrate the precipitate with a standard sodium hydroxide solution, using phenolphthalein as the indicator.

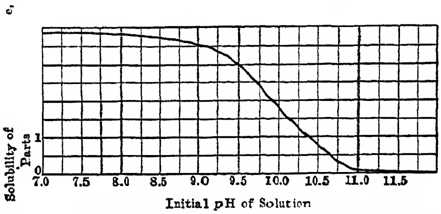


FIG. 1. Relation between *p*H and Solubility of Iron in Deaerated Water at Room Temperature

The turbidity method consists of adding hydrochloric acid and barium chloride to a measured sample of water, causing a white precipitate of barium sulphate to form. The sample is stirred to keep precipitate in suspension, and the mixture slowly poured into a graduated tube with a small light below it. When sufficient mixture has been added to just obscure the light filament, when looking down the tube, height of liquid is read, and equivalent sodium sulphate in parts per million or grains per gallon is estimated or read from the graduated tube.

Dissolved Oxygen is an important test in controlling deaeration of feedwater. It involves sampling water through a cooling coil to reduce temperature to below 70° F., flowing water from the coil through a glass-stoppered sample bottle to wash out any air not in the sample. The sample is then fixed with three reagents, usually manganous sulphate, alkaline potassium iodide, and sulphuric acid. A measured volume of the sample is titrated with a standard sodium thiosulphate solution, using starch as an indicator. If dissolved oxygen is absent there will be no blue coloration when the indicator is added. Result is expressed in cc. per liter or parts per million of dissolved oxygen.

Dissolved Solids may be estimated in several ways. In the laboratory they are determined by evaporating a measured volume of sample and weighing the dried residue. For boiler water, hydrometer, densimeter and conductivity tests are used, suitable calibrations being made for the type of water.

TURBIDITY tests are made by several methods, depending on the amount of suspended solids. For certain boiler waters containing considerable suspended matter, some type of turbidimeter may be used to regulate blow-down for suspended solids. The sulphate meter is operated by pouring liquid containing suspended matter into a tall glass cylinder until a light filament under the cylinder no longer is visible. Height of liquid in the cylinder is then read. Another method, for waters containing less suspended matter, involves the immersion of a graduated rod holding a wire at the end into the sample until the wire no longer can be seen.

References.—Standard Methods of Water Analyses, Am. Public Health Assoc., New York City; Power Station Chemistry Committee reports, Edison Electric Inst., New York City. Control Tests for Treatment of Feed and Boiler Water, by J. K. Rummel, *Jour.* Am. Water Works Assoc., Vol. 24, No. 12, Dec., 1932.

2. TREATMENT OF FEEDWATER

CAUSES OF SCALE FORMATION.—Hard scale and incrustations of softer deposits result from the presence or formation of insoluble solids in feed and boiler water. Certain slightly soluble solids, when treated with water-softening chemicals, or heated and concentrated in the boiler water, become less soluble and precipitate. The most objectionable are calcium sulphate and silica, which have a strong tendency to crystallize and precipitate, forming hard scale which is difficult to remove. Both may act as a cement for other insoluble matter and hasten the formation of a sufficiently heavy scale, which, due to

its poor heat conductivity, will cause overheating and failures of boiler tubes. Calcium sulphate, and, to a slightly less extent, silica, tend to form scale on the hottest tubes. Calcium carbonate is somewhat more likely to precipitate in the boiler water than on the tubes. Its deposits tend to be greater in the cooler parts of the boiler. Due to external heating of feedwater containing calcium bicarbonate, the less soluble calcium carbonate often is formed and deposited in heaters and pipe lines. With this chemical present, the same result may be obtained by continuous addition of caustic soda to feedwater. Calcium phosphate tends to deposit in feed lines when sodium phosphate is used as a treating agent. Tannates have been used to delay precipitation of calcium compounds in the feed system. In general calcium phosphate does not give serious trouble in boiler water, but periodic cleaning is advisable.

For explanation of the process of scale formation see Hall, A Physico-Chemical Study of Scale Formation and Boiler Water Conditioning, Bull. 24, Carnegie Inst. of Tech., Pittsburgh, Pa.

SCALE PREVENTION.—No single method of treatment can be recommended which will apply to all feedwaters. In general, as much as possible of insoluble and slightly soluble material should be removed from feedwater without unduly increasing soluble solids. Any residual scale-forming material then should be treated in the boiler water to form non-adherent precipitates. Mechanical cleaning of the boilers should be performed as often as necessary to avoid trouble.

Methods of External Treatment

External treatments include coagulation and filtration, sedimentation of suspended matter, lime-soda treatment, zeolite treatment and evaporation. Combinations of some of these treatments often are desirable.

COAGULATION, SEDIMENTATION, AND FILTRATION may be used alone or in conjunction with lime-soda or other treatments. The removal of suspended matter is an important part of the treating system. Coagulation and sedimentation are carried on in large basins, or in tanks, depending on quantity of water to be handled. Usual coagulants are iron sulphate (copperas), aluminum sulphate, sodium aluminate, and lime. Cold water filtration is carried on usually with gravity or pressure-type sand filters. In hot-water filtration, less soluble materials, as calcite or magnetite, should be used to avoid formation of calcium silicate, which may result in hard, dense scale. Sand filters usually are designed for a capacity of from 2 to 4 gal. of water per min. per sq. ft. of cross-sectional area.

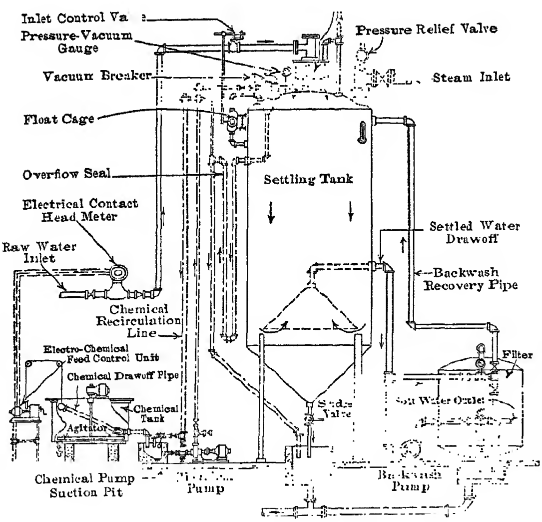


Fig. 2. Hot Lime Soda Water Softener (Permutit Co.)

LIME-SODA TREATMENT is applied in several ways, with considerable variation in design of equipment. The principal differences are in the temperature of water, cycle of operation (whether continuous or intermittent flow), and method of applying and agitating chemicals. Hot-process equipment usually consists of chemical mixing tanks, chemical proportioner (which introduces chemicals in proportion to flow of water), a deaerating type heater, placed above a reaction and settling tank, and a filter to remove unsettled suspended matter. The cycle usually is continuous. Water flows through

the heater into the top of the reaction and settling tank, where chemicals are introduced and thence to the bottom of the tank, where suspended matter settles and is blown out. The water then rises through a central duct and is discharged through the side of the tank, at a point below the water level. Treated water finally passes through a closed or pressure-type filter. Retention time of water in the tank is preferably not less than one hour.

In the cold process, equipment may be the same as in the hot process without the heater. Usually, it is desirable to have a longer reaction and settling time, and several treating tanks are provided. In these, water is treated, agitated, settled, and finally drawn off from the top to the filters. Each tank is treated in rotation. The cycle is so timed that treated water always flows from one tank to the filters while water in the others is being treated or settling. Filters frequently are of the open gravity type. Depending on analysis of raw water and excess of treating chemicals used, effluent water from a cold-process softener may have a hardness of from 2 to 5 grains per gal., or from the hot-process softener a hardness of from 1 to 3 grains per gal.

The chemicals used in these treatments are one or more coagulants (iron sulphate, aluminum sulphate, sodium aluminate), calcium hydroxide, and sodium carbonate. Coagulants are added either before or after lime and sodium carbonate, depending on the ease of coagulating suspended matter. Iron sulphate and aluminum sulphate both create acidity when added to neutral water, which must be corrected by alkaline chemicals. Sodium aluminate gives an alkaline reaction and besides being a coagulant for suspended matter, it has water-softening properties similar to that of sodium carbonate.

Calcium hydroxide or hydrated line combines with excess carbon dioxide, and reacts with calcium and magnesium bicarbonates to form less soluble calcium carbonate and magnesium hydroxide. Also when sodium carbonate is present, sodium hydroxide is formed. These reactions for lime are:

$2Ca(OH)_2$ Calcium hydroxide	$Mg(HCO_3)_2$ Magnesium bicarbonate	$2CaCO_3$ Calcium carbonate		$Mg(OH)_2 + 2H_2O$ Magnesium hydroxide
$Ca(OH)_2$ Calcium hydroxide	$Ca(HCO_3)_2$ Calcium bicarbonate	$2CaCO_3$ Calcium carbonate		
$Ca(OH)_2$ Calcium hydroxide	H_2CO_3 Carbonic acid	$CaCO_3$ Calcium carbonate		$2H_2O$
$Ca(OH)_2$ Calcium hydroxide	Na_2CO_3 Sodium carbonate	$CaCO_3$ Calcium carbonate	+	$2NaOH$ Sodium hydroxide

The principal use of sodium carbonate is to react with calcium sulphate to form a less objectionable scale-forming compound. Also, the treatment is effective in changing acid-forming salts to neutral salts. The resulting magnesium hydroxide and calcium carbonate are largely precipitated in the softener. The principal reactions are:

		$CaSO_4$ Calcium sulphate	+	Na_2CO_3 Sodium carbonate	=	Na_2SO_4 Sodium sulphate	+	$CaCO_3$ Calcium carbonate		
		$CaCl_2$ Calcium chloride	+	Na_2CO_3 Sodium carbonate	=	$2NaCl$ Sodium chloride	+	$CaCO_3$ Calcium carbonate		
$MgSO_4$ Magnesium sulphate	+	Na_2CO_3 Sodium carbonate	+	$Ca(OH)_2$ Calcium hydroxide	=	$Mg(OH)_2$ Magnesium hydroxide	+	$CaCO_3$ Calcium carbonate	+	Na_2SO_4 Sodium sulphate
$MgCl_2$ Magnesium chloride	+	Na_2CO_3 Sodium carbonate	+	$Ca(OH)_2$ Calcium hydroxide	=	$Mg(OH)_2$ Magnesium hydroxide	+	$CaCO_3$ Calcium carbonate	+	$2NaCl$ Sodium chloride

Note: Sodium hydroxide may be substituted for sodium carbonate and calcium hydroxide in the last two equations.

The amounts and kinds of coagulants added is regulated mainly by coagulation tests. Lime and sodium carbonate additions are regulated by chemical analysis and control tests. The control tests usually made are soap hardness and alkalinity.

After a complete analysis of raw water, theoretically required amounts of lime (calcium hydroxide) and soda (sodium carbonate) may be calculated from the reacting molecular weights of the compounds shown in above equations. See Table 5. Actually some excess of treating agent is desirable and due to impurities, an allowance of 5 to 10% should be made for lime and of 1 to 2% for sodium carbonate.

ZEOLITE TREATMENT.—The term "zeolite" is applied to such materials as have the property of base exchange. In water softening they exchange their sodium for calcium and magnesium in the raw water, these undesirable materials remaining with the

zeolite. Chemically, zeolites are hydrated silicates having a base such as sodium, a metal oxide as alumina, silica, and water of hydration. Their general composition is shown by the formula: $Na_2O \cdot Al_2O_3 \cdot (SiO_2)x \cdot (H_2O)n$. As used in water treatment zeolite is a hard granular material which may be of natural or synthetic origin. Synthetic material is more porous.

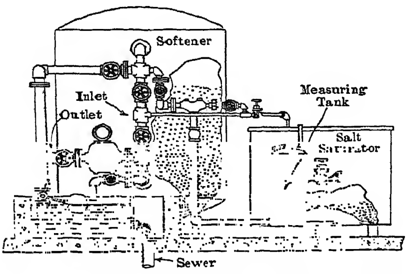


Fig. 3. Zeolite Water Softener (Wm. B. Scaife & Sons Co.)

A zeolite softener is similar to a pressure-type sand filter, except that a bed of zeolite replaces the fine sand. If raw water contains suspended matter, suitable filters should precede the zeolite softener, otherwise the zeolite may become coated. If water contain a large amount of temporary hardness, the zeolite treatment may be advantageously preceded by a cold lime coagulating and filtering treatment, resulting in less total solids in the treated feedwater. Hot water is destructive to zeolite, and temperature should not exceed 120° F.

Substituting Z for the zeolite radical, when water passes through a zeolite bed are

Sparingly Soluble Impurity	Treating Agent	Soluble Product		Insoluble Product
$CaSO_4$ Calcium sulphate	Na_2Z Sodium zeolite	Na_2SO_4 Sodium sulphate	+	CaZ Calcium zeolite
Calcium bicarbonate	Sodium zeolite	$2NaHCO_3$ Sodium bicarbonate	+	CaZ Calcium zeolite
$Mg(HCO_3)_2$ Magnesium bicarbonate	Na_2Z Sodium zeolite	$2NaHCO_3$ Sodium bicarbonate	+	MgZ Magnesium zeolite

Similar reactions occur with other calcium and magnesium compounds.

After a certain amount of raw water has passed through the zeolite the sodium must be replaced before the zeolite becomes completely exhausted and the hardness in the effluent water excessive. Regeneration is effected by washing the zeolite with strong salt solution for 30 to 40 minutes. The reactions are

Exhausted Zeolite	Treating Agent	Regenerated Zeolite	Soluble Waste Product
CaZ Calcium zeolite	$2NaCl$ Sodium chloride	Na_2Z Sodium zeolite	$CaCl_2$ Calcium chloride
MgZ Magnesium zeolite	$2NaCl$ Sodium chloride	Na_2Z Sodium zeolite	$MgCl_2$ Magnesium chloride

Calcium and magnesium chlorides from these reactions must be thoroughly washed out of the zeolite before resuming use of treated water.

When raw water contains large amounts of calcium and magnesium carbonates, a large amount of sodium carbonate alkalinity will be generated, which, in the absence of sufficient sodium sulphate, may promote caustic embrittlement. To avoid this, sufficient sulphate is added, either as sodium sulphate or by sulphuric acid treatment after the

Table 5.—Required Quantities of Reagents to Treat Calcium and Magnesium Compounds

One Part of	Requires Parts of	Produces Parts of	Parts of
$Ca(HCO_3)_2$.	0.457 $Ca(OH)_2$	1.235 $CaCO_3$	
$Ca(HCO_3)_2$.	0.494 NaOH	0.617 $CaCO_3$	0.654 Na_2CO_3
$CaCl_2$.	0.955 Na_2CO_3	0.901 $CaCO_3$	.053 NaCl
	0.779 Na_2CO_3	0.735 $CaCO_3$	.044
$Mg(HCO_3)_2$.	1.013 $Ca(OH)_2$	0.399 $Mg(OH)_2$	.368 $CaCO_3$
Mg(HC	1.093 NaOH	0.399 $Mg(OH)_2$	.449 Na_2CO_3
$MgSO_4$	0.881 Na_2CO_3	0.700 $MgCO_3$	.180 Na_2SO_4
$MgSO_4$	0.664 NaOH	0.484 $Mg(OH)_2$	.180 Na_2SO_4
$MgCl_2$	1.113 Na_2CO_3	0.885 $MgCO_3$	.227 NaCl
	0.840 NaOH	0.612 $Mg(OH)_2$	.227 NaCl

Note.—Lime and sodium hydroxide are the preferred reagents for treating the magnesiu compounds. A mixture of lime and sodium carbonate gives the same result.

zeolite treatment. The acid is added by automatic feeders, to insure against acidifying the feedwater. Phosphoric acid and acid phosphates have been added to reduce excess alkalinity and precipitate small amounts of calcium and magnesium passing the zeolite.

Natural treated zeolite material, usually known as greensand, weighs approximately 90 lb. per cu. ft. when dry. It has a rated exchange value of approximately 3000 grains per cu. ft. of hardness, as equivalent calcium carbonate. Synthetic zeolites weigh approximately 50 lb. per cu. ft. and have rated exchange values up to 9000 grains of hardness per cu. ft. The salt needed for regeneration varies with individual units, but for estimating, approximately 0.5 lb. of salt per 1000 grains of hardness may be used.

The zeolite treatment produces treated water with less hardness than any of the external chemical-treating methods. With a properly designed plant, hardness of treated water should not exceed 0.5 grain per gal. and should average less.

Evaporation of make-up water is especially useful in power plants where the percentage make-up is small and a minimum quantity of boiler deposit is desirable. This process produces purer water than any other process now available. The amount of make-up water which can be so prepared is limited only by economic considerations. See pp. 3–36 to 3–48 for a discussion of evaporators. Condensate from low-pressure boilers sometimes is used to supply make-up water for high-pressure boilers in which no appreciable quantity of scale can be tolerated. To assist continuous operation, feedwater to evaporators may be pretreated in the same manner as for boilers. In other cases it is preferred to crack scale from the evaporator tubes by the introduction of cold water when the scale is brittle. As with boilers, total solids in the water in the evaporator must be regulated to inhibit carrying undesirable quantities of solids into the steam.

Internal Boiler Water Treatment

Internal treatment consists mainly of maintaining desirable concentrations of chemicals in the boiler water. Chemicals to be added and concentrations to be maintained vary with the nature of feedwater and results of operation. The usual agents for inhibiting scale formation are sodium carbonate, sodium phosphate and sodium aluminate. Sodium hydroxide, tannates, and various prepared boiler compounds are used on occasion, or for some specific need. In general, prepared compounds are viewed with disfavor from both practical and economic standpoints.

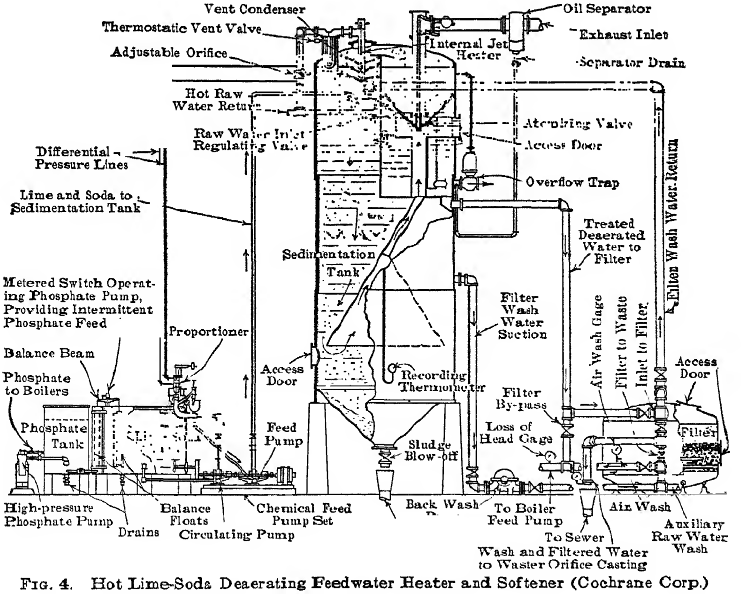


FIG. 4. Hot Lime-Soda Deaerating Feedwater Heater and Softener (Cochrane Corp.)

SODIUM CARBONATE is used to promote a desirable alkalinity and to inhibit formation of calcium sulphate scale. It also may retard formation of silica scale. In boilers, it hydrolyzes to form sodium hydroxide and CO_2 gas, the latter passing off with the steam. The extent of this reaction depends mainly on the amount of carbonate in the feedwater, but, in general, 70 to 90% of the sodium carbonate becomes sodium hydroxide.

Experiments indicate that under favorable conditions 2 to 3 grains of sodium carbonate in boiler water will inhibit calcium sulphate scale. The final concentration should be regulated by results of practical experience.

SODIUM PHOSPHATE is used principally to precipitate as tricalcium phosphate, calcium entering with the feedwater. This finely divided material has considerably less tendency to form objectionable deposits than the calcium compounds that otherwise would be present. A similar reaction may occur with magnesium, but in practice sufficient alkalinity usually is present to precipitate magnesium as magnesium hydroxide. The common forms of sodium phosphate are trisodium phosphate, Na_3PO_4; disodium phosphate, Na_2HPO_4; monosodium phosphate, NaH_2PO_4; sodium metaphosphate, $(NaPO_3)x$.

The less-alkaline phosphates, as monosodium phosphate, are used when there is excess alkalinity in feedwater or when calcium is present largely as calcium carbonate. Total alkalinity and total solids in boiler water thus are more easily controlled. Typical reactions of trisodium phosphate are

Soluble Impurity		Treating Agent	Insoluble Product		Soluble Product
$3CaCO_3$ Calcium carbonate	+	$2Na_3PO_4$ Trisodium phosphate	$Ca_3(PO_4)_2$ Calcium phosphate	+	$3Na_2CO_3$ Sodium carbonate
$3CaSO_4$ Calcium sulphate	+	$2Na_3PO_4$ Trisodium phosphate	$Ca_3(PO_4)_2$ Calcium phosphate	+	$3Na_2SO_4$ Sodium carbonate

Calcium phosphate thus formed has considerable tendency to adhere to feed lines. It generally is safer to add the phosphates direct to the boiler, or in intermittent doses, so that a minimum of precipitate is formed external to the boiler.

Provided there is sufficient alkalinity in the boiler water, the less-alkaline phosphates give the same type of reaction as above. The following is an example:

$3CaCO_3$	+	$2NaH_2PO_4$	+	$4NaOH$	=	$Ca_3(PO_4)_2$	+	$3Na_2CO_3$	+	$4H_2O$
Calcium carbonate		Monosodium phosphate		Sodium hydroxide		Calcium phosphate		Sodium carbonate		

Only a small amount of phosphate need be maintained in boiler water to inhibit scale. Phosphates, unlike sodium carbonate, do not lose efficiency by hydrolysis in boiler water. As with other forms of internal treatment it is important that boilers and other equipment in feedwater and steam systems be inspected and cleaned periodically.

SODIUM ALUMINATE usually is given the formula $Na_2Al_2O_4$, but in liquid form may contain a higher ratio of sodium to alumina. While generally recommended as a coagulant for external treatment, it also is used as a substitute or aid to sodium carbonate and sodium phosphate in internal treatments. It has many of the qualities of sodium carbonate. In addition it tends to make calcium and magnesium precipitates less adherent than if precipitated alone. Under certain conditions it will reduce silica concentration of boiler water, and form calcium or magnesium aluminum silicates, which are not generally adherent. However, under certain adverse conditions adherent silicates have formed, and the manufacturer of the material should be consulted.

For further information, see publications of National Aluminate Corp., Chicago.

Other Chemicals.—The use of other chemicals, as tannates and special boiler compounds usually is directed by vendors of the materials, or by consulting chemists.

3. EFFECTS OF IMPURE FEEDWATER

CORROSION.—Corrosion in boilers or feedwater equipment may be explained by the electro-chemical theory. See Eshbach, Handbook of Engineering Fundamentals, Section 11, forming Vol. 1 of this series.

The principal accelerators of corrosion in boilers are: 1. Dissolved oxygen. 2. Acids. 3. Surface deposits, especially those that are electro-negative to steel. 4. Unlike metal couples, as brass and steel. 5. Electrolytes, as strong salt solutions. Common methods of preventing corrosion are: *a*. Removal of dissolved gases in feedwater, especially dissolved oxygen and carbon dioxide. This can be done by deaerating processes. *b*. Neutralization of acids and maintenance of desirable alkalinity and hydrogen ion concentration (*p*H value) in feed and boiler water. See p. 6–68. *c*. Periodic mechanical cleaning

Protective paints may be applied if desired. *d.* Avoidance of excessive salt concentrations.

HIGH-TEMPERATURE CORROSION.—At higher temperatures, especially above 950° F., steam may react with iron according to the equation

$3Fe$	+	$4H_2O$	=	Fe_3O_4	+	$4H_2$
Iron		Steam		Black magnetic oxide		Hydrogen

At temperatures up to 950° F. this corrosion reaction is slow. Above this temperature the rate is accelerated, and alloys, like the Cr-Ni alloy KA2 are recommended. These materials form a protective oxide film on the metal surface, whereas steel does not. See J. K. Rummel, Corrosion by Superheated Steam (*Iron Age*, Dec. 5, 1929).

CORROSION FATIGUE.—Under certain conditions of corrosion and stress, metals may corrode in the form of deep sharp pits, which later develop into fissures or cracks. If the action is not stopped the parts attacked eventually will fail. The action may be produced by subjecting metal to ordinary corrosive conditions while under stress, which may be intermittent in nature. Microscopic examination shows the action to occur in a characteristic manner, producing cracks which are transcrystalline and in a nearly straight line. The products of corrosion fill the crack and the metal shows little or no elongation or distortion.

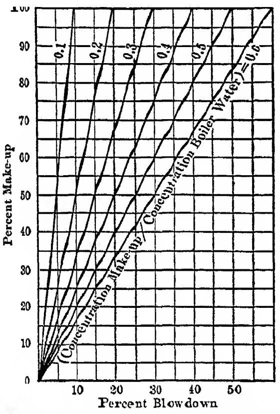
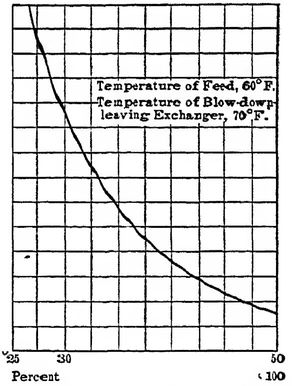


Fig. 5. Blow-down as Affected by Make-up and Ratios of Concentrations



G. 6. Percentage of Blow-down that can be Used to Heat Feed to 210° F.

CAUSTIC EMBRITTLEMENT.—With certain types of feedwater, cracks of a peculiar form may appear in the boiler plates, particularly at the riveted seams below the water line. These cracks are intercrystalline and do not follow the lines of the maximum stress, as they sometimes run past each other and around parts or islands of the plate. This action has been termed caustic embrittlement, as it occurs where concentrated sodium hydroxide (caustic soda) is present in the water in the boiler in the absence of certain other elements.

Embrittlement is inhibited when a certain ratio of sodium sulphate to sodium carbonate is maintained in the feedwater. This has led to ratios, given in Table 6, of sodium sulphate to total sodium hydroxide and sodium carbonate alkalinity, calculated to equivalent sodium carbonate, being recommended for different working pressures in the Suggested Rules for Care of Power Boilers of the A.S.M.E. Code.

Embrittlement is caused by concentration of caustic soda at joints and through the effect of the stress in the metal at the joints. The trouble experienced with caustic embrittlement was a factor that led to the use of fusion-welded drums. See F. G. Straub, Embrittlement in Boilers, Univ. of Ill. Engg. Expt. Station Bull. No. 216.

Table 6.—Recommended Ratios of Sodium Sulphate to Total Sodium Hydroxide

Working pressure of boiler, lb. gage	0 to 150	150 to 250	250 and over
Sodium sulphate	1	2	3
Total sodium hydroxide and carbonate alkalinity as equivalent sodium carbonate	1	1	1

FOAMING AND PRIMING.—Foaming may be described as the formation of a large amount of foam in the boiler, due to failure of steam bubbles to coalesce and break. It is accompanied by considerable increase in moisture content of the steam.

Priming is characterized by large amounts of water passing out of the boiler with the steam, usually in intermittent slugs, which endanger steam lines, turbines, and engines. It may occur simultaneously with foaming. High water levels in boilers promote priming.

Foaming and priming generally are caused by high concentration of dissolved and suspended solids, possibly accompanied by oil and soaps in boiler water, and sudden changes in boiler capacity. These conditions may be prevented by reducing boiler water concentrations by blow-down, elimination of sources of feedwater contamination, periodic cleaning of boiler, and proper regulation of water levels. Since operating conditions and the boiler equipment influence the amount and kind of solids which may be permitted in boiler water, no general concentration limits can be given.

REDUCTION OF CONCENTRATION BY BLOW-DOWN.—Maintenance of reasonably low concentrations in high-capacity boilers is facilitated by economical continuous blow-down, with or without flash tanks, and with one or more heat exchangers. Fig. 5 indicates percentage blow-down required for any given percentage of make-up and ratio of concentration in make-up to that in boiler water. Fig. 6 indicates maximum percentage blow-down that may be utilized to heat feedwater from 60 to 210° F. for various steam pressures in the boiler.

Blow-down also may be used to heat part or all of the air for combustion. Fig. 7 shows the amount that can be so used for various steam pressures. The curves are based on air heated from 80° F. to a temperature 100° F. less than saturation temperature of the steam, with water leaving the heat exchanger at 190° F., assuming 1 lb. of air per 1 lb. of steam. For other air-steam ratios multiply percent blow-down by (lb. of air/lb. of steam). Tubular air heaters with extended surface on the air side and the continuous blow-down flowing inside the tubes are economical in cost, space, weight, and draft loss. With a small blow-down, part of the air, as primary air in pulverized coal firing, may be heated by the blow-down for mill drying. With a blow-down of 20 to 30% of the steam, all the combustion air may be heated in this way, using extraction feed heaters, and an economizer to secure desired boiler efficiency. The blow-down water leaving the air heater may be still further reduced in temperature in a feedwater heat exchanger before going to waste.

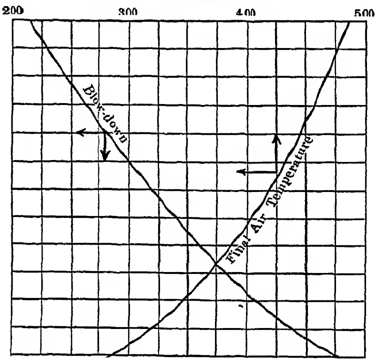


Fig. 7. Use of Blow-down for Heating Air

BOILER FURNACES *

By W. A. Carter

1. BURNING OF COAL

The complete combustion of coal in a furnace requires a sufficient supply of air, proper mixing of air and coal, adequate temperature to ignite the coal and maintain combustion, and sufficient time to complete the process. Hence, the furnace must provide a combustion chamber of sufficient volume and proper arrangement.

EXCESS AIR.—The amount of excess air beyond that theoretically required (see p. 4–07) depends on: 1. Composition and properties of the fuel. 2. General method of burning the combustible. 3. Manner in which air is supplied and mixed with fuel. 4. Arrangements and proportions used for grate and combustion chamber. 5. Furnace temperature desired, or that will be withstood by the furnace and related parts. Excess air will run from 25 to 50% for coal burned on grates or stokers. Half of the air, called primary air, controls rate of combustion; the remainder, or secondary air, controls completeness of combustion.

FURNACE TEMPERATURE LIMITATIONS.—Furnace temperature influences ignition of the fuel, rate of distillation of the volatile matter, proportion of CO formed, rate of combustion, and rates of heat absorption by various parts of the steam-generating unit. It is controlled by the amount of excess air supplied for combustion. If too high, furnace walls of refractory material will soften and deteriorate, and ash in the fuel bed may fuse into clinkers, which may adhere to walls and grates or stoker parts. This impairs operation and increases maintenance costs. Molten fly-ash carried in suspension in the gases may be deposited on refractory walls and cause slag erosion, or it may adhere to boiler tubes, thereby choking gas passages and limiting steaming rate.

COMBUSTION SPACE REQUIRED.—The size and arrangement of the combustion spaces should be such that flame will not be chilled and extinguished by contact with relatively cold surfaces before combustion is complete, and also such that all space will be used as fully as possible. Length of flame varies from a few inches with coke or anthracite, burning at low rates on grates or stokers, to 40 ft. or more with some volatile coals at high burning rates.

INFLUENCE OF CHARACTERISTICS OF COAL.—The physical characteristics of fuels may influence design or operation. Free-burning coals can be burned in a quiescent state; caking coals must be continually agitated while being burned, to permit passage of air. The ash of some coals fuses at temperatures of 2500° F. or higher and gives little trouble by clinkering; others have ash fusing at lower temperatures, causing much clinker trouble. Anthracite must be of uniform size to burn satisfactorily, while other coals burn best in run-of-mine or slack sizes. See U. S. Bureau of Mines Bull. 135, for quantitative information as to furnace volumes and lengths needed to burn three representative coals under various combustion conditions.

General Requirements of Combustion Equipment

SELECTION OF COMBUSTION EQUIPMENT.—The important factors to consider are: Grate area; draft for supplying primary air; provision for supply of secondary air; means for regulating and properly apportioning primary and secondary air; arrangement for mixing secondary air and volatile gases; means for maintaining combustion of the gases; combustion space of such volume as will provide for the length of flame; means for overcoming difficulties arising from coking of fuel or clinkering of ash; first cost of equipment; cost of labor, repairs, and maintenance.

Equipment should be selected with respect to: Character of fuel; ability to carry normal load at high efficiency and to meet maximum demand; ability to meet rapid changes in load; arrangement required for boiler, furnace, and other parts of the unit; ability to use preheated air, if desired; nuisance from smoke, cinder and fly-ash; loss from unburned fuel; loss from unit when fire is banked, and rapidity with which it can

* The author has drawn freely, by permission, on Heat Power Engineering by Barnard, Ellenwood and Hirshfeld (John Wiley & Sons) for many of the data in this chapter.

be brought from bank to full steaming capacity; durability of combustion apparatus and related equipment.

OPERATION.—To obtain maximum efficiency of combustion, the fuel bed should be uniform in thickness and character, and rate of feeding fuel should be constant. The evolution of gases should be unchanging in character and rapidity, sufficient primary air should be supplied to distill the volatile matter of the coal, with sufficient secondary air to complete the combustion process. Caked masses formed should be broken up constantly, and ash and clinker should be disposed of at a uniform rate.

2. HAND-FIRED GRATES

In hand-fired boilers, ideal uniform combustion cannot be maintained as firing is intermittent. The best condition of fuel bed is obtained when the coal is fired frequently, in small amounts, and with proper distribution; when caked masses of coal (if any) are broken up as rapidly as formed; and when ashes and clinker are not allowed to clog the fuel bed.

METHODS OF FIRING.—Three methods in general use for hand-firing boilers are:

Spread Firing.—A small amount of fresh coal is distributed evenly over the entire surface of the fire. It is commonly used with anthracite and other low-volatile coals.

Alternate Firing.—Fresh coal is fired on but one-half of the grate at a time. The freshly liberated volatile matter absorbs the necessary heat for combustion from the brighter parts of the fire. It is particularly suitable for non-caking coals.

Coking-Firing.—Especially suited to caking coals. Fresh coal is placed on the front edge of the fire and allowed to coke. After distillation is complete, the coke is spread over the fire. Lower rates of combustion are obtained with this method than with the other two.

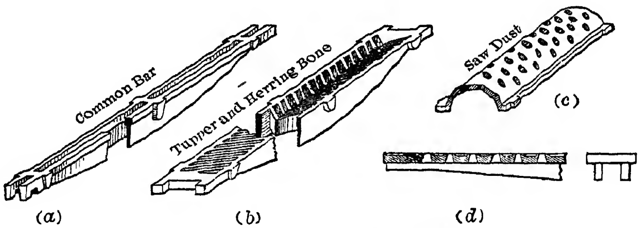


FIG. 1. Forms of Grate Bars

AIR APPORTIONMENT should be about equal between ash-pit door and openings in fire door. Draft should be adjusted by the boiler-outlet damper, either by hand or by an automatic damper regulator.

THICKNESS OF FUEL BED depends on furnace design; kind, size, and condition of coal; characteristics of ash; draft available, and steaming rate. In general, with natural draft, a thickness of from 4 to 8 in. is common with run-of-mine bituminous and anthracite buckwheat; with semi-bituminous coals, thickness is from 10 to 14 in. See U. S. Bureau of Mines Tech. Paper No. 80, Hand Firing Soft Coal under Power-plant Boilers; Univ. of Ill. Engg. Expt. Station Circular No. 7, Fuel Economy in Operation of Hand-fired Plants; Finding and Stopping Waste in Modern Boiler Rooms, Cochrane Corporation. Philadelphia.

COMBUSTION CHAMBERS in hand-fired boilers usually have firebrick walls. These, when hot, help to maintain the high temperature required for combustion. The combustion chamber often is of special form to compel the volatile gases to mix with secondary air.

With anthracite and other low-volatile coals a firebrick arch sometimes is sprung over the grate to assist ignition. The more volatile coals produce longer flames, and consequently the furnace must be made longer by setting it in front of the boiler as a Dutch-oven, or using a deflecting arch under the boiler to postpone contact of burning gases with the relatively cold boiler surface. The Dutch-oven furnace helps to attain smokeless combustion, except when the burning rate is high. Arrangements to mix air and volatile gases include multiple arches, piers or wing walls, and jets of air or steam injected through the front, sides, or bridge wall of the furnace.

COMBUSTION RATES depend on the characteristics of the coal and ash, thickness of fuel bed, total grate surface, air passage area of the grates, and the draft. Average rates, in lb. of fuel per sq. ft. of grate surface per hr. are:

Type of Draft	Anthracite	Semi-anthracite	Semi-bituminous	Bituminous Eastern	Bituminous Western	Lignite	Coke Breeze
Natural......	15	16	18	20	30	25	..
Forced.......	20	25	35	30	35	35	20

GRATE AREAS.—The old method of proportioning grate area, *i.e.*, $^1/_{30}$ to $^1/_{60}$ of boiler heating surface, may produce unsatisfactory results, and generally is inadvisable. The total width of grate surface usually is the distance between the walls of the setting; the length is made sufficient to provide the area necessary to burn the coal at reasonable combustion rates. For hand-fired anthracite, lengths up to 12 ft. have been used with dumping grates. For soft-coal the limit of length of ordinary grates usually is 6 ft.; if longer, it is difficult to clean fires. Table 1 gives the standard dimensions of grates for the setting shown in Fig. 9, p. 6-13.

Grate Bars should be so shaped as to present relatively small areas to the fire and expose large surfaces to be cooled by the currents of primary air. Fig. 1 shows several typical forms. The flat grate bar (*d*) for fine coal, is recessed on top to hold a layer of fine ash that protects the bar from heat and prevents adherence of clinker. Grate bars usually should not be over 3 ft. long. The width of air passage varies with the type and size of fuel as well as the type of draft, as shown in Table 2.

Shaking Grates are of many forms. The fire is more easily cleaned with shaking than with stationary grates, and it is not necessary to open furnace doors during cleaning. They cost more than ordinary grates, but with them from 1 to 5% better boiler and furnace efficiency is possible. Some forms of shaking grates also dump.

Inclined shaking grates, arranged to feed coal progressively from the front to a rear dump are known as hand-stokers.

3. STOKERS

The principal advantages of stokers over hand firing are: Continuous delivery of coal; progressive and gradual distillation of volatile matter; ability to obtain better performance and smokeless combustion because of the ease with which the operations can be regulated at all times; greater combustion capacity obtainable in a furnace; ability to meet varying load demands quickly; ability to burn poorer and cheaper grades of coal with less smoke and higher efficiency; relief of operators from strenuous duties, thus permitting more time for adjusting operating conditions; decreased labor costs in large boiler plants where the number of firemen can be reduced.

CLASSIFICATION.—Stokers are classified as overfeed or underfeed. Certain stokers of each type require forced draft; others operate with natural draft.

Overfeed Stokers include: 1. Front-feed, inclined-grate stokers; coal enters at the front, and is fed down an incline to the ash dump at the bottom; 2. Double-inclined,

Table 1.—Dimensions of Grates for Horizontal Tubular Boilers for Various Classes of Coal

(For hand-firing; combustion rate not over 25 lb. of coal per sq. ft. of grate surface per hr.)

Diameter of Boiler, in.	Length of Tubes, ft.	Size of Grates: Width	Size of Grates: Length	Size of Grates: Area, sq. ft.	Height of Bottom of Boiler, in.: Volatile > 35%: Above grate	Volatile > 35%: Above bridge-wall	Volatile, 18-35%: Above grate	Volatile, 18-35%: Above bridge-wall	Volatile < 18%: Above grate	Volatile < 18%: Above bridge-wall
54	14	4′-0″	4′-0″	16	32	12	28 1/2	11	28	10
54	16	4′-0″	4′-0″	16						
60	16	4′-6″	4′-6″	20 1/4	36	14	32	12	28	10
60	18	4′-6″	5′-0″	22 1/2						
66	16	5′-0″	5′-0″	25	40	16	36	14	32	12
66	18	5′-0″	5′-6″	27 1/2						
72	16	5′-6″	5′-6″	30 1/4	44	18	40	16	36	14
72	18	5′-6″	6′-0″	33						
72	20	5′-6″	6′-6″	35 3/4						
78	16	6′-0″	6′-0″	36	48	20	44	18	40	16
78	18	6′-0″	6′-6″	39						
78	20	6′-0″	7′-0″	42						
84	18	6′-6″	7′-0″	45 1/2	52	22	48	20	44	18
84	20	6′-6″	7′-0″	45 1/2						

side-feed stokers; coal is fed from both sides, down inclined grates to a refuse pocket at the center. 3. Chain-grate or traveling-grate stokers; the entire coal bed moves horizontally from front to rear.

Underfeed Stokers include: 1. Single-retort stokers, usually horizontal, with lateral ash dumps. 2. Multiple-retort stokers, usually inclined with refuse discharge at the rear.

Table 3 shows types of stokers generally most suitable for the various fuels, as given by T. A. Marsh in Design and Application of Traveling-Grate Stokers (*Power*, Feb. 21, 1928). Table 3 should be considered as tentative, as the art of burning coal is still (1935) in the development stage. For example, the use of preheated air now permits fuels high in moisture to be burned by methods heretofore partly or wholly inapplicable.

Overfeed Stokers

Inclined overfeed stokers usually operate with natural draft. A coking arch at the front of the furnace, maintained at a high temperature, reflects heat to and distils volatile gases from the entering coal. Air, heated or otherwise, usually is admitted with coal under the arch. As a rule, these stokers require more attention than other types and seldom are used on boilers larger than 600 Hp.

Practically all kinds of coal, sawdust, tan bark, and hog fuel can be burned in these stokers, but they principally are used with high-volatile, high-ash mid-western coal. Average combustion rate with free-burning coals is from 15 to 25 lb. per sq. ft. of horizontal projected grate surface per hr., with a maximum of 35 lb. With caking coal, the maximum combustion rate is 25 lb.

INCLINED FRONT-FEED STOKERS include a hopper, coal-pusher feeding device, dead plate, coking arch, and inlet for secondary air under the arch. The action of the pushers and grate bars can be so regulated that when the coal arrives at the ash table, it has been completely burned.

DOUBLE-INCLINED SIDE-FEED STOKERS have coal magazines at each side of the furnace. These feed coal to a coking plate, where it meets heated secondary air brought through a refractory arch that covers the entire stoker. The grate bars are inclined, each alternate bar being in constant motion to feed coal down to the clinker

Table 2.—Specifications for Hand-Fired Grates

From Finding and Stopping Waste in Modern Boiler Rooms (Cochrane Corp.)

Kind of Coal	Service	Kind of Plant	Type of Grate	Size of Coal	Width of Openings in Grate, in.	Free Air Space, Approximate Percent of Total Grate Area	Kind of Draft.
Bituminous	High Pressure	Industrial	Shaking....	Run-of-mine....	3/8	43	These widths of grate openings will work satisfactorily under natural draft. Forced draft may be used without change of the grates where steaming rates above rating are desired.
				Slack..........	1/4	35	
		Heating..	Shaking....	Run-of-mine....	3/8	43	
				Slack..........	1/4	35	
	Low Pressure	Heating..	Shaking....	Run-of-mine ...	3/8	43	
				Slack	1/4	35	
			Shaking and dumping	Run-of-mine ...	3/8	44	
				Slack	1/4	38	
Anthracite.	High Pressure	Industrial	Dumping...	No. 1 buckwheat	1/4	32	Natural
				No. 2 buckwheat	1/8	19	Forced
				No. 2 buckwheat	1/4	32	Strong natural
				No. 3 buckwheat	3/32	12	Forced
				Culm..........	5/64	7	Forced
		Heating..	Dumping...	No. 1 buckwheat	1/4	32	Natural
				No. 2 buckwheat	1/8	19	Forced
				No. 2 buckwheat	1/4	32	Strong natural
				No. 3 buckwheat	3/32	12	Forced
				Culm..........	5/64	7	Forced
			Shaking and dumping	Pea............	3/8	44	Natural
				No. 1 buckwheat	1/4	38	Natural
				No. 2 buckwheat	5/32	29	Forced
				No. 2 buckwheat	1/4	38	Strong natural
	Low Pressure	Heating..	Shaking and dumping	Pea............	3/8	44	Natural
				No. 1 buckwheat	1/4	38	Natural
				No. 1 buckwheat	5/32	29	Forced
				No. 2 buckwheat	1/4	38	Strong natural

grinder. Exhaust steam from the stoker engine sometimes is admitted to the grinder to assist in breaking the refuse of clinkering coal.

Stokers of this type have large coking spaces, ample coking arches and large combustion chambers. Ordinarily they are satisfactory for both uniform and varying loads, but at high combustion rates and with certain types of coal, the fuel may avalanch.

Chain- and Traveling-Grate Stokers

CHAIN- AND TRAVELING-GRATE STOKERS comprise series of small links, forming a broad endless belt conveyor carried on rolls or skids. In the traveling-grate type, crossbars, extending from endless chains on either side of the furnace, support short interlocking grate bars. Both types are driven by sprockets at variable speeds, in conformity with the load on the boiler. Raw fuel is fed at one end and discharged as burned-out refuse at the other. The fuel bed is undisturbed while passing through the furnace. Natural or forced draft may be used, depending on the design. A minimum ash content of 7% is necessary to protect the back end of the stoker from heat. With a properly-designed furnace, this type of stoker can burn high-volatile coals without smoke, with either natural or forced draft, and also non-caking, clinkering coal, high in ash. Special designs can burn small-size anthracite and breeze, using forced draft. Modern (1935) designs are relatively free from siftings along the stoker sides, and from cold air past the sides, back end, and through the rear portion of the grate.

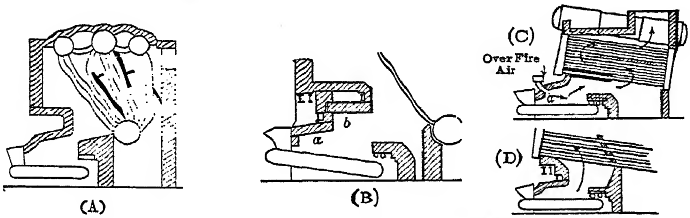


Fig. 2. Arch Arrangements with Natural Draft Chain-grate Stokers

Stokers of this type are relatively costly, but require little attention, and maintenance is low. They are not so well adapted as the underfeed stokers to meet sudden, heavy variations in demands for steam, unless forced draft is used.

Arches over the fire are necessary to cause mixing of gases from the rear of the grate, which are deficient in air, with excess air from the front. Another function is to maintain sufficient temperature to support combustion and to radiate heat to the front of the fuel bed to ignite entering fuel and distil volatile matter from it. Arches also prevent carrying away by a strong draft much of the fly fuel, which otherwise would be lost. Fig. 2 shows typical installations.

Secondary Air introduced under the front arch prolongs its life. A fan is preferable to a steam jet or induced draft.

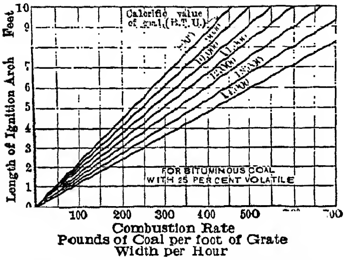
Combustion Space Requirements.—Adequate combustion chamber volume ordinarily should be provided by suitably locating boiler surface with respect to the grate according to recommended dimensions in Table 4.

Table 3.—Application of Stokers for Various Fuels

Fuel	Preferable Stoker
Anthracite	Traveling grate,* forced draft
Coke Breeze	Traveling grate,* forced draft
Semi-anthracite	Traveling grate,* forced draft
Semi-bituminous (coking)	Underfeed and inclined overfeed
Bituminous (coking)	Underfeed and inclined overfeed
Bituminous (free-burning, ash > 10 or 12%)	Traveling grate,* forced or natural draft
Bituminous (free-burning, ash < 10 or 12%)	Traveling grate,* underfeed or side inclined †
Sub-bituminous	Traveling grate,* forced or natural draft
Lignite	Traveling grate,* forced or natural draft

* "Traveling grate," is used to cover chain grates as well as traveling carrier-bar stokers.
† If ash fuses at a temperature below 2400° F., traveling grates are preferable. If the percentage of ash is less than 7, underfeed stokers are preferable.

Air Leakage around the grate may be minimized by: 1. Adjustable ledge plates to seal gaps between sides of stoker and furnace wall. 2. A tight ashpit to reduce infiltration at rear of stoker. 3. A well-fitted damper at the rear, between upper and lower runs. 4. A seal below lower run. A water-back, connected in the boiler circuit, set into an overhanging bridge wall, close to the grate, will compress the back of the fuel bed, and increase its density, thus decreasing air infiltration at this point. It also will reduce the amount of unburned fuel discharged to the ashpit, protect the bridge wall, and prevent adherence of clinker. Sidewall water-boxes may be necessary to prevent clinker building up on the furnace walls, which then may cause increased air leakage.



G. 3. Relation of Length of Arch and Combustion Rate

Operation.—The coal hopper outlet gate should be adjusted for proper thickness of fuel bed for the grade of coal burned, *i.e.*, 2 1/2 to 5 in. for fine anthracite, 4 to 6 in. for mid-west bituminous and lignite coals. The bed should be as thin as possible consistent with ignition and burning out at the proper point. Stoker speed and draft should be varied for load variations. For best results with the more volatile coals, coal that will pass through a 1 in. ring should be used, although screenings up to 2 in. will burn satisfactorily. A fuel bed of uniform density, offering correct resistance to air flow, is obtained with coal containing 50% fines. The addition of 3% moisture to the coal before firing will reduce sifting and blow-holes in the fire, and thereby reduce excess air and improve ignition. This extra moisture, reduces overall efficiency only a few tenths of 1%.

NATURAL-DRAFT CHAIN-GRATE STOKERS.—Free burning, high-volatile bituminous and sub-bituminous coals and lignite can be burned with high efficiency on natural-draft chain-grate stokers. The design of the arch depends on the percentage of volatile matter and heating value of the fuel, the combustion rate and the stoker length. With natural draft, an arch height of about 36 in. at the front has been found to give good results. Fig. 3 (T. A. Marsh, *Elec. World*, Oct. 23, 1920) shows the relation of arch length to ignition rate. The intersection of the ordinate corresponding to the product (combustion rate × stoker length) with the curve representing calorific value of fuel gives the length of the arch in feet. Fig. 2 shows several arrangements of furnaces for natural-draft chain-grate stokers. Turbulence of the gases is insured by the arches *a* and *b* in furnace *A*, and by increased velocity of gases in the narrow throats of the other furnaces.

Table 4.—Height of Settings for Various Types of Stoker-equipped Boilers *
(From Heat Power Engineering by Barnard, Ellenwood and Hirshfeld)

Type of Boiler	Underfeed Stoker				Chain Grate Stoker				Overfeed Stoker			
	Multiple Retort		Single Retort		Natural Draft		Forced Draft §		Side Feed		Front Feed	
	Types‡ 4, 6, 11, 13, 15, 16		Types‡ 2, 4, 5, 11, 16		Types‡ 1, 2, 7, 8, 10		Types‡ 1, 2, 3, 7, 8, 10		Types‡ 4, 12		Types‡ 9, 14	
	Min.†	P.M.†	Min.†	P.M.†	Min.†	P.M.†	Min.†	P.M.†	Min.†	P.M.†	Min.†	P.M.†
Water Tube												
Horizontal, all sizes	11′ 0″	13′ 0″	9′ 0″	11′ 0″	10′ 0″	12′ 0″	12′ 0″	14′ 0″	9′ 0″	11′ 0″	9′ 0″	11′ 0″
Inclined (H.M.D.), all sizes†	7′ 6″	8′ 6″	6′ 6″	8′ 6″	6′ 0″	8′ 0″	7′ 0″	8′ 0″	5′ 0″	7′ 0″	6′ 6″	8′ 0″
Inclined (V.M.D.), all sizes†	6′ 0″	7′ 0″	5′ 0″	7′ 0″	4′ 0″	5′ 0″	6′ 0″	8′ 0″	3′ 6″	5′ 0″	4′ 0″	5′ 6″
Vertical (H.M.D.), all sizes†	3′ 6″	5′ 0″	3′ 6″	5′ 0″	3′ 6″	4′ 6″	4′ 0″	5′ 0″	3′ 0″	4′ 0″	3′ 6″	5′ 0″
Vertical (V.M.D.), 1500 sq. ft.†	4′ 6″	5′ 0″	4′ 6″	5′ 0″	4′ 6″	5′ 0″	5′ 0″	5′ 6″	3′ 3″	......	3′ 6″	4′ 6″
Vertical (V.M.D.), 2500 sq. ft.†	5′ 6″	6′ 0″	5′ 6″	6′ 0″	4′ 6″	5′ 0″	5′ 0″	5′ 6″	3′ 3″	......	3′ 6″	4′ 6″
Vertical (V.M.D.), 5000 sq. ft.†	6′ 0″	6′ 6″	6′ 0″	6′ 6″	4′ 6″	5′ 0″	6′ 0″	6′ 6″	3′ 3″	......	3′ 6″	4′ 6″
Horizontal Return Tubular												
72 in.	8′ 0″	10′ 0″	7′ 6″	8′ 6″	7′ 0″	8′ 0″	8′ 0″	10′ 0″	7′ 0″	8′ 0″	6′ 0″	8′ 0″
84 in.	8′ 0″	10′ 0″	7′ 6″	8′ 6″	7′ 0″	8′ 0″	8′ 0″	10′ 0″	7′ 0″	9′ 0″	6′ 0″	8′ 0″

* Setting heights are defined as follows: *Water-tube Boilers:* Horizontal tubes, floor line to bottom of header above stoker; Inclined tubes (H.M.D.), Vertical tubes (H.M.D.), floor line to center of mud drum; Inclined tubes (V.M.D.), Vertical tubes (V.M.D.), floor line to top of mud drum. *Horizontal Return Tubular Boilers:* Floor line to under side of shell

† H.M.D. = horizontal mud drum; V.M.D. = vertical mud drum; Min. = absolute minimum; P.M. = preferred minimum.

‡ Types: 1. Babcock & Wilcox; 2. Burke; 3. Coxe; 4. Detroit; 5. Type E; 6. Frederick; 7. Green; 8. Harrington; 9. Huber; 10. Illinois; 11. Jones; 12. Murphy; 13. Riley; 14. Roney; 15. Taylor; 16. Westinghouse.

§ When burning coke breeze and anthracite fines, the setting heights indicated should be materially increased to provide for proper arch and furnace design.

Combustion Rates for most efficient operation with free-burning coal range from 20 to 30 lb. per sq. ft. of grate surface per hr., with a maximum rate of 40 lb. and a minimum of 5 lb. Draft loss, up to combustion rates of 35 lb., is approximately 0.1 in. of water per 10 lb. of coal per hr. per sq. ft. of grate, the loss increasing at higher combustion rates.

Operating Results possible with proper operation, without an economizer or preheater, are monthly efficiencies of boiler and furnace of over 70%, with CO_2 at the boiler outlet of 12 to 13%. Combustible in refuse should range from 15 to 25% at combustion rates of 25 to 40 lb. per sq. ft. per hr.

FORCED-DRAFT TRAVELING-GRATE STOKERS differ from natural-draft stokers in that a series of transverse independent forced-draft compartments, under the upper run of the traveling grate, are supplied, by a fan, with air under pressure from an air duct along the side of the boiler. Connections between the duct and compartments have dampers to control under-fire pressures in the various compartments. If the furnace has but a single arch, maximum under-fire pressure is carried only in the front compartment, the pressure tapering off to nearly zero in the rear compartment as shown in Fig. 4. In furnaces with front and rear arches, maximum under-fire pressure is at about two-thirds of the distance to the rear, pressures of $^1/_4$ to $^1/_2$ in. of water being carried in front and rear compartments.

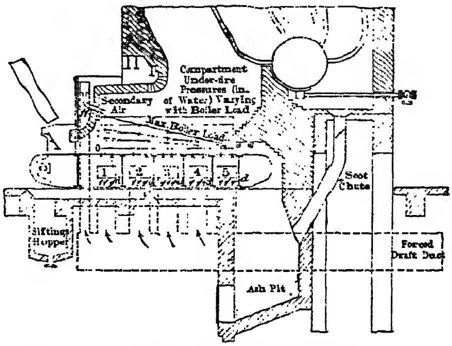


Fig. 4. Forced-draft Traveling-grate Stoker with Independent Air Compartments

Used with Low-volatile Coals.—For small-size anthracite, air openings in the grate must be small enough to prevent sifting. Special arrangements are necessary for primary kindling of the coal before it reaches the first forced-draft air compartment, utilizing heat radiated from the hot refractory surface of the arch. Entering fuel must "see" the arch through a greater angle, as *A* in Fig. 5 than that at *B*, through which it sees any relatively cold surface. In one stoker design, a small suction compartment at the front of the stoker draws some hot furnace gas down through the fresh fuel to ignite quickly moist or low-volatile fuels.

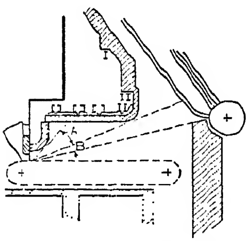


Fig. 5. Arch Arrangement with Traveling-grate Stoker for Anthracite

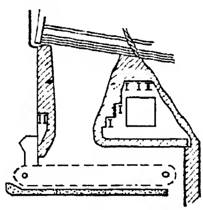


Fig. 6. Arch Arrangment to Reduce Stratification and Fly Ash

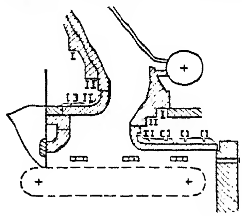


Fig. 7. Traveling-grate Stoker with Front and Rear Arches

Stratification of gases and carrying of fly ash by the furnace gases can be overcome by introducing air over the fire or placing the arch at the rear. See Fig. 6.

Front and rear arches often are used to form a narrow throat in which gases from the front and rear portions of the grate are mixed. Combustion is completed in the upper combustion chamber. See Fig. 7.

Used with Bituminous Coals and Lignite.—Forced draft under traveling-grate stokers permits higher combustion rates of free-burning coals than does natural draft. Efficiency curves are higher (5 to 6%). They also are flatter. Arches are smaller than with natural-draft stokers, but they must be set higher to prevent erosion. Side-wall water-boxes

and water-backs are necessary. Caking and coking coals that could not be burned on these stokers with natural draft are burned successfully with forced draft by reason of the air pressure breaking up the fuel bed. Water-cooled arches and side walls will avoid rapid destruction of brickwork by heat. An additional arch over the rear of the stoker is desirable when the coal varies in quality.

Combustion Rates for best results with bituminous coal should range from 30 to 40 lb. per sq. ft. per hr., with a maximum of 60 lb. when ash content ranges from 10 to 25%. A survey by N.E.L.A. in 1927 showed an average combustion rate of 43.5 lb. and average stoker maintenance cost of 4 cts. per ton of coal burned. With anthracite or coke breeze combustion rates range from 30 to 38 lb. per sq. f grate per h forced draft of 1.5 to 2 in. of water, with maximum and minimum rates of 55 and 10 lb. respectively.

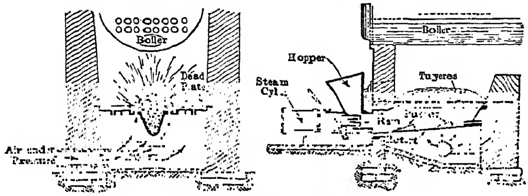


Fig. 8. Simple Horizontal Single-retort Stoker

Operating Results with bituminous coal should show combined boiler and furnace monthly efficiencies of 70 to 77%; CO_2 of 12 to 15%; combustible in refuse, 10 to 20%. With anthracite and coke breeze, monthly efficiencies of boiler and furnace should range from 72 to 76%.

Underfeed Stokers

Underfeed stokers operate at combustion rates as high as 110 lb. per sq. ft. of grate per hr. if ash fusion temperature is not below 2400° F. The field of the underfeed stoker comprises bituminous and semi-bituminous caking or free-burning coals, and to a lesser extent other grades of coal, including culm, coke breeze and small-size anthracite mixed with bituminous coal.

The essential principle of the underfeed stoker is a reciprocating ram or rams which feed coal from hoppers at the front of the furnace into the bottom of horizontal or slightly inclined retorts. The raw coal is underneath burning coal at the top of the fuel bed, which distils the volatile matter from the fresh coal. The liberated gases pass upwards through the burning coal, and are burned with air entering through tuyeres at the upper edges of the retorts. The coke which remains after distillation of the gases gradually is pushed upwards by entering fresh fuel and burns on the surface of the fuel bed. The entire fuel bed is worked toward the rear of the stoker or on to dead plates at the sides of the retort, ash and refuse being discharged into an ash hopper or removed by hand.

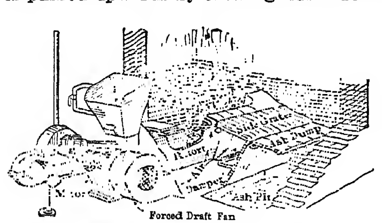


Fig. 9. Single-retort Stoker with Lateral Grates

Forced draft always is necessary. Rams and pushers, and sometimes also the ash disposal equipment, are driven by a motor or engine. Fuel and air supply can be regulated automatically by variations in steam pressure. Arches are unnecessary and considerable heat is transmitted to the boiler by radiation. This results in a relatively low temperature of gases passing through the boiler, even at high combustion rates.

SIMPLE SINGLE-RETORT STOKERS, Fig. 8, use a steam-driven ram or a screw feed, together with supplementary adjustable-stroke pushers, to properly distribute coal in the retort. From the surface of the fuel bed, refuse is deposited on dead plates whence it is removed by hand through doors in the front. In some designs the dead plates may be dropped to dump to the ashpit. Access doors on the sides of the furnace are unnecessary.

At moderate combustion rates, even with high-volatile coals, combustion is complete within a short distance of the surface of the fuel bed. The capacity of these stokers is from 800 to 1200 lb. of coal per hr. For greater capacities, two or three stokers may be set in a single furnace.

SINGLE-RETORT STOKERS WITH LATERAL GRATES resemble simple single retort stokers, except that stationary or movable overfeed grates are interposed between

the retort and the dead plates or dump plates. Air to the overfeed grates should be suitably regulated. The capacity of these stokers ranges from 1200 to 9000 lb. of coal per hr. The feed ram may be steam or hydraulically driven, or a motor-driven crank-shaft and connecting-rod may be used. This type of stoker may be used in boilers set side by side in continuous batteries, as no access doors in the sides of the furnace are necessary.

MULTIPLE-RETORT UNDERFEED STOKERS occupy the full width of the furnace. The fuel bed constantly moves from front to rear, and refuse is fed continuously to an ash dump. See Fig. 10. These stokers are from 6 to 28 ft. or more wide, with 3 to 16 retorts, and from 8 to 27 ft. or more long. Each retort may have from 13 to 69 or more replaceable tuyeres. Underfeed stokers can operate at higher combustion rates than other stokers, and in large units occupy a greater proportion of the area under the boiler. For a given rate of steam generation, they require less heat-absorbing surface, and permit individual units to have high steam generating capacity. These stokers can be brought quickly from bank to full capacity and can meet wide and rapid changes in load. Some furnaces have a stoker at each end discharging to a common ashpit.

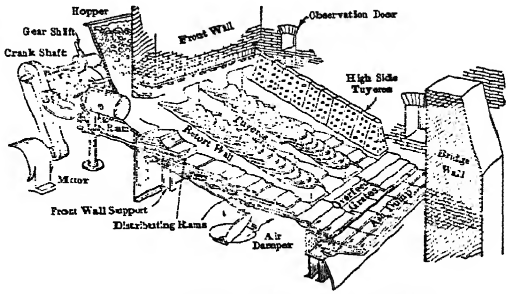


Fig. 10. Multiple-retort Underfeed Stoker

Control of the shape of the fuel bed to give proper air distribution is by adjustment of the length of pusher strokes and speeds of the various groups of rams. This also keeps the fuel bed open and free of clinkers. The active area of the fuel bed may be zoned, with independent regulation of air supply to each zone. A typical arrangement is shown in Fig. 11.

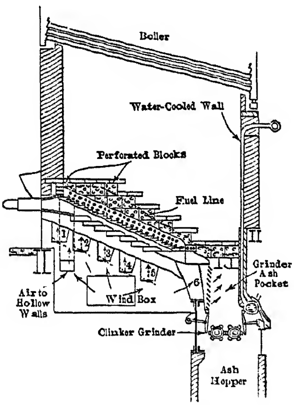


Fig. 11. Underfeed Stoker with Zoned Air Supply

Refuse discharging equipment comprises simple dump plates, double dumping grates, rocker plates and clinker grinders. With clinker grinders, the final combustible in the refuse can be reduced to 5% if the grinder pocket is large enough to hold ash for 12 hr., and air is forced through its walls. Shortening the time of burning out refuse to 6 hr. will raise the combustible to 15%. With dump grates, combustible in refuse may be from 15 to 25%.

Furnace walls for high combustion rates must withstand high furnace temperatures and erosive and slagging action of molten fly ash. Materials used are special grades of firebrick, silicon carbide blocks (if ash is not high in iron oxide), hollow perforated blocks through which secondary air is discharged, or water-cooled refractory or metallic blocks (see p. 6—95). The walls may be made hollow, and primary combustion air circulated through them. Protection from erosion and adhesion of molten clinkers may be obtained by the use of high air-cooled side-wall tuyeres (see Fig. 10) or water-cooled metallic surfaces (see Fig. 12).

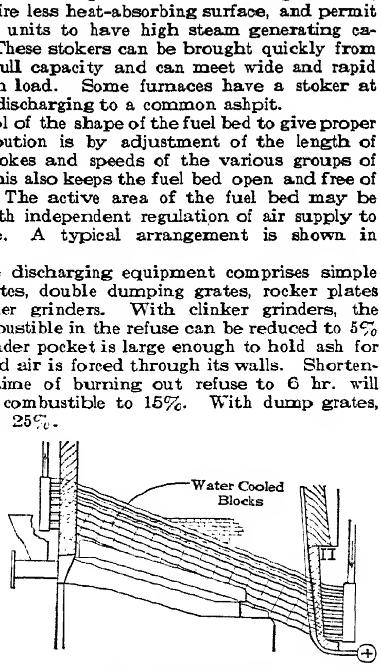


Fig. 12. Underfeed Stoker with Water-cooled Walls

Boilers fitted with underfeed stokers must be set in batteries of not more than two, as

access doors for inspection and cleaning of side walls are necessary in at least one side of the furnace.

Water-cooled underfeed stokers have been developed to burn low grade mid-western semi-bituminous coal, with ash fusion temperature as low as 1900° F., at a rate of 48 lb. per sq. ft. of grate surface per hr. (*Mech. Engg.*, Dec., 1935). Stoker tuyeres are cooled by forced circulation of water through groups of three tubes laid lengthwise of each tuyere stack, extending downward over stationary extension grates to a header near the clinker grinders. Groups of shorter tubes protect the remainder of the extension grates that register with the lower ends of the retorts. Side and rear furnace walls of such installations should be water-cooled to withstand the action of ash with such low fusion temperature.

Combustion Rates may range from bank to 75 lb. of coal per sq. ft. of projected grate surface per hr. The upper limit depends on kind of coal, furnace design, and available draft. With zoned-air control, combustion rates as high as 110 lb. have been carried. Without zoned-air control, best operation is at combustion rates of 35 to 45 lb., although rates as high as 90 lb. have been carried satisfactorily. The usual average in central stations, according to an extensive N.E.L.A. survey in 1928, was 50 lb.; average total stoker maintenance cost was 9.5 cts. per ton of coal burned.

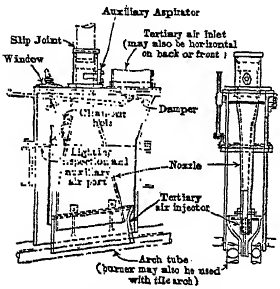


FIG. 13. Streamline Fan-tail Burner

Operating Conditions.—Excess air required with underfeed stokers is relatively low, but it should not be reduced to a point where boiler exit gases contain CO, or furnace temperatures are greater than furnace walls can withstand. Forced-draft pressures range from $^3/_4$ to 1 in. of water per 10 lb. of coal burned per sq. ft. of projected grate surface per hr. Air preheated to 300 to 500° F. sometimes is used. The closure of stoker air passages by expansion and growth of metals must be avoided by proper design and material. Prohibitive stresses and distortion also must be avoided.

Power required to operate such stokers may be, under extreme conditions, as much as $^3/_4$ to 1 Hp. per retort, burning from 700 to 1100 lb. of coal per hr.

Gross Efficiency of large steam-generating units with economizers, but without air preheaters, and equipped with multiple-retort stokers ranges from 90% at low loads to 75% at high loads. Under such conditions, excess air will vary from 20 to 10%.

Manufacturers of representative stokers are: *Overfeed Stokers:* Detroit Stoker Co., Detroit; Riley Stoker Corp., Worcester, Mass. *Chain- and Traveling-grate Stokers:* Babcock and Wilcox Co., New York; Combustion Engineering Co., New York; Laclede Stoker Co., St. Louis; Riley Stoker Co., Worcester, Mass. *Underfeed Stokers:* American Engg. Co., Philadelphia; Combustion Engg. Co., New York; Detroit Stoker Co., Detroit; Hoffman Combustion Engg. Co., Detroit; Riley Stoker Corp., Worcester, Mass.; Westinghouse Elec. and Mfg. Co., East Pittsburgh, Pa.; Whiting Corp., Harvey, Ill.

4. PULVERIZED COAL*

To insure complete combustion, the coarser coal particles must have motion relative to the gases surrounding them, which removes products of combustion and volatile matter, and supplies additional oxygen. This is accomplished by intensive mixing of the pulverized coal, resulting in turbulence. When secondary air is used, it is injected, at velocities of 60 to 150 ft. per sec., crosswise into the coal-carrying primary air. Turbulence reduces flame length, increases the effectiveness of the furnace and permits a smaller furnace to be used for a given output. Flame length varies with size of the fuel particles, being shortened by finer and more uniform pulverization; other factors are percentage and composition of volatile constituent, furnace temperature, and excess air. Turbulence may reduce flame length to less than 10 ft. Long flames can be given helical or U-shaped paths, the shape of the furnace being adapted to the space available.

BURNERS.—The essentials of a good burner are uniform turbulent mixture of coal and primary air, with provision so to regulate the flow of both coal and air as to give good combustion at minimum, normal and maximum loads. It should not produce flames of excessive length or velocity, nor offer excessive resistance to air and coal flow. The metal exposed to radiation should be a minimum, and the burner should be capable of easy cleaning. Furnace volume may be decreased by improving combustion conditions in the burner.

* See also pp. 4–28 to 4–37.

Factors influencing burner design and operation are the fineness, surface moisture, and volatile matter of the coal, and the size and porosity of the coke particles formed, the quantity, temperature and pressure of primary and secondary air, and the method of mixing air and coal. Furnace design and load variations also are important factors. The usual range of burner capacity is from 2 to 3 $^{1}/_{2}$ to 1; if a wider range in boiler capacity is needed, it is obtained by changing the number of burners in operation. Burners are of either stream-line or turbulent type.

To prevent flare-backs, the velocity of fuel and primary air through the piping and burner always must be greater than that of flame propagation. The tendency of turbulence is to equalize conditions across the coal-air stream, and both shorten and widen the flame. It is easier to produce initial turbulence than to maintain it further along in the furnace.

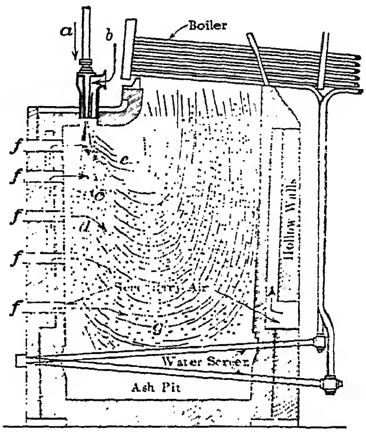


Fig. 14. Action of Stream-line Burner
a, Coal and 10 per cent air; *b*, 20 per cent air; *c*, velocity 100 to 150 ft. per sec.; *d*, velocity 10 to 15 ft. per sec.; *e*, fine coal particles; *f*, about 65 per cent air; *g*, coarse coal particles

Primary Air Ratio.—The ratio of primary air to coal should be kept as nearly constant as possible. In unit systems it varies widely with load, and adjustments are made at the burner outlet to obtain proper velocity of mixture entering the furnace. With the bin system, the primary-air-coal ratio is under control at all times. With a single burner, the ratio is set for moderate fuel-burning rates, and a lower ratio is maintained for higher loads than for lighter ones. With a multiple-burner bin-system installation, the ratio can be held constant, and the fuel burning rate varied by the number of burners in operation.

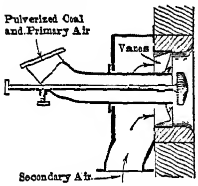


Fig. 15. Turbulent Circular Burner

With refractory furnace walls, it sometimes is necessary to increase the air-fuel ratio at higher loads in order to limit furnace temperature. The primary air fed with the coal varies with the type of burner, from a small percentage to 100% of the total air.

Stream-line Burners produce long flames with little or no turbulence. Fig. 13 shows a simple form of stream-line fan-tail burner, used in vertical firing. It discharges thin, flat streams of primary air and coal downward, as shown in Fig. 14. The streams are parallel and edgewise to the front furnace wall. Secondary air passes through the hollow refractory wall, thereby cooling it, and enters the furnace through ports in the front wall at sufficient velocity to produce turbulence when it meets the air-coal stream from the burner. The nozzles of these burners sometimes are ribbed, serrated, or rifled; deflectors at the outlet may cause jets of tertiary air to strike the primary stream. Such expedients tend to shorten the flame.

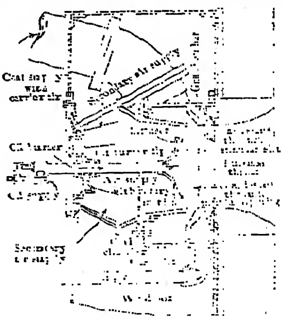


Fig. 16. Combination Burner for Coal, Oil or Gas

Turbulent Burners may have circular or straight narrow outlets. Secondary air passes through them, under either forced or induced draft, and meets the coal-laden primary air as the two enter the furnace. In comparison with a stream-line burner, the flame is short and flaring. Fig. 15 shows a simple form of circular burner, which usually discharges horizontally through refractory walls. Primary air and coal flow through the central nozzle, at the end of which is a diffuser. Secondary air flows under pressure through the annular chamber surrounding the coal stream, and which frequently contains adjustable guide vanes. Provision is made to regulate the flow of both coal and air. Fig. 16 shows a combination burner that can burn pulverized coal, gas, or oil, or all in combination.

Fig. 17 shows a turbulent intertube burner installation between the vertical tubes of water-cooled furnace walls. Secondary air jets entering under pressure through oblique ports, arranged alternately on either side of each stream, break up the vertical streams of primary air and coal. These burners are suitable for furnaces with bottoms designed for coals whose ash has high fusion temperatures.

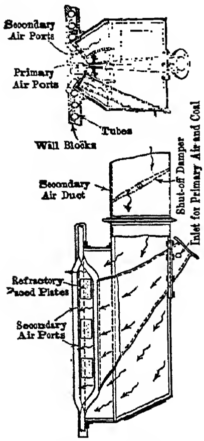


Fig. 17. Turbulent Intertube Burner

The cross-tube burner, Fig. 18, was developed for furnaces with slag-tap bottoms, in which low-fusion temperature ash must be held molten. The burner discharges primary air and coal through a horizontal slot into spaces between the vertical water-wall tubes, either horizontally or at an angle. Secondary air enters through adjustable tuyeres in the furnace wall, above and below the coal stream, and distributes the flame

FEEDERS to regulate delivery of pulverized coal into the primary air stream are required with the bin system; with the unit system feeders are unnecessary as primary air is fed through the unit pulverizer and picks up the coal when it is suitably pulverized. Feeders must be designed to prevent flooding, because of the fluid character of dry pulverized coal.

Feeders use revolving screw conveyors, belt or chain conveyors, revolving plates, star wheels, rocking gates, revolving pockets, or reciprocating pushers. Conveyor types utilize a spring-loaded regulator to prevent flooding. In one screw-conveyor type, primary air, as it passes the end of the screw under pressure of 8 to 20 in. of water, entrains the coal and carries it to the burner. Dampers control the primary air.

Feeders usually are attached to the bottom of pulverized coal bunkers with louvre gates between, and have agitators to pre- coal from packing or bridging.

FURNACE DESIGN The burning of coal in pulverized form imposes certain requirements in the design of boiler furnaces chiefly in respect to the fusion temperature of the ash and the type of furnace walls. Larger furnace volume for a given heat release rate is required than with other fuels. The report of the International Railway Fuel Assoc. (*Power*, June 19, 1928) shows heat release rates ranging

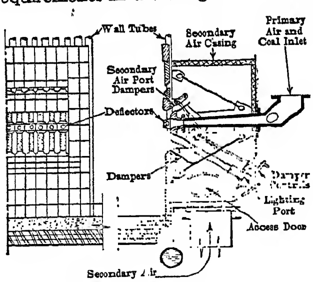


Fig. 18. Cross Tube Burner

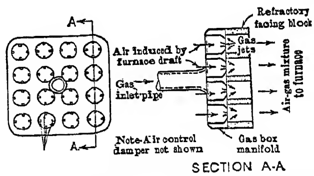


Fig. 19. Gas Burner Giving Moderate Mixing

from 15,000 B.t.u. per cu. ft. of furnace per hr. with solid refractory walls to 30,000 B.t.u. with water walls, with ash fusion temperature above 2400° F.; and from 12,500 to 22,000 B.t.u. with ash fusion temperatures of 2100–2400 B.t.u. See also p. 6–89.

Manufacturers of representative burners and feeders are: Babcock & Wilcox Co., New York; Combustion Engg. Co., New York; Riley Stoker Corp., Worcester, Mass.

5. GAS BURNERS

Gas burners used in boiler furnaces differ in the degree of mixing fuel and air that occurs in the burner. Long, luminous flames result from burners in which mixing is slight; short, non-luminous flames come from burners that mix gas with all of the combustion air.

Aspirating burners generally are used in boiler furnaces. Fig. 19 shows a type that produces a moderate amount of mixing. Gas is introduced in various ways. In Fig. 16, which shows a combination burner for gas, pulverized coal, and oil, a film of gas flowing

around the circumference of the burner throat replaces the numerous small jets of Fig. 19. Either natural or forced draft may be used.

The Venturi-type burner, with a central nozzle for gas injection is used for rapid mixing. Primary air is induced by the reduced pressure in the Venturi throat. See Fig. 20. A modified Venturi-type burner, in which mixing is done in two stages, is shown in Fig. 21.

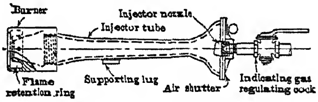


FIG. 20. Venturi-type Gas Burner

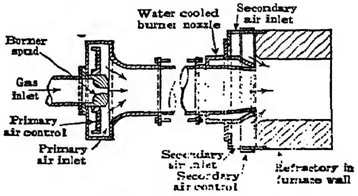


FIG. 21. Modified Venturi-type Gas Burner

FURNACE REQUIREMENTS.—Special requirements are imposed on the design of boiler furnaces by the burning of gas. See p. 6–11.

Manufacturers of representative gas burners are: Bethlehem Steel Co., Bethlehem, Pa.; Hauck Mfg. Co., Brooklyn, N. Y.; Peabody Engg. Corp., New York; Todd Combustion Equipment, Inc., New York.

6. OIL BURNERS *

The principal requisites of an oil burner are: It must completely atomize or vaporize oil; it must not clog or drool; the jet must be so shaped that it will completely mix with the air necessary for combustion; combustion must be complete and excess air at a minimum over the entire operating range; the burner must be accessible for cleaning, and require a minimum of attention.

Vaporizing burners are not used in large boiler furnaces. Two classes of atomizing burners are used: 1. Those that effect atomization by spraying, usually by steam jets, although jets of compressed air may be used. 2. Those that atomize mechanically, without any atomizing fluid.

STEAM-ATOMIZING BURNERS use the atomizing fluid to break the oil into minute particles and carry them into the furnace. These burners are either outside-mixing or inside-mixing. Steam for atomization should be at a pressure of 25 to 80 lb. per sq. in., gage. The amount of steam required for atomizing, pumping and heating the oil depends on the burner design and the method of control, ranging from 2 to 7% of the total steam generated. The temperature of the oil delivered to the burner at a pressure of 40 to 60 lb. per sq. in., gage, is 150° to 190° F. This type of burner seldom is designed to pass more than 1000 lb. of oil per hr. when using natural draft, but at least one design can burn 1500 lb. when using air at a pressure of 2 in. of water in the air register around the burner.

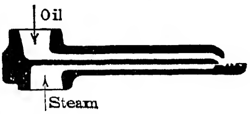


FIG. 22. Outside-mixing Burner

FIG. 23. Inside-mixing Burner

Outside-mixing Burners usually are confined to boilers operating at moderate rates, with little change in load. Fig. 22 shows a simple form giving a flat flame. Combustion air enters through checker work forming part of the furnace hearth. The combination of flame shape and method of supplying air limits the furnace to a single row of burners. Forcing the burner causes incomplete atomization, resulting in slower burning, smoking fouling of boiler surfaces, and decreased efficiency.

Inside-mixing Burners may be built to give either flat or hollow conical flames. Fig. 23 shows one of the former. In the latter the flame is short, and air readily enters it. Air is induced, through controllable registers enclosing the burner, by the furnace

* See also p. 4–49.

draft and by aspiration of the steam jet. Air pressure, at high combustion rates, is up to 2 in. of water. These burners can be set in multiple rows, providing any reasonable range of boiler operation, limited only by furnace volume.

Under certain operating conditions, steam atomizing burners may be noisy. The blow-pipe action may injure the walls of improperly constructed furnaces. Other objections are additional moisture produced in flue gases and the cost of steam for atomizing. Nevertheless, they are widely used in small plants because of their low initial cost and freedom from complication.

MECHANICAL-ATOMIZING BURNERS comprise rotary burners and spray-nozzle burners. The former is used only under low-pressure boilers.

Spray-nozzle Burners are practically the only ones used in power-plant boiler furnaces. Oil under a pressure of 50 to 300 lb. per sq. in., and at temperatures of from 100° to 350° F. issues in a hollow cone from a small orifice in the nozzle. Suitable passages in the nozzle cause a whirling motion of the oil as it is liberated. Combustion air enters, under furnace or forced draft, through a register around the burner; in some cases it is given relative motion with respect to the flame. Fig. 16 shows a typical combination burner for oil, gas and pulverized fuel.

The steam equivalent of the power required to spray the oil seldom is more than 1% of the total steam generated. Oil-burning capacity ranges from 2000 to 2500 lb. per hr. with combustion air pressures up to 6 in. of water. Several rows of burners can be installed in a furnace wall to obtain higher capacities.

Earlier methods of regulating the weight of oil delivered by spray-nozzle burners were not entirely satisfactory. Variation of oil pressure does not permit a large range of regulation, as proper atomization is not obtained with pressures below 50 lb. per sq. in. Regulation by changing burner tips is objectionable as it interrupts operation. Regulation by varying the number of burners in operation is undesirable, as it causes poor air distribution.

A more satisfactory method of regulation is to by-pass a regulated quantity of oil from the burner to supply tank or burner header. Another method provides two independent sets of oil feeds to the burner tips, using either or both in accordance with the demand for steam. These methods permit of variation of the oil burning rate, from the maximum to a minimum rate of 15 to 35% of maximum, with slight change in the spray.

Manufacturers.—A list of manufacturers of representative oil burners is given on p. 4–52.

7. BOILER FURNACE DETAILS

General

FACTORS INFLUENCING FURNACE DESIGN.—Fuel and character of load are the most important items to consider in furnace design. The kind and characteristics of the fuel, including the properties of its ash, determine the method of burning it. For instance, solid fuels may be burned on grates, on stokers or in pulverized form; the method of firing and type of burner are factors. Other items that will need to be considered in connection with the fuel are the amount of excess air, which influences boiler capacity and efficiency, and allowable carbon in fly-ash and in refuse. The load characteristics include the minimum, normal and maximum loads, and the duration of each. Heat release rates also are important, in that an increase in rate will tend to decrease the size of the boiler for a given output of steam. This, in turn, affects the material and construction of furnace walls. Maximum temperatures for a given type of wall construction also must be determined. The number of variables involved require, for the most economical arrangement and construction, that each furnace be considered as a special case.

TYPES OF FURNACE WALLS, in the decreasing order of furnace volume per unit of steam output, and in the increasing order of heat release rates and furnace temperatures, are: Solid refractory walls; hollow air-cooled refractory walls; bare water-cooled metallic walls, covered water-cooled metallic walls. The water-cooled walls are necessary for long-continued operation at high combustion rates and high temperatures. Solid refractory walls are suitable and economical for moderate rates and temperatures. For intermediate conditions, the hollow air-cooled wall or a combination of refractory and water-cooled walls may be satisfactory. Superheater or reheater surface may be substituted for some refractory or water-cooled surfaces.

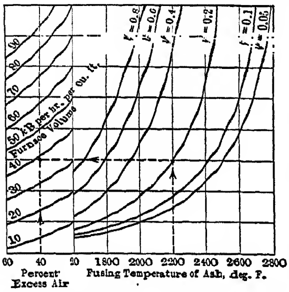
Increasing excess air, to reduce furnace temperatures and decrease wall failures, is inadvisable in ordinary operation, as it also reduces efficiency; it may be justified at peak loads. The use of preheated air usually causes higher furnace temperatures than

the use of room air. For long-continued, high-temperature operation, furnace walls should be designed with these conditions in mind.

MAXIMUM ALLOWABLE FURNACE TEMPERATURE depends on the behavior of the particular combination of fuel, ash, and material in the hot faces of the furnace walls. Depending on the composition of the ash, its fusion temperature, and the furnace-wall temperature, a refractory wall may be affected by slag penetration, chemical reaction, or erosion by molten slag running down the wall. If the temperature of a coal-fired furnace is not quite high enough to cause any of these effects on a solid refractory wall, solidified fly ash may deposit on it until the combined thickness will become so great that the temperature at the surface equals the ash fusion temperature. Variation in furnace temperature will cause the fly-ash to melt or build up until equilibrium is established. The same is true of air-cooled or water-cooled refractory walls. Metallic walls give the least difficulty from adhering fuel ash, although fused ash flowing over them will, in time, be destructive.

Furnace Volume

Furnace volume depends on the total amount of heat required in a given time and on the permissible B.t.u. release per hr. per cu. ft. of furnace volume. This heat-release rate depends on type of furnace construction, flame length, ash fusion temperature, method of firing, amount of excess air and amount of turbulence in furnace. Table 5 gives permissible heat-release rates.



ψ = fractions cold. kB = 1000 B.t.u.

FIG. 24. Relations Existing in Furnace

Since the heat-release rate in hand-fired furnaces is limited, the grate in horizontal return tubular boilers can be set within 3 or 4 ft. of the boiler surface, and within 4 to 4 1/2 ft. in horizontal water-tube boilers fired with volatile coal. With anthracite the distance can be slightly less. With stoker-fired furnaces, the distance should be made greater. See Table 3. For very high rates of combustion, even greater distances are required.

Pulverized-coal Furnace Volume usually is larger or the heat-release rates lower, than in furnaces of equal capacity burning any other fuel. In general, under identical conditions, higher heat-release rates are [illegible] all units than in large, since the rati[o] of wall surface to volume is greater. Fig. 24 (Some Factors in Furnace Design for High Capacity, E. G. Bailey, *Trans.* A.S.M.E., FSP–50–78, 1928) shows approximate relations between heat-release rates, amounts of excess air, fusing temperatures of ash, and fractions cold (ψ).

$$\psi = \frac{\text{(actual extent of cold surface in furnace)}}{\text{(maximum possible extent of cold surface in furnace)}}$$

Furnace design should consider the conditions to be met by the various elements of volume and wall surface. The use in design of average heat-release rates and average temperatures (which have been used in the above discussion) may lead to trouble because localized temperatures may be much higher than average temperatures.

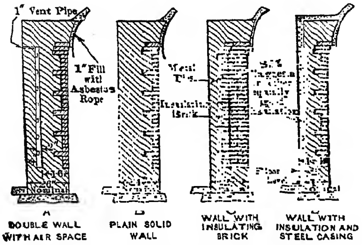
Solid Refractory Walls

Solid refractory walls are usual in the furnaces of externally fired boilers, with low heat release rates. The walls usually are integral with the boiler setting, and are built of high-grade firebrick, second grade firebrick, insulation, or some combination thereof. Some typical furnaces are shown below.

Table 5.—Average Heat-Release Rates

Method of Firing	Solid Refractory Walls		Water-cooled Metallic Walls	
	Continuous Operation	Peak Operation	Continuous Operation	Peak Operation
	B.t.u. per hr. per cu. ft. of furnace volume			
Chain- or traveling-grate stoker....	15,000	25,000	30,000	45,000
Underfeed stoker................	25,000	40,000	30,000	45,000
Pulverized coal firing.............	15,000	20,000	25,000	35,000
Oil firing........................	20,000	40,000	30,000	60,000
Gas firing.......................	20,000	40,000	30,000	60,000

HORIZONTAL RETURN TUBULAR BOILERS.—The conventional setting is shown on p. 6–13. Fig. 25 and Table 6 (Boiler Book of Hartford Steam Boiler Inspection and Insurance Co.) show approved constructions. Type B is the least costly but is more liable to air leaks than Type A. The hollow space in Type A should be filled with sand or ashes to retard air infiltration in the event of cracks in the inner wall. Type C is more costly than type B. The insulation reduces heat loss through the walls, and raises the furnace temperature. Higher grade lining, therefore, is necessary. Metal ties are advisable to bond the lining to the outer walls, as the insulating brick have little mechanical strength. Type D is the most costly setting. It is similar to Type B, excepting that 85% magnesia replaces the hollow space, and a steel casing of No. 8 gage steel plate encloses the entire setting. Heat radiation is a minimum and air infiltration is completely inhibited. The sections shown apply to the side walls at the rear of the bridge wall. In the furnace section a batter out of 6 in., from the grate level to the closing-in line near the middle of the boiler shell is recommended. Allowance for expansion should be made between the bridge wall and side walls. Relieving arches sometimes are built into the tops of furnace side walls, to permit of easy replacement of furnace walls without shoring up of the upper parts of the walls. Ample foundations are necessary to prevent settling and cracking of walls. Vertical buck stays and cross-tie rods are essential to prevent bulging of side walls. It usually is desirable to protect the blow-off pipe from flame impingement by a brick baffle. See Fig. 9, p. 6–13.



Sections apply to side walls at rear of bridge wall

FIG. 25. Typical Horizontal Return Tubular Furnace Settings

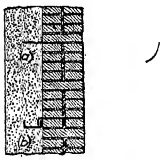


FIG. 26. Monolithic Wall Lining

Jointless Monolithic Wall Linings are made of plastic fireclay rammed into position, and tied to the outer walls: three methods are shown in Fig. 26. Space should be allowed for expansion at the abutting ends of the lining; it is desirable to fill this space with loosely-packed mineral wool, to prevent it filling with slag. Plastic fireclay also may be used to patch or reline old walls, if the old brickwork first is chipped free of slag and the openings to be filled are undercut to provide a lock for the new material. Refractory cements, best applied with a spray gun, often are used to resurface old walls.

Outer surfaces of brick settings are made impervious to air-infiltration by applying special sealing materials that remain semi-plastic even on the hot surfaces. Although such materials improve efficiency of combustion in furnaces with leaky walls, they also cause hotter furnace linings by eliminating air leakage, which has a cooling effect on the brickwork.

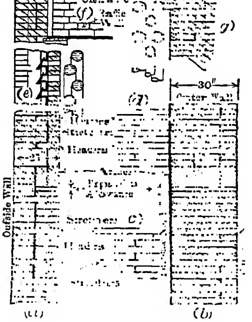


FIG. 27. Wall Construction for Horizontal Water-tube Boilers

WATER-TUBE BOILERS.—Stoker-fired furnaces may have solid refractory walls, whose arrangement depends on the type of stoker and boiler. The boiler should be so suspended from overhead beams that it cannot at any time come in contact with the furnace walls. Bridge walls and furnace linings should be high-grade firebrick. Cheaper grades of brick can be used behind the lining. Relieving arches, amply buttressed to carry thrust, may be built into the walls to relieve the load on the lower brick, in high settings, or to assist in wall repairs; expansion space should be provided below them. The upper part of high furnace walls sometimes is anchored to external steel work to prevent the wall falling inward as a result of alternate heating and cooling.

The common wall of furnaces grouped in batteries of two should be entirely of high-grade firebrick and much thicker than the side walls. Fig. 27 gives

Table 6.—Dimensions and Material Quantities of Horizontal Return Tubular Boiler Settings. (See Fig. 25)

Boiler Diam., in.	Floor to Center, in.	Length of Tube, ft.	Width of Setting		Length of Setting		Common Brick Required				Firebrick Required *	
							Overhung Front		Flush Front			
			1 Boiler	2 Boilers	Overhung Front	Flush Front	1 Boiler	2 Boilers	1 Boiler	2 Boilers	Overhung Front	Flush Front
Type A												
54	79	14	9′ 4″	16′ 0″	18′ 2″	19′ 5″	15,800	23,100	17,700	25,900	2450	2500
54	79	16	9′ 4″	16′ 0″	20′ 2″	21′ 5″	17,300	25,100	19,100	27,900	2650	2700
60	84	16	9′ 10″	17′ 0″	20′ 2″	21′ 5″	18,300	27,000	20,300	30,100	2750	2800
60	84	18	9′ 10″	17′ 0″	22′ 2″	23′ 5″	19,900	29,000	21,800	32,100	2950	2950
66	91	16	10′ 4″	18′ 0″	20′ 2″	21′ 7″	19,700	29,000	22,300	32,900	3150	3200
66	91	18	10′ 4″	18′ 0″	22′ 2″	23′ 7″	21,300	31,200	23,900	35,100	3350	3400
72	98	16	10′ 10″	19′ 0″	20′ 2″	21′ 8″	21,300	31,700	23,900	35,700	3450	3500
72	98	18	10′ 10″	19′ 0″	22′ 2″	23′ 8″	23,000	34,000	25,600	37,900	3700	3750
72	98	20	10′ 10″	19′ 0″	24′ 2″	25′ 8″	24,600	36,200	27,300	40,200	3950	4000
78	105	16	11′ 4″	20′ 0″	20′ 2″	21′ 10″	23,600	35,000	27,100	40,600	3800	3900
78	105	18	11′ 4″	20′ 0″	22′ 2″	23′ 10″	25,400	37,500	29,000	43,100	4100	4150
78	105	20	11′ 4″	20′ 0″	24′ 2″	25′ 10″	27,200	40,000	30,800	45,600	4350	4400
84	112	18	11′ 10″	21′ 0″	22′ 8″	24′ 5″	27,600	40,800	32,400	48,200	4600	4750
84	112	20	11′ 10″	21′ 0″	24′ 8″	26′ 5″	29,500	43,300	34,200	50,800	4900	5050
Type B												
54	79	14	9′ 0″	15′ 8″	18′ 0″	19′ 3″	14,900	22,100	16,700	24,800	2450	2500
54	79	16	9′ 0″	15′ 8″	20′ 0″	21′ 3″	16,300	24,000	18,100	26,700	2650	2700
60	84	16	9′ 6″	16′ 8″	20′ 0″	21′ 3″	17,400	25,800	19,300	28,800	2750	2800
60	84	18	9′ 6″	16′ 8″	22′ 0″	23′ 3″	18,800	27,800	20,700	30,800	2950	2950
66	91	16	10′ 0″	17′ 8″	20′ 0″	21′ 5″	18,700	27,800	21,100	31,600	3150	3200
66	91	18	10′ 0″	17′ 8″	22′ 0″	23′ 5″	20,300	29,900	22,700	33,700	3350	3400
72	98	16	10′ 6″	18′ 8″	20′ 0″	21′ 6″	20,400	30,600	22,900	34,400	3450	3500
72	98	18	10′ 6″	18′ 8″	22′ 0″	23′ 6″	22,000	32,800	24,500	36,600	3700	3750
72	98	20	10′ 6″	18′ 8″	24′ 0″	25′ 6″	23,700	35,000	26,100	38,900	3950	4000
78	105	16	11′ 0″	19′ 8″	20′ 0″	21′ 8″	22,500	33,700	25,900	39,100	3800	3900
78	105	18	11′ 0″	19′ 8″	22′ 0″	23′ 8″	24,300	36,100	27,700	41,600	4100	4150
78	105	20	11′ 0″	19′ 8″	24′ 0″	25′ 8″	26,100	38,500	29,500	44,000	4350	4400
84	112	18	11′ 6″	20′ 8″	22′ 6″	24′ 3″	26,400	39,300	30,900	46,500	4600	4750
84	112	20	11′ 6″	20′ 8″	24′ 6″	26′ 3″	28,300	41,800	32,800	49,000	4900	5050
Type C												
54	79	14	8′ 8″	15′ 4″	17′ 10″	19′ 1″	10,500	17,000	12,100	19,500	3350	3450
54	79	16	8′ 8″	15′ 4″	19′ 10″	21′ 1″	11,500	18,400	13,000	20,900	3650	3750
60	84	16	9′ 2″	16′ 4″	19′ 10″	21′ 1″	12,100	19,600	13,800	22,300	3900	4000
60	84	18	9′ 2″	16′ 4″	21′ 10″	23′ 1″	13,100	21,100	14,800	23,800	4250	4300
66	91	16	9′ 8″	17′ 4″	19′ 10″	21′ 3″	13,100	21,200	15,200	24,700	4350	4450
66	91	18	9′ 8″	17′ 4″	21′ 10″	23′ 3″	14,100	22,800	16,300	26,300	4700	4800
72	98	16	10′ 2″	18′ 4″	19′ 10″	21′ 4″	14,100	23,100	16,300	26,700	4900	5000
72	98	18	10′ 2″	18′ 4″	21′ 10″	23′ 4″	15,100	24,800	17,400	28,400	5300	5400
72	98	20	10′ 2″	18′ 4″	23′ 10″	25′ 4″	16,200	26,400	18,400	30,000	5700	5750
78	105	16	10′ 8″	19′ 4″	19′ 10″	21′ 6″	15,600	25,400	18,700	30,000	5350	5450
78	105	18	10′ 8″	19′ 4″	21′ 10″	23′ 6″	16,700	27,300	19,900	32,400	5800	5800
78	105	20	10′ 8″	19′ 4″	23′ 10″	25′ 6″	17,900	29,100	21,000	34,300	6200	6250
84	112	18	11′ 2″	20′ 4″	22′ 4″	24′ 1″	18,300	29,900	22,300	36,600	6450	6700
84	112	20	11′ 2″	20′ 4″	24′ 4″	26′ 1″	19,500	31,700	23,500	38,400	6850	7100
Type D												
54	79	14	8′ 0″	14′ 8″	17′ 6″	18′ 9″	9,600	16,000	10,900	18,200	2450	2500
54	79	16	8′ 0″	14′ 8″	19′ 6″	20′ 9″	10,500	17,400	11,800	19,600	2650	2700
60	84	16	8′ 6″	15′ 8″	19′ 6″	20′ 9″	11,100	18,500	12,600	21,100	2750	2800
60	84	18	8′ 6″	15′ 8″	21′ 6″	22′ 9″	12,000	19,900	13,500	22,600	2950	2950
66	91	16	9′ 0″	16′ 8″	19′ 6″	20′ 11″	12,000	20,100	13,900	23,300	3150	3200
66	91	18	9′ 0″	16′ 8″	21′ 6″	22′ 11″	13,000	21,600	14,800	24,800	3350	3400
72	98	16	9′ 6″	17′ 8″	19′ 6″	21′ 0″	13,200	22,200	15,100	25,400	3450	3500
72	98	18	9′ 6″	17′ 8″	21′ 6″	23′ 0″	14,200	23,800	16,100	27,000	3700	3750
72	98	20	9′ 6″	17′ 8″	23′ 6″	25′ 0″	15,300	25,500	17,100	28,700	3950	4000
78	105	16	10′ 0″	18′ 8″	19′ 6″	21′ 2″	14,600	24,400	17,200	29,100	3800	3900
78	105	18	10′ 0″	18′ 8″	21′ 6″	23′ 2″	15,700	26,200	18,400	30,900	4100	4150
78	105	20	10′ 0″	18′ 8″	23′ 6″	25′ 2″	16,900	28,000	19,500	32,700	4350	4400
84	112	18	10′ 6″	19′ 8″	22′ 0″	23′ 9″	17,100	28,600	20,600	34,700	4600	4750
84	112	20	10′ 6″	19′ 8″	24′ 0″	25′ 9″	18,300	30,300	21,800	36,500	4900	5050

* For one boiler; for two boilers double the number required for one boiler.

typical sections through one type of wall construction for horizontal water-tube boilers. To avoid overheating, steel work supporting the boiler should not be enclosed in the brickwork.

In tall settings, walls often are built concave toward the fire, and the upper parts are so built that when cold they lean outward as much as 5 deg. from the vertical. This prevents their bulging or leaning inward when heated.

Clinker Belts, *i.e.*, the lower parts of side walls of the furnace, adjacent to and just above the fuel bed, usually require special construction, as they are subject both to intense heat and to adhesion of clinker if the wall is of ordinary firebrick. Clinker not only cuts down effective grate surface, but allows excessive air to flow through as a result of their scraping the sides of the moving fuel bed. Removal of clinker from a solid refractory wall by a slice-bar destroys the wall. Several solutions are available to meet these severe conditions. The wall at this point may be built of special slag-resisting blocks which may be solid, or hollow (see Fig. 11), or air- or water-cooled metallic wall sections may be used. (See Figs. 10 and 12.)

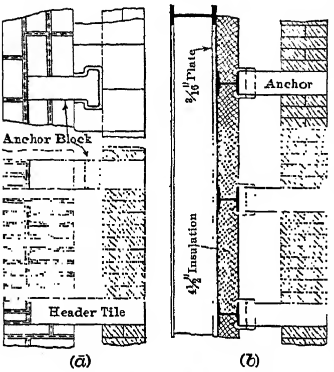


FIG. 28. Self-supporting Air-cooled Refractory Wall

Joints in Refractory Walls are vulnerable points for slag attack. The brick should be laid with a fireclay mortar with refractory properties equal to those of the brick itself. Finely-ground raw fireclay and as much finely-ground calcined fireclay or ground firebrick, free from slag, as will stay in suspension in a batter should be used in laying the brick. Such material serves as a filler and is adequate; bonding mortar seldom is required. Joints should be as thin as possible. If accurately sized brick are not available, each course of brick should be rubbed level with a silicon-carbide block before laying the next course. A thin coat of filler, of the consistency of batter, should be poured over the leveled course, and each brick, after being dipped in the batter, should be slid and tapped into place.

Backing-up brick is bonded to the inner lining by header and stretcher courses. If the lining is only 4 1/2 in. thick, every fourth or fifth course should be a stretcher course as *d* in Fig. 27. A 9-in. wall can be laid as header courses with every fourth or fifth course a stringer course, *i.e.*, a header course behind a stretcher course (*a*, Fig. 27).

Air-cooled Refractory Walls and Arches

Air-cooled refractory walls are either entirely self-supporting, or sectionally self-supporting. Cooling air flows through ducts in the walls and into the furnace.

SELF-SUPPORTING AIR-COOLED REFRACTORY WALLS may be built entirely of standard size brick, or with special refractory tile in the furnace lining, bonded to the outer wall. Fig. 28 shows two forms of this type of wall. In each, the inner wall is flexibly bonded to the outer wall, to provide for differences in expansion. Sometimes larger blocks are used, instead of the standard brick, to reduce the number of inner wall joints.

Special forms of air-cooled blocks sometimes are used in the clinker belt, as shown in

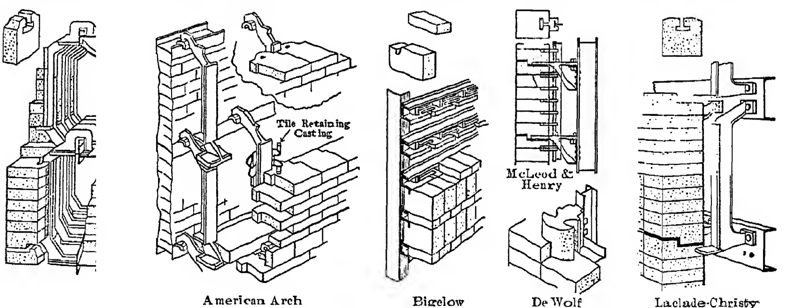


FIG. 29. Sectional Self-supporting Air-cooled Refractory Walls

Fig. 11, and sometimes for lining the entire wall. Some blocks have openings that permit flow of air through the block and into the furnace in order to cool the surface next to the fire, thereby reducing adherence of clinker.

Self-supported hollow walls cannot be used in extremely high furnaces because of the inability of the lower part of the walls, when hot, to carry the load of the upper part.

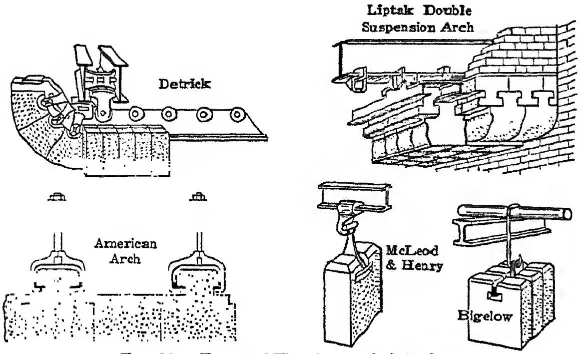


Fig. 30. Types of Flat Suspended Arches

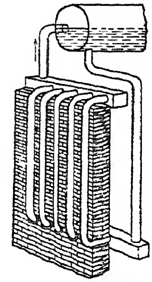


Fig. 31. Typical Arrangement of Bare-tube Furnace Wall

SECTIONALLY-SUPPORTED AIR-COOLED REFRACTORY WALLS usually are built in horizontal belts 2 to 3 ft. high, attached to an outside steel structure. Static load on the brickwork thus is reduced and a means provided to support the wall when refractory replacements are made. Fig. 29 shows typical forms. Different makes vary in shape of the refractories, number of special shapes, brackets and type of supporting steel, methods of providing for expansion and for sealing joints.

ARCHES over the fuel beds are seldom curved or sprung. Flat suspended arches (see Fig. 30) are more desirable. They require less skill in erection, exert no end thrusts, do not distort when heated, and if necessary can be repaired while the furnace is in operation. The refractory tile are air-cooled on the back side, flexibly supported, can expand or contract freely, and have no additional weight to support. One make, not shown, incorporates a veneer of silicon carbide, enabling the arch to withstand very high temperatures, rapid temperature changes and slagging action.

Properties desired in a good refractory are relative infusibility, relatively low thermal conductivity, flexibility of structure, low thermal expansion, impermeability toward gases and liquids, chemical inertness, and resistance to abrasion. See Refractories in Vol. 3 of this series. Failure of a refractory in a boiler furnace may be due to one or more of the following: Fusion; subsidence under load; spalling; slag action; changes in dimension.

Water-cooled Furnace Wall

WATER-COOLED METAL FURNACE WALLS are of three types: Bare-plate, bare-tube and covered-tube walls. They are more costly than refractory walls, but can withstand more severe conditions. In general, they are used only in locations where the refractory wall would deteriorate rapidly, for instance in the bridge wall of underfeed stoker furnaces, or that part of the side wall immediately adjacent to the fuel bed of traveling-grate stokers operating at moderate rates. If higher rates of combustion are maintained with either type of stoker, the entire wall surface and arches may require water cooling.

Bare-plate Wall furnaces are those in all internally fired boilers, as Scotch marine boilers, locomotive boilers, etc.

Bare-tube Walls are connected into the boiler circulation system, as in Fig. 31. They may be constructed of plain tubes (Fig. 32) or fin tubes (Fig. 33). In some arrangements, the water walls form a separate boiler that discharges its steam into an upper drum of the main boiler. The plain tube walls usually comprise tubes fairly closely spaced, connected to external headers and backed with firebrick walls. Other arrangements stagger the tubes in two rows or use special bifurcated tubes; the space between tubes then is but 1/4 in., forming a practically continuous water wall. In Fig. 32*a*, the back sides of the tubes receive heat by radiation from the firebrick backing, if the space between tubes does

not fill with sintered fly ash or molten slag. Arrangements b, c, d, Fig. 32 often are used in gas- or oil-fired furnaces. The tube is embedded to half its depth in plastic refractory or tile. The enclosure of the furnace wall in sheet metal and insulation permits reduction of the refractory to 9 in. thickness. In Fig. 32e, machined cast-iron blocks, clamped to the tube by welded studs, protect the refractory backing and increase heat transmission to the tubes.

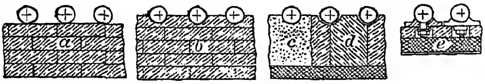


Fig. 32. Arrangement of Wall Behind Tubes

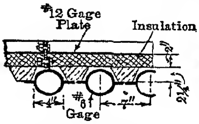


Fig. 33. Fin-tube Water Wall

Fig. 34. Drake Integral Block Water Wall

In the fin-tube construction, Fig. 33, the longitudinal fins are welded to each tube at opposite ends of a diameter. If the fins are too wide, the edges will burn back. If they are too long, they will warp and tear away from the tubes. Insulation behind the tubes, when used instead of refractory, is supported on lugs welded to the tubes.

COVERED-TUBE WALLS usually consist of tubes protected either by integral blocks or attached blocks; the latter may be all metal, all refractory, or metal coated with refractory. The blocks generally are rectangular, with flat faces, and form a substantially continuous flat surface when placed close together.

Integral Block construction is obtained by casting iron blocks on boiler tubes. See Fig. 34. Thermal contact between block and tube is good. Space must be left between blocks to permit growth of the cast iron. A close approximation to the good thermal contact of this construction is obtained by shrinking internally-machined cast-iron blocks on accurately-sized tubes. See Fig. 35.

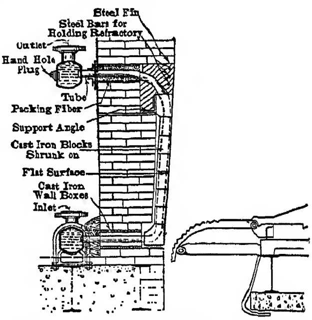
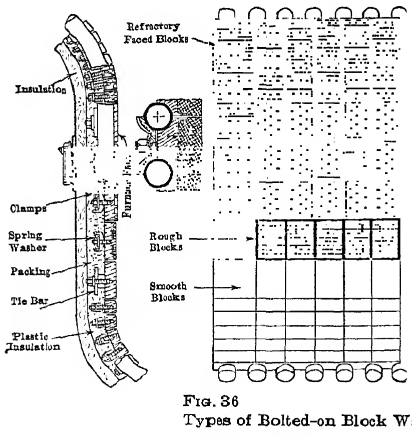
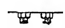


Fig. 35. Foster-Wheeler Shrunk Block Water Wall

Attached Block construction comprises metallic blocks bolted to the water-wall tubes. Various types are shown in Figs. 36, 37 and 38. With such construction, furnace temperatures are higher than with bare-tube walls under identical conditions, because of the lower heat transmission of the block-tube walls. This may be important at light loads.

Either cast iron or steel may be used for the blocks shown in Fig. 36. Depending on





. 37

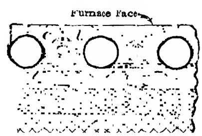


Fig. 36 . 38

Types of Bolted-on Block Water Walls

furnace conditions, the face exposed to the fire may be bare or coated with refractory; the bare face may be plain or ribbed. Refractory-faced blocks are used where high-tempera-

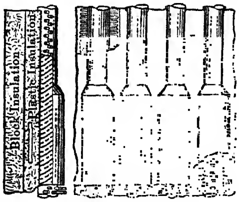


FIG. 39

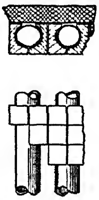


FIG. 40

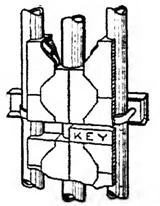


FIG. 41

Types of Refractory-protected Water-tube Walls

ture walls are necessary to assist combustion, and bare blocks where cooling surface is desirable. The blocks span the space between the tubes to which they are attached, and make good thermal contact by reason of ground joints and a plastic filler.

In Fig. 37, embossed sheets of a special alloy, which decreases the thermal resistance, are interposed between the cast-iron blocks and the tubes. In Fig. 38, the bare or refractory-faced blocks and tubes are brought in close contact by channels and toggle joints.

Refractory-protected Water Tubes are shown in Figs. 39, 40, and 41. Walls of this type usually transmit heat less rapidly than do walls of all-metal blocks. Refractory-protected water-tube walls materially assist in maintaining high furnace temperatures at low fuel-burning rates. In the stud-wall construction, Fig. 39, short iron studs are welded on the tube surface where plastic refractory is to be installed. The entire wall-tube surface in the hot parts of a furnace can be completely covered with a thickness of plastic refractory that will give the desired rate of heat absorption, while tubes in the cooler parts of the wall can be bare except for the refractory-covered studs between tubes. The studs support the refractory and cool it by providing a good heat conductor to the water in the tubes. In Fig. 40, small fireclay blocks are slipped around ordinary boiler tubes. The rate of heat transfer to the tubes can be increased by using silicon-carbide blocks. Such walls may be backed with refractories, block insulation, or a combination of both. The outer surface of the wall should be coated with a sealing cement or steel casing to prevent infiltration of air. In Fig. 41, interlocking fireclay, silicon carbide or cast-iron blocks maintain intimate contact with the tubes without the use of clamping devices. Horizontal structural-steel channels so support belts of the blocks that by the removal of key blocks, any block can be removed without disturbing any of those above it.

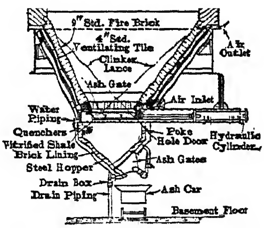


FIG. 42. Ash Hopper for Stoker-fired Furance

Furnace Bottoms

The type of furnace bottom used depends on the fuel, characteristics and methods of removal of the ash, method of firing, initial cost and maintenance costs. Hand-fired or stoker-fired furnaces, operated at moderate rates, usually have ashpits, cleaned by hand. See Fig. 4.

Stoker-fired furnaces, operating at higher rates, have ash hoppers of large capacity, The steel hopper is lined with second-grade, hard-burned firebrick, paving brick, or cast-iron air-cooled plates. The hot ashes usually are quenched by water sprays. In some installations, the ashes are carried away by hydraulic sluices. See Section on Material Handling, Vol. 3 of this series.

Oil- or gas-fired furnaces have solid bottoms, or bottoms with air-cooled passages. Air-cooled bottoms may be of refractory hollow tile, or of several layers of flat interlocking tile carried on standard brick on edge, but not in contact with each other.

Pulverized-coal-fired furnaces have either dry or wet bottoms.

Dry Bottoms are the more common. In them, the ash deposited on the bottom is solidified. To prevent the deposited ash from forming into large clinkers, the design must incorporate water screens of 4-in. tubes on 14-in. centers, above the furnace floor, air-cooled furnace bottoms, combinations of these two, or water-cooled bottoms.

Wet Bottoms, used in slag-tap or slagging boiler furnaces, form a hearth in which the molten ash collects in a pool. It remains molten and is tapped off either continuously or periodically, similar to the tapping of a foundry cupola. The molten ash, as it flows out, is granulated by a high-velocity water jet driving it against a plate, or by falling through a spray of multiple water jets. Wet-bottom furnaces originally were built to handle ash of fusion temperatures of 1900 to 2000° F., but inasmuch as the flat incandescent furnace bottom aids combustion, an additional advantage accrues from the saving in space requirements. As a result, furnaces have been developed to burn coals with ash fusion temperatures as high as 2600° F. In these furnaces, flames from the burners must bathe the hearth. Operation at high combustion rates only may be necessary, as the ash may solidify at low rates. Fluidity of the ash can be increased by adding limestone or other flux. See P. Nicholls and W. T. Reid, Fluxing of Ashes and Slags as Related to the Slagging Type of Furnace, *Trans.* A.S.M.E., RP-54-9, 1932.

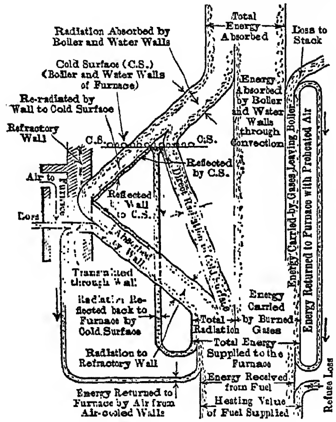


Fig. 43. Heat-flow Diagram of Boiler Furnaces

Preheated air can be used to full advantage to aid combustion of pulverized coal in wet-bottom furnaces without the troubles of ash-removal from the furnace bottom that occur in dry-bottom furnaces.

A type of all-refractory wet bottom, usually installed in furnaces with water-cooled walls, is built on steel plates carried on an air-cooled structure of piers and I-beams. Three or four courses of $2\,{}^1/_2$-in. firebrick are laid on the plates, and covered with $7\,{}^1/_2$ to 9 in. of burned dolomite or plastic chrome refractory. Extra courses of firebrick, laid near the furnace walls, form a saucer-shaped bottom. The tap-hole at the side of the furnace, is plugged by a ball of fireclay.

Furnaces with all-refractory wet bottoms are fairly satisfactory only when used with coals of low ash-fusion temperatures, at uniform, high combustion rates. Whenever the bottom cools, cracks may develop which will fill with slag. With frequent cooling the size of the bottom continually increases, ruining the seal at the furnace walls and displacing the water-cooled side walls. Iron sulphide, formed from iron pyrites in the coal, has a particularly bad erosive effect on the refractory, especially in cracks. The amount of iron sulphide formed can be reduced by pulverizing the coal until 80 to 90% passes through a 200-mesh sieve, as compared with the usual 65 to 70%.

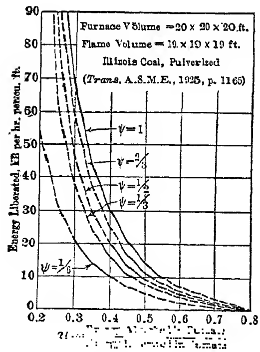


Fig. 44. Relation of ψ and Energy Release Rate (kB = 1000 B.t.u.)

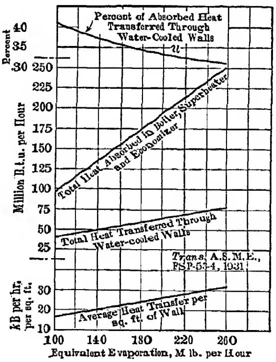


Fig. 45. Typical Heat Transfer Relations in Pulverized Coal Furnace

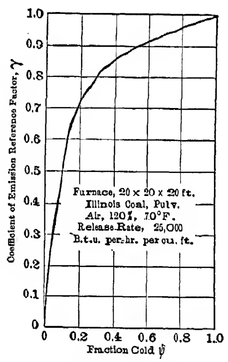


Fig. 46. Typical Relations of ψ and γ

The most satisfactory wet bottoms up to 1936 have been built of smooth cast-iron blocks (Fig. 36) on water tubes connected into the boiler circulation system. R. Shellenberger describes (Furnace Bottoms for Tapping Ash in the Molten States, *Mech. Engg.*, Jan., 1936) a construction that provides horizontal buck stays capable of exerting a pressure of several thousand pounds per foot of wall between the furnace bottom and the water-cooled side walls to insure tightness against leaks. Bottom-supported walls thus may be held constantly in contact with the bottom and can expand and contract on rollers as the unit is put into and out of service. The furnace may be designed for either periodical or continuous tapping of the slag. Tapping-spout maintenance is reduced by water cooling. Continuous tapping requires a fairly large tap hole, centrally located in the furnace bottom, kept free and open by withdrawing through it some of the hot furnace gases.

8. ENERGY INPUT AND OUTPUT OF A BOILER FURNACE

The heat energy input and output of a boiler furnace having a combination of air-cooled and water-cooled walls may be shown by a heat-flow diagram. See Fig. 43. Energy supplied to the furnace is distributed to the boiler, to the exit flue gases, and to the furnace walls. The latter is by radiation from solid masses of fuel and slag, by radiation from gases and fly-ash, and by convection. The temperature of the furnace-face of the wall at any point depends on the temperatures, relative area and location of the other surfaces in the furnace, as part of the heat received by any part of the furnace wall is radiated and reflected to colder surfaces " seen " by it.

FRACTION COLD. All water-cooled surfaces in a furnace, *i.e.*, walls and slag screens plus that part of the boiler surface receiving radiation, is considered collectively as *Cold Surface.* The ratio (actual cold surface ÷ maximum possible cold surface) is defined as the *Fraction Cold* (ψ). The value of ψ has an important bearing on furnace temperature. Cold surface can be used instead of refractory surface in all or part of the furnace walls to limit furnace temperature, at maximum combustion rates, to a value at which fly-ash will not damage the refractories. If the substitution is carried too far, furnace temperature at low combustion rates may be too low for good combustion. The proportion of energy released in a boiler furnace varies with the value ψ as shown, in a general way, in Fig. 44.

Fig. 45 shows, for a particular pulverized-coal-fired furnace, with $\psi = 1$, the effects of load variation on total energy release, on the relative amount of energy absorbed by water walls, and on the rate of heat transfer per unit of surface.

In a proposed steam-generating unit, the total heat release rate necessary in the furnace to carry the load may be calculated, and an apparently satisfactory specific heat-release rate selected for the desired furnace wall construction and operating conditions. From these figures, the tentative dimensions of the furnace can be determined. These are modified to give the final design, after investigation of furnace and wall temperatures, and heat transmission rates to and from the walls.

In calculating probable heat transfer to the furnace walls, only that transferred by radiation need be considered, as the total transfer by conduction and by convection is approximately only 6 or 7%. The rate at which the walls will receive heat by radiation can be computed by the method given in Heat Power Engineering, by Barnard, Ellenwood, and Hirshfeld (John Wiley & Sons), Part II, chap. xxvi.

For combined refractory and water-cooled furnace walls, the proportion of total heat released in the furnace that is transmitted to the walls is intermediate between that transmitted to all-refactory walls or all-water-cooled walls. Heat is interchanged between the refractory and the water-cooled surfaces. The analytical methods of calculation are complicated, but Wohlenberg and Lindseth (The Influence of Radiation in Coal-Fired Furnaces on Boiler-Surface Requirements, and a Simplified Method for Its Calculation, *Trans.* A.S.M.E. xlviii, p. 849, 1926) have presented a simpler method, using charts which show the influence of the fraction cold. The fraction u_c of furnace energy liberated, that would be absorbed by the walls of a completely water-cooled furnace ($\psi = 1$) equal in size to the furnace being analyzed, first is determined. This may be estimated from a chart similar to Fig. 44. Then u = the fraction absorbed by the water walls of the actual furnace being considered, $= u_c \times \gamma$, where γ = a suitable factor based on the actual fraction cold. γ is the coefficient of emission reference factor, and is defined as

$$\gamma = \frac{\text{energy absorbed by water walls in given furnace cavity}}{\text{energy absorbed in same cavity with } \psi = 1}$$

Typical relations between γ and ψ are given in Fig. 46.

For any set of conditions, a certain value of ψ will require, for a given final gas temperature, the minimum total surface in furnace and boiler, which value is the most effective fraction cold for the stated conditions.

Heat Transmission Through Furnace Walls

The rate of heat transmission through furnace walls depends on the character of the materials in the wall, the location of the various elements of the wall relative to each other, and the method of cooling.

As the heat is transmitted by conduction, rate of transmission depends on the thermal conductivity of the materials used, and varies inversely with thickness. Many furnaces are built with insulation behind a 9-in. firebrick lining, instead of with the former 18- to 22 1/2-in. firebrick walls. The effect of insulation in the wall is to flatten the temperature gradient through the firebrick lining. See Fig. 47. This may be beneficial to the firebrick, but excessive thickness of insulation together with high furnace temperatures may cause softening or incipient fusion of the lining, and promote slag action. Table 7 and Fig. 48 show heat losses through various types of solid walls.

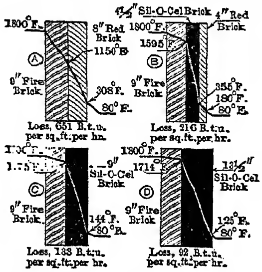


Fig. 47. Variation of Temperature Gradient and Heat Loss in Furnace Walls

So-called dead-air spaces should not be used between inner and outer walls of furnaces. A considerable amount of heat would be radiated across them, and the air would carry heat from one wall to the other by convection. If double-wall construction must be used, the space should be filled with granular material as sand or ashes.

AIR-COOLED WALLS.—Forced cooling of refractory walls by passing air through ducts in the walls is a complex process of heat transmission. Figs. 49 and 50 show thermal relations existing in two typical installations. The decrease in rate of heat transmission along the duct has little significance, and only one average rate for the entire length usually is considered. For the installation represented by Fig. 49, average heat transmission is about 2600 B.t.u. per hr. per sq. ft. of wall surface; the overall coefficient of heat transfer from the hot surface of the lining to the air is 1.01 B.t.u. per hr. per sq. ft. per deg. F. temperature difference. This value of overall coefficient is representative for all the common examples of air cooling through fireclay brick linings. In the installation represented by Fig. 53, the average overall coefficient was 6.3.

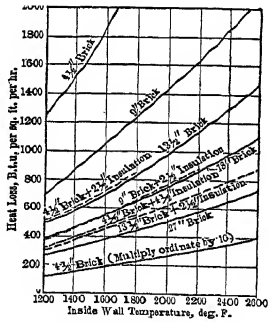


Fig. 48. Heat Loss through Furnace Walls, with Atmospheric Pressure in Furnace

WATER-COOLED WALLS.—The difference in temperature between the hot surface and the water in the tubes depends on the temperature of the furnace gases and surrounding bodies, the heat absorbing and transmitting power of the various elements making up the wall, and the temperature of the inner surface. This last depends on the rate at which the water in the tube

Table 7.—Heat Losses Through Solid Furnace Walls*

Room Temperature = 80° F.

Thickness, in.			Hot-surface Temperatures, deg. F.						
Firebrick	Insulation †	Red Brick	1000	1200	1400	1600	1800	2000	2500
			B.t.u. per hr. per sq. ft.						
13 1/2	0	8	240	305	375	445	525	605	835
9	0	8	295	370	455	545	645	745	1015
9	2 1/2 *A*	0	170	217	264	312	365	417	...
9	4 1/2 *A*	0	110	138	169	203	237	...	...
9	4 1/2 *A*	8	95	118	143	170	198	228	310
9	4 1/2 *B*	0	215	267	325	383	450	515	700
9	4 1/2 *B* + 4 1/2 *A*	0	....	....	....	150	175	200	275
9	4 1/2 *C*	0	....	....	....	390	455	520	700
9	4 1/2 *C* + 4 1/2 *A*	0	....	....	....	....	176	203	273
4 1/2	4 1/2 *A*	0	122	153	187	220	255	...	...
0	4 1/2 *C* + 4 1/2 *A*	0	96	120	147	175	205	235	...

* Taken by permission from curves prepared by the Johns-Manville Corp., 1931.

† Insulation *A* is natural Sil-O-Cel, for temperatures up to 1600 deg. F. Insulation *B* is Sil-O-Cel C-22 for temperatures up to 2000 deg. F. Insulation *C* is Sil-O-Cel "Super," for temperatures up to 2500 deg. F.

removes the heat transmitted through the wall. Temperatures must not be permitted to rise to a point that will cause failure of any portion of the wall.

The temperature gradients through wall-protecting blocks, as shown in Fig. 36, and the corresponding heat-transmission rates encountered in actual service are shown in Fig. 51. Fig. 52 shows probable relative heat-absorption rates of several kinds of water-walls, of another make. The safe rate of heat absorption by bare water-wall tubes (4 in. O.D., No. 8 gage) is 85,000 B.t.u. per hr. per sq. ft. of projected area exposed to radiation.

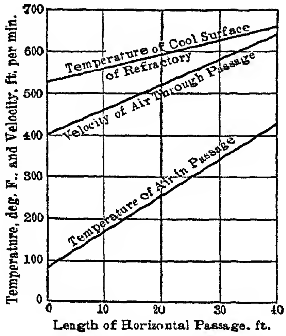
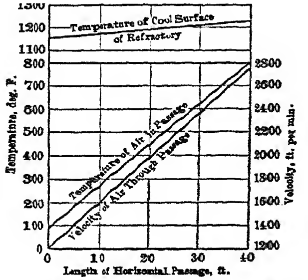


Fig. 49. Thermal Relations in a 9-in. Fireclay Air-cooled Refractory Wall. Initial Air Velocity, 400 ft. per min.



Initial air temp., 90° F.; temp. of hot surface of refractory, 2800° F.; passage 9 in. wide, 24 in. high.

Fig. 50. Thermal Relations in a 9-in. Silicon Carbon Air-cooled Refractory Wall. Initial Air Velocity, 1200 ft. per min.

The heat transmission rate through the stud-tube water walls (Fig. 39) is approximately equal to that of construction BAG, Fig. 51, if the length of stud on the front face of the tube is 1 in. and if the refractory is flush with the end of the stud, which is standard construction. For partially-studded walls, with studs and refractory between 3 1/2-in. tubes on 6 in. centers, the resulting surface is about 50% bare tube and 50% covered tube. For the refractory-covered portion between the tubes of such walls, the heat transmission rate is about 65% greater than the mean value for the full stud wall.

The heat-absorption rate of fin-tube water-cooled walls, Fig. 33, probably is no greater than that of ordinary tubes backed by refractory walls, if both walls are clean. In the latter case the refractory walls radiate and reflect heat to the back sides of the tubes. The fin-tube wall has the advantage in actual service, as the refractory wall between and behind the standard tubes usually is coated with slag and ash, which may build out over a portion of the front side of the tubes, partially shielding them from radiant heat.

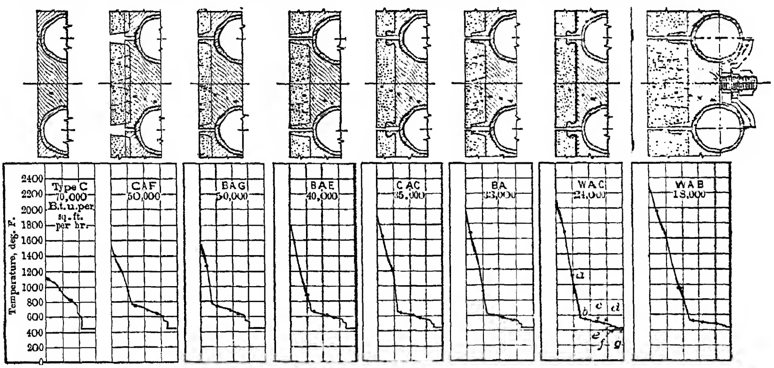


Fig. 51. Heat Drop through Water-cooled Furnace Walls

a–b Drop through Refractory
b–c Drop through Cast Iron
c–d Drop through Heat-conducting Bond
d–e Drop through Tube
e–f Drop through Steam Film
f–g Saturated Steam Temperature

Influence of Different Variables

Many individuals have investigated the influence of various factors on design and operation of boiler furnaces. For a comprehensive bibliography of 60 articles, see An Experimental Investigation of Heat Absorption in Boiler Furnaces by Wohlenberg, Mullikin, Armacost, and Gordon, *Trans.* A.S.M.E., RP-57-4, 1935.

EFFECT OF FRACTION COLD AND HEAT-RELEASE RATE.—In Some Fundamental Considerations in the Design of Boiler Furnaces, by Wohlenberg and Brooks (*Trans.* A.S.M.E., FSP-50-39, 1928) the analyses assume the use of Illinois bituminous coal of 12,800 B.t.u. per lb. heating value, and ash fusion temperature, 2000° F., and of a certain specified ultimate analysis. The outstanding features of the results are shown in Figs. 53 to 58, in which fraction cold is represented by ψ, heat-release rate in B.t.u. per hr. per cu. ft. by R, radiation mean temperature of the gases in the furnace by t_U, mean temperature of the refractory walls by t_R, and the ratio of heat added by the furnace walls to the heat released in the furnace by u. Figs. 55 and 56 show that the permissible rate of heat release R may be increased as the fraction cold is increased.

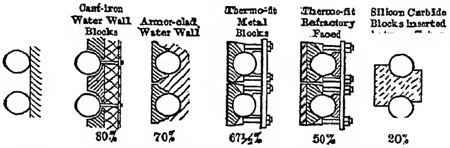


FIG. 52. Heat Absorption Rates of Water-cooled Furnace Walls

For powdered-coal firing, Fig. 60 shows that with a given flame temperature, t_U, higher heat release rates R, or smaller fraction cold, ψ, can be adopted in furnace design if excess air is increased, but with a decrease in efficiency. In operating a given unit, an increase of excess air permits heat release rates to be increased without exceeding limiting gas temperature. Wall temperatures indicated in Figs. 53–54 are mean temperatures that may be expected. Actually, local hot spots may be 300 to 400° F. higher. These locations must have greater cooling and fraction cold. Otherwise, the materials would attain temperatures above that at which they can survive.

Figs. 53 and 54 are approximately correct for a furnace volume of 27,000 cu. ft. when using West Virginia coal with a heating value of 14,500 B.t.u. per lb. and ash fusion temperature of 2500° F., except that the values of u would be a little higher than those of the curves, which are for a volume of 8000 cu. ft. Other results show that if the air temperature is 500° F., t_U and t_R increase not over 150° F., and u is raised about 0.08.

EFFECT OF CHARACTER OF COAL.—Wohlenberg and Anthony (Influence of Coal Type on Radiation in Boiler Furnaces, *Trans.* A.S.M.E., FSP-51-36, 1929) found

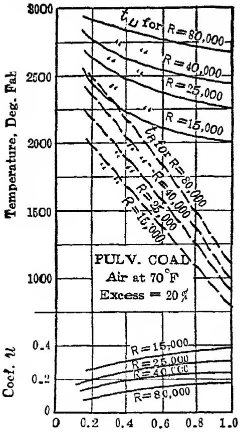


FIG. 53

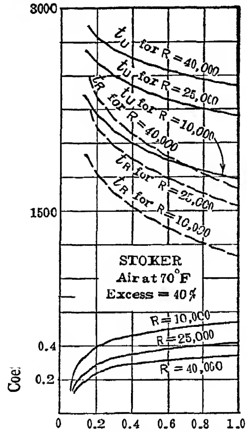


FIG. 54

Effect of Fraction Cold ψ and Heat Release Rate R on Furnace Conditions

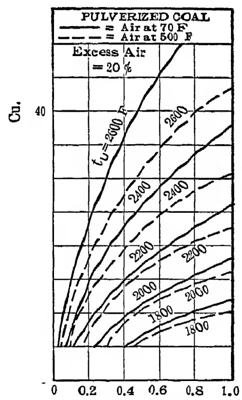


FIG. 55. Effect of ψ and R on Flame Temperature

t_U = Radiation mean temperature of gases in furnace, deg. F. t_R = Mean temperature of inner surface of refractory, deg. F. u = Energy absorbed by furnace walls ÷ energy released

the heating value of the coal to be the most significant index of the influence of type of coal on flame temperature. The investigation covered furnaces having various values of fraction cold and heat release rates. Based on the assumption that different kinds of coal were burned in pulverized form with 20% excess air at 500° F.,

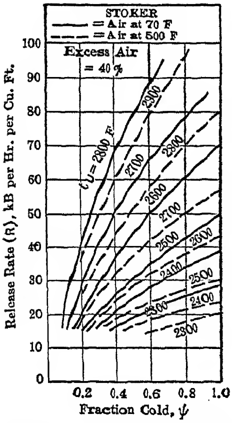


FIG. 56. Effect of Fraction Cold ψ and Heat Release Rate on Furnace Flame Temperature

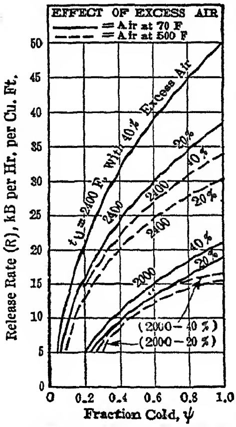


FIG. 57. Effect of Excess Air, Powdered Coal Firing t_U = furnace gas temperature, deg. F.

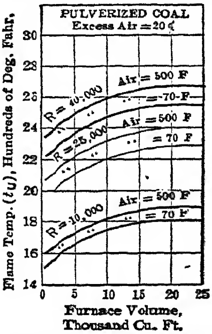


FIG. 58. Effect of Increased Furnace Volume (ψ = 1)

it was indicated that a heat absorption rate of about 160,000 B.t.u. per hr. per sq. ft. of projected area of bare iron water-cooled surfaces changes about 6000 B.t.u. with each 1000 B.t.u. per lb. change in heating value of the coal, heat release rate being maintained constant; when the heat-absorption rate is about 20,000 B.t.u. its change is negligible with variations in heating value of the coal. Further conclusions were that, with coal of a heating-value of 10,000 B.t.u. per lb. and flame temperature of about 3000° F., each 1000 B.t.u. per lb. change in heating value of the coal changes flame temperature about 50° F., heat release rate remaining unchanged; when flame temperature is around 2500° F., the change is about 25° F. The rate of change in flame temperature with variations in heating value decreases from the above figures with increases in heating value of coal above 10,000 B.t.u. per lb., and increases with decreases in heating value.

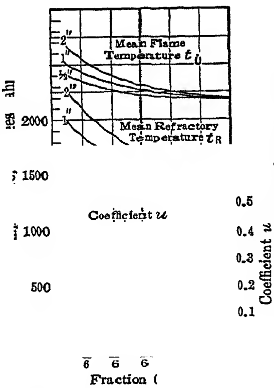


FIG. 59. Relations of Flame Temperature, Refractory Temperature, Refractory Thickness, ψ and u. Heat Release Rate, 25,000 B.t.u. per hr. per sq. ft. per deg. F.

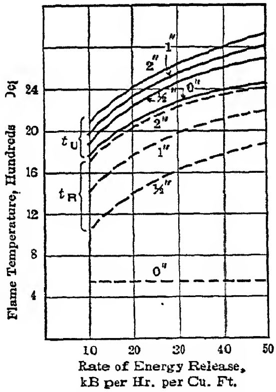


FIG. 60. Relation of Refractory Thickness and Energy Release Rate in Furnace where ψ = ½

EFFECT OF FURNACE VOLUME.—The effect of furnace volume with respect to flame temperature when burning pulverized coal in a completely water-cooled furnace, is shown in Fig. 58. A similar effect is shown on refractory-wall temperatures. Fig. 57 shows that, with pulverized coal, increasing furnace volume has a decreasing effect on flame temperature with larger furnaces. Changes in volume of stoker-fired furnaces affect these temperatures but little.

REFRACTORY FACINGS on water-cooled wall surfaces improve combustion conditions, especially at lower fuel-burning rates. Larger furnaces are required for the same steam output. Refractory-faced water-cooled walls permit higher furnace temperatures and greater heat-release rates than solid refractory walls.

Wohlenberg and Brooks have shown (*Trans.* A.S.M.E., FSP–50–39, 1928) that water cooling results in but small thermal advantage with flat water-cooled walls if the refractory face is over 2 in. thick, intimately bonded to the metal wall. Conductivity of refractory assumed at 10 B.t.u. per hr. per deg. F. per sq. ft. Fig. 59 shows the variation, with fraction cold, ψ, and thickness of refractory, of flame temperature, of temperature of inner surface of refractory and of ratio u (= heat absorbed by furnace wall ÷ heat released in furnace). Conditions given are as follows: Coal, West Virginia or Illinois; furnace, 20 × 20 × 20 ft.; air, 70° F., 20% excess; refractory, maximum thickness, 2 in., conductivity k, 10 B.t.u. per hr. per deg. F. per sq. ft. (for any other conductivity, multiply thickness in Fig. 59 by 0.1 k). Fig. 60 shows the relation between flame temperature t_U and refractory temperature t_R when heat release rate and refractory thickness are varied, furnace conditions being as above. Change of entering air temperature from 70° to 500° F. will have the following effects: t_U increases about 50° F.; t_R increases about 25° F. with refractory thickness of $1/2$ in., and about 100° F. with 2 in. thickness. Increasing thickness of refractory facing requires larger furnace volume for maintenance of the same flame temperature. Thus, for a given furnace temperature, heat release rates in B.t.u. per hr. per cu. ft. would be as follows: Bare water walls, 30,000; $1/2$-in. facing, 23,000; 2-in. facing, 16,000. Relative furnace volume would be inversely as the heat release rates.

Wall temperatures indicated in Figs. 53–54 are mean temperatures that may be expected. Actually, local hot spots may be 300 to 400° F. higher. These locations must have greater cooling and fraction cold. Otherwise, the materials would attain temperatures above that at which they can survive.

Manufacturers of representative air-cooled and water-cooled furnace walls are: *Air-cooled Walls:* American Arch Co., New York; Bernitz Furnace Appliance Co., Boston; Bigelow-Liptak Corp., Detroit; M. H. Detrick Co., Chicago; DeWolf Furnace Corp., Rochester, N. Y. *Water-cooled Walls:* American Engineering Co., Philadelphia; Babcock & Wilcox Co., New York; Combustion Engineering Co., Inc., New York; Riley Stoker Corp., Worcester, Mass.; Superheater Co., New York.

CHIMNEYS AND DRAFT*

1. NATURAL DRAFT

STATIC DRAFT.—The height and diameter of a properly-designed chimney depend primarily on type and amount of fuel burned, temperature of flue gases, summation of resistances in the apparatus and connections, and altitude of plant above sea level. As yet no formulas have been evolved to take all factors into consideration, and existing formulas are largely empirical.

Flow of gases through a chimney is produced by the difference in weight between the column of hot gases in the chimney and a column of equal height of outside air. Draft is the difference in pressures exerted by the two columns, and is measured by the weight per unit area. The difference of pressure usually is expressed in inches of water. The intensity, for a given gas and air temperature, varies with the height of the comparative columns, and therefore, height of chimney is a prime factor in producing the required intensity of draft. The intensity of the draft is

$$D = 0.52\,PH\{(1/T) - (1/T_1)\} \;.\;.\;.\;.\; [1]; \qquad D = KH \;.\;.\;.\;.\; [2]$$

where D = draft produced, in. of water; P = atmospheric pressure, lb. per sq. in.; H = height of stack above breeching entrance, ft.; T = atmospheric temperature, deg. F., abs.; T_1 = temperature of stack gases, deg. F., abs.; K = a factor, depending on temperature of stack gases (see Table 1). Density of flue gases is assumed to be the same as that of air. This assumption introduces, for usual fuels and operating conditions, an error of less than 1.5%.

Temperature T_1 should be considered as the effective mean temperature in the stack. It is likely to be lower than that of the gases leaving the boiler setting, as it is affected by air infiltration at leaky joints in flues or stack connections, and also by heat losses from the flues and the stack itself. Considerable doubt exists as to dependable methods for

Table 1.—Values of K at Atmospheric Pressure, 14.7 lb. per sq. in.

Stack gas temp., deg. F.	750	700	650	600	550	500	450	400	350
K at 60° F.	0.0084	.0081	.0078	.0075	.0071	.0067	.0063	.0058	.0053
K at 80° F.	0.0078	.0076	.0073	.0070	.0066	.0062	.0057	.0053	.0048

* Staff revision.

determining effective mean temperature in the stack, but for nominally tight connections, it is justifiable to use boiler outlet gas temperature, reduced 20° to 50° as a margin of safety, as a basis for calculating the chimney height for power or industrial process boilers. For domestic boilers see p. 11–11.

The effective mean temperature of the gases in the chimney, according to Cotton, is $C(T_b - T_a) + T_a$, where T_a, T_b = respectively, atmospheric air and boiler exit gas temperatures, deg. F., and C = factor from Table 2. Smallwood (*Mech. Engg.*, Jan. 1933) believes that the factors of Table 2 are low, and that actual average temperatures in the stack are very nearly equal to entering gas temperature, provided there is no air infiltration.

AVAILABLE DRAFT. LOSS DUE TO FRICTION AND INERTIA.—The intensity of draft by formula [1] is static, and never is observed with a draft gage or recorder. It does not obtain after the gases begin to flow. When boiler is in operation, *available draft* is the difference between static draft (formula [1]), and the amount lost in overcoming inertia and frictional resistance within the chimney proper, or draft indicated by a draft gage connected to the base measuring point of the stack. Loss of draft due to friction is

. . . . [3]

d = draft loss, in. of water; W = weight of gas, lb. per sec.; M = average perimeter inside of stack, ft.; H = height of stack, ft.; A = average area inside of stack, sq. ft.; f = friction coefficient. Approximate values of f are: 0.0008 for steel stacks, temperature of gases, 600° F. (0.0006 at 350°); 0.00105 for brick or brick-lined stacks, tempera-

Table 2.—Factors for Effective Mean Temperatures of Gases in Chimney.*

Calculated from Fig. 5 in Determination of Chimney Sizes, *Mech. Engg.*, Sept., 1923

Height of Chimney, ft.	Chimney Diameter, ft.					
	6	10	14	18	22	26
	Factor C					
100	0.86	0.875	0.89	0.91	0.93	0.95
200	.775	.80	.825	.85	.88	.91
300	.74	.775	.80	.83	.87	.90
400	.725	.76	.79	.825	.865	.895

* Brick Chimney. These factors may be greatly modified if the chimney is not air tight.

Table 3.—Available Draft for 100-ft. Steel Stacks

Based on atmospheric temperature of 80° F., stack temperature of 500° F. and 2 lb. of gas per lb. of steam from and at 212° F. at sea level. For other heights of stack, H, multiply draft by $H/100$*.

Steam per Hour, lb.	Diameter of Stack, Inches																		
	36	42	48	54	60	66	72	78	84	90	96	102	108	120	132	144	156	168	180
	Available Draft, Inches of Water																		
5,000	0.60	0.61	0.61	.....	.....	.....	.....	.....	.....	.....	.....	.....	.....	.....	.....	.....	.....	.....	.....
10,000	.54	.58	.60	0.61	0.61	0.61	.....	.....	.....	.....	.....	.....	.....	.....	.....	.....	.....	.....	.....
15,000	.44	.53	.58	.60	.60	.60	0.61	.....	.....	.....	.....	.....	.....	.....	.....	.....	.....	.....	.....
20,000	.30	.47	.55	.58	.59	.60	.61	.....	.....	.....	.....	.....	.....	.....	.....	.....	.....	.....	.....
25,000	.....	.38	.51	.56	.58	.59	.60	0.60	0.61	.....	.....	.....	.....	.....	.....	.....	.....	.....	.....
30,000	.....	.28	.46	.53	.56	.58	.60	.60	.61	.....	.....	.....	.....	.....	.....	.....	.....	.....	.....
35,000	.....	.....	.40	.50	.54	.57	.59	.60	.60	0.61	.....	.....	.....	.....	.....	.....	.....	.....	.....
40,000	.....	.....	.33	.46	.52	.56	.58	.59	.60	.60	0.61	.....	.....	.....	.....	.....	.....	.....	.....
45,000	.....	.....	.....	.42	.50	.54	.57	.58	.60	.60	.61	0.61	0.61	.....	.....	.....	.....	.....	.....
50,000	.....	.....	.....	.37	.46	.52	.56	.57	.59	.60	.60	.61	.61	.....	.....	.....	.....	.....	.....
60,000	.....	.....	.....	.....	.40	.48	.53	.56	.58	.59	.60	.60	.61	.....	.....	.....	.....	.....	.....
70,000	.....	.....	.....	.....	.32	.43	.50	.54	.57	.58	.59	.59	.60	0.61	.....	.....	.....	.....	.....
80,000	.....	.....	.....	.....	.....	.37	.48	.52	.55	.57	.58	.59	.60	.61	.....	.....	.....	.....	.....
90,000	.....	.....	.....	.....	.....	.31	.42	.49	.53	.56	.57	.58	.59	.60	0.61	.....	.....	.....	.....
100,000	.....	.....	.....	.....	.....	.....	.37	.46	.51	.54	.56	.57	.58	.60	.61	0.61	.....	.....	.....
120,000	.....	.....	.....	.....	.....	.....	.....	.38	.46	.50	.54	.55	.57	.60	.60	.61	0.61	.....	.....
140,000	.....	.....	.....	.....	.....	.....	.....	.30	.40	.46	.52	.54	.56	.58	.59	.60	.61	.....	.....
160,000	.....	.....	.....	.....	.....	.....	.....	.....	.33	.41	.47	.51	.54	.57	.58	.60	.60	0.61	0.61
180,000	.....	.....	.....	.....	.....	.....	.....	.....	.....	.36	.43	.47	.50	.56	.58	.59	.60	.61	.61
200,000	.....	.....	.....	.....	.....	.....	.....	.....	.....	.30	.38	.44	.48	.54	.56	.58	.59	.60	.60
240,000	.....	.....	.....	.....	.....	.....	.....	.....	.....	.....	.30	.37	.42	.51	.55	.58	.58	.59	.60
280,000	.....	.....	.....	.....	.....	.....	.....	.....	.....	.....	.....	.....	.36	.45	.53	.56	.57	.58	59

* For other stack and atmospheric temperatures add or deduct before multiplying by H 100 as follows: For 60° F. atmospheric temperature, add 0.05 in.; for 100° F. atmospheric temperature, deduct 0.05 in.; for 750° F. stack temperature add 0.17 in.; for 700° F., add 0.14 in.; for 650° F., add 0.11 in.; for 600° F., add 0.08 in.; for 550° F., add 0.04 in.; for 450° F., deduct 0.04 in.; for 400° F., deduct 0.09 in.; for 350° F., deduct 0.14 in.

ture of gases 600° F. (0.0008 at 350°). If the stack is required to impart velocity to the gas stream, as from space of infinite volume where initial velocity is zero, further correction of theoretical draft is necessary.

The available draft, D_a, for any stack is, using the same notation as before,

$$D_a = 0.52\,PH\{(1/T) - (1/T_1)\} - (fW^2\,MH/A^3). \quad [4]$$

$$= KH - (fW^2\,MH/A^3) \quad [5]$$

Table 3 calculated from formula [5], gives available draft that a steel stack, 100 ft. high from base measuring point, will produce when serving boilers at different steaming rates.

ECONOMICAL SIZE OF CHIMNEY.—Formula [5] indicates that if a stack of given diameter be made higher, it will produce the same available draft as one of larger diameter, the additional height being required to overcome the added friction losses. The most economical stack is the one that will meet draft requirements at the least construction costs. Deilein and Cotton have suggested that the least expensive of several possible chimneys will be the one in which (height × diameter) is the smallest.

A Convenient Method for Determining Chimney Diameters, that has been accepted by many as satisfactory, is to use a diameter giving 1 sq. ft. of cross-sectional area for each 1000 lb. per hr. of flue gas.

CORRECTION FOR ALTITUDE. ACTION OF WIND MOVEMENT.—From equation [1], draft produced by a given height of chimney is proportional to barometric pressure. Conversely, for a given draft, required height is inversely proportional to barometric pressure. Hence, stacks operating at altitudes higher than sea level must be increased both in height and diameter, as increased height causes added frictional resistance in the stack. The increase in height over that at sea level is proportional to the inverse ratio of the barometric pressures, while stack diameter increases as the $^2/_5$ power of the inverse ratio. Table 4 gives correction factors for altitudes above sea level.

When calculating draft available with any stack, possible effects from the action of the air at the top of the stack are not considered.

KENT'S EMPIRICAL FORMULA.—William Kent's empirical formula (*Trans.* A.S.M.E. vi, 1885, p. 81) is based on: 1. Velocity of the gas varies as the square root of the height. 2. Retardation of ascending gases by friction within the stack has the effect of decreasing the inside-cross-sectional area, or of lining the chimney with a layer of gas of no velocity. Thickness of lining is assumed to be 2 in. for all chimneys, or a decrease in area equal to the perimeter multiplied by 2 in. (neglecting overlapping of corners of the lining). If D = diameter, ft.; A = area, sq. ft.; effective area E, sq. ft., is

Square chimneys, $$E = D^2 - (8D/12) = A - {}^2/_3\sqrt{A} \quad [6]$$

Round chimneys, $$E = A - 0.591\sqrt{A} \quad [7]$$

The coefficient of $\sqrt{A}$ may be assumed as equal to 0.6, thus reducing the formulas to

$$E = A - 0.6\sqrt{A} \quad [8]$$

3. Boiler horsepower capacity varies as effective area E. 4. Available draft is sufficient to effect combustion of 5 lb. of coal per hr. per rated boiler horsepower.

Since power of the chimney varies directly as effective area E, and as the square root of height H, the formula for the horsepower of a boiler for a given size chimney becomes

$$\text{Boiler Hp.} = CE\sqrt{H} \quad [9]$$

C is a constant. Its average value, from plots of actual test results, is 3.33; therefore

$$\text{Boiler Hp.} = 3.33\,E\sqrt{H} \quad [10]$$

$$= 3.33(A - 0.6\sqrt{A})\sqrt{H} \quad [11]$$

With a given boiler Hp., height usually is assumed and

$$E = 0.3\,\text{Boiler Hp.}/\sqrt{H} \quad [12]$$

Table 4.—Altitude Correction Factors for Stack Capacity

Altitude Above Sea Level, ft.	Normal Barometer, in. Hg	R, Ratio of Increase in Stack Height	$\sqrt[5]{R^2}$, Ratio of Increase in Stack Diameter	Altitude Above Sea Level, ft.	Normal Barometer, in. Hg	R, Ratio of Increase in Stack Height	$\sqrt[5]{R^2}$, Ratio of Increase in Stack Diameter
0	30.00	1.000	1.000	6,000	23.87	1.257	1.096
1,000	28.88	1.039	1.015	7,000	22.97	1.306	1.113
2,000	27.80	1.079	1.030	8,000	22.11	1.357	1.130
3,000	26.76	1.121	1.047	9,000	21.28	1.410	1.147
4,000	25.76	1.165	1.063	10,000	20.49	1.464	1.165
5,000	24.79	1.210	1.079				

Table 5.—Sizes of Chimneys for Steam Boilers *

Based on Kent Chimney Formula [11]

Diam., in.	Area A, sq. ft.	Effective Area, $E = A - 0.6\sqrt{A}$, sq. ft.	Height of Chimney, ft. — Commercial Horsepower of Boiler: 40	50	60	70	80	90	100	110	125	150	175	200	225	250	275	300	325	350	375	400	Equivalent Square Chimney, Side of Square, in. $(12\sqrt{E})+4$
18	1.77	0.97	21	23	25	27	29	...	...	...	...	...	...	...	...	...	...	...	...	...	...	...	16
21	2.41	1.47	31	35	38	41	44	...	...	...	...	...	...	...	...	...	...	...	...	...	...	...	19
24	3.14	2.08	44	49	54	58	62	66	...	...	...	...	...	...	...	...	...	...	...	...	...	...	22
27	3.98	2.78	59	65	72	78	83	88	...	...	...	...	...	...	...	...	...	...	...	...	...	...	24
30	4.91	3.58	...	84	92	100	107	113	119	...	...	...	...	...	...	...	...	...	...	...	...	...	27
33	5.94	4.48	...	...	115	125	133	141	149	156	...	...	...	...	...	...	...	...	...	...	...	...	30
36	7.07	5.47	...	...	141	152	163	173	182	191	204	...	...	...	...	...	...	...	...	...	...	...	32
39	8.30	6.57	...	...	...	183	196	208	219	229	245	268	...	...	...	...	...	...	...	...	...	...	35
42	9.62	7.76	...	...	...	216	231	245	258	271	289	316	342	...	...	...	...	...	...	...	...	...	38
48	12.57	10.44	...	...	...	...	311	330	348	365	389	426	460	492	...	...	...	...	...	...	...	...	43
54	15.90	13.51	...	...	...	...	...	427	449	472	503	551	595	636	675	...	...	...	...	...	...	...	48
60	19.64	16.98	...	...	...	...	...	536	565	593	632	692	748	800	848	894	938	...	...	...	...	...	54
66	23.76	20.83	...	...	...	...	...	...	694	728	776	849	918	981	1,040	1,097	1,150	1,201	...	...	...	...	59
72	28.27	25.08	...	...	...	...	...	...	835	876	934	1023	1105	1,181	1,253	1,320	1,385	1,447	1,506	...	...	...	64
78	33.18	29.73	...	...	...	...	...	...	...	1038	1107	1212	1310	1,400	1,485	1,565	1,642	1,715	1,785	1,852	...	...	70
84	38.48	34.76	...	...	...	...	...	...	...	1214	1294	1418	1531	1,637	1,736	1,830	1,920	2,005	2,087	2,166	2,241	...	75
90	44.18	40.19	...	...	...	...	...	...	...	...	1496	1639	1770	1,893	2,008	2,116	2,219	2,318	2,413	2,504	2,592	2,677	80
96	50.27	46.01	...	...	...	...	...	...	...	...	1712	1876	2027	2,167	2,298	2,423	2,541	2,654	2,762	2,866	2,967	3,064	86
102	56.75	52.23	...	...	...	...	...	...	...	...	1944	2130	2300	2,459	2,609	2,750	2,884	3,012	3,136	3,254	3,368	3,479	91
108	63.62	58.83	...	...	...	...	...	...	...	...	2090	2399	2592	2,771	2,939	3,098	3,249	3,393	3,532	3,665	3,794	3,918	96
114	70.88	65.83	...	...	...	...	...	...	...	...	...	2685	2900	3,100	3,288	3,466	3,635	3,797	3,952	4,101	4,245	4,384	101
120	78.54	73.22	...	...	...	...	...	...	...	...	...	2986	3226	3,448	3,657	3,855	4,043	4,223	4,396	4,562	4,722	4,876	107
132	95.03	89.18	...	...	...	...	...	...	...	...	...	3637	3929	4,200	4,455	4,696	4,925	5,144	5,354	5,556	5,751	5,939	117
144	113.10	106.72	...	...	...	...	...	...	...	...	...	4352	4701	5,026	5,331	5,618	5,893	6,155	6,407	6,649	6,882	7,108	128
156	132.73	125.82	...	...	...	...	...	...	...	...	...	...	5542	5,925	6,285	6,624	6,948	7,264	7,553	7,838	8,114	8,380	138
168	153.94	146.51	...	...	...	...	...	...	...	...	...	...	6454	6,899	7,318	7,713	8,091	8,459	8,795	9,127	9,448	9,758	150
180	176.71	168.76	...	...	...	...	...	...	...	...	...	...	7434	7,947	8,430	8,886	9,318	9,734	10,131	10,514	10,883	11,240	160
192	201.06	192.55	...	...	...	...	...	...	...	...	...	...	8482	9,068	9,618	10,138	10,633	11,106	11,559	11,996	12,417	12,824	172
204	226.98	217.94	...	...	...	...	...	...	...	...	...	...	...	10,263	10,886	11,475	12,035	12,570	13,083	13,577	14,054	14,515	181
216	254.47	244.90	...	...	...	...	...	...	...	...	...	...	...	11,533	12,233	12,894	13,524	14,125	14,702	15,257	15,792	16,310	192
228	283.53	273.43	...	...	...	...	...	...	...	...	...	...	...	...	13,658	14,397	15,099	15,771	16,414	17,034	17,632	18,210	202
240	314.16	303.53	...	...	...	...	...	...	...	...	...	...	...	...	15,161	15,981	16,761	17,507	18,222	18,909	19,573	20,215	213

* Calculated on basis of 1 Boiler Horsepower = 5 lb. of coal burned per hour. For any other rate of combustion multiply the value in the table by the ratio of 5 to the actual rate of combustion.

For round chimneys diameter, in inches, is $d = (13.54\sqrt{E} + 4)$, and the side of a square chimney, in inches, is $s = (12\sqrt{E} + 4)$. Table 5 has been calculated from formula [11].

If the coal consumption varies from 5 lb. per boiler Hp. per hr., the actual capacity of the stack expressed in terms of boiler Hp. will be the boiler Hp. indicated by the formula, multiplied by the ratio of 5 to the actual coal consumption per boiler horsepower.

An approximate formula for chimneys above 1000 boiler Hp. is

$$\text{B.Hp.} = 2.5 D^2 \sqrt{H} \qquad [13]$$

RELATION OF AVAILABLE DRAFT AND DRAFT LOSSES.—Available draft must equal the summation of draft losses, which are the various flow resistances encountered. They can be expressed as $D_t - h_f = D_A = h_F + h_B + h_R + h_A$, in which D_t = static draft; h_f = loss of draft due to friction and inertia of chimney gases; D_A = available draft; h_F = loss of draft through fuel bed; h_B = loss of draft through the boiler, h_R = draft loss through breeching, including loss due to turns, bends, and damper friction; h_A = additional losses due to added auxiliary equipment (economizers, air heaters, etc.) and to the sudden enlargement of an opening or section.

Draft Loss in Furnace or through fuel bed varies with the type and condition of coal, rate of combustion, volume of interstices of the coal, percentage of ash, and thickness of fuel bed. In general, when burning coal, the loss of draft through the fuel bed increases as the percentage of volatile matter decreases and as the percentage of fixed carbon increases. The loss, therefore, is least for free-burning bituminous coals and greatest for small sizes of anthracite. Table 6 gives values which may be used as a basis for estimating. With pulverized fuel, gas, and oil there is no draft loss through the fuel bed, and h_F represents draft loss through burner parts.

Draft Loss through Boiler varies between wide limits, and depends on the type and size of boilers, arrangement of heating surface and baffles, type and quantity of fuel burned, combustion conditions, and design of furnace and boiler setting. Losses in well-designed water-tube boilers will be about 0.25 in., 0.4 in., and 0.65 in., at 100, 150, and 200% of boiler rated capacity, respectively. Fire-tube boilers with tubes 18 ft. long, 2 in. O.D., and an entering gas temperature of 1800° F., will have a draft loss of approximately 1 in. of water at rated boiler capacity, and 2 1/4 in. water at 150% of rated capacity. These figures should be used only in preliminary calculations, since they may represent values far from the actual loss of draft through the boiler actually used. Value selected should represent the draft loss through the boiler, when its steaming rate is a maximum.

Draft Loss in Straight Flues due to friction can be calculated approximately from formula [3], taking M as actual perimeter of the flue in ft., and H as length in ft. The greater resistance of the more or less uneven surface in concrete flues is provided for in the values of the constants given for formula [3]. The retarding effect of a square flue is 12% greater than that of a circular flue of equal area. Short, 90-deg. turns reduce draft by approximately 0.05 in., for each turn. Turns from the boiler into the flue and from a flue into the stack should be included in calculations. Cross-sectional area of flues should be of ample size to provide against undue frictional loss, and it is wise to allow 1 sq. ft., of flue area for each 2500 lb. per hr. of flue gas. The area of a flue at any point should be proportional to the volume of gases passing at that point; therefore, the area of a flue, serving more than one boiler, should progressively increase as it approaches the stack. With circular flues of approximately the same size as the stack, or reduced proportionately to the volume of gases they will handle, a convenient rule is to allow 0.1 in. draft loss per 100 ft. of flue length, and 0.05 in. for each 90-deg. turn. These figures are good for square or rectangular flues with areas sufficiently large to provide against excessive frictional loss. For brick or concrete flues they should be increased 35%.

STACKS FOR FUELS BURNED IN SUSPENSION, as pulverized coal, oil, or natural gas, require heights and diameters less than those for similar stoker-fired boilers. At a given steam rate the required area of stack, and the draft loss through the boiler, when burning fuels in suspension, will be less because of the smaller volume of gases. However, with oil or natural gas, the temperature of gases entering the stack will be lower, resulting

Table 6.—Draft Loss through Fire with Various Grades of Coal

Kind of Coal	Lb. of Dry Coal per sq. ft. of Grate per hr.						
	15	20	25	30	35	40	45
	Force of Draft, in. of Water						
Ill., Ind., Kan., bituminous	0.14	0.20	0.26	0.33	0.40	0.48	0.57
Ala., Ky., Pa., Tenn., bituminous	.16	.23	.31	.40	.49	.60	.72
Md., Pa., Va., W. Va., semi-bituminous	.18	.26	.35	.45	.57	.71	.87
Anthracite pea	.30	.45	.64	.88	1.23	......	......
Anthracite buckwheat No. 1	.43	.68	1.00	1.50	......	......	......

in a decrease of available draft from a stack of given height. A draft suction must be assured within all parts of the setting, under all operating conditions. If the suction is not sufficient to remove products of combustion the heat is detrimental to the setting.

Present day practice of burning fuels in suspension is to admit combustion air with the fuel by means of a turbulent type of burner. When burning relatively small amounts of fuel, it may be possible to provide sufficient draft by introducing air at room temperature through the burner throat. Most turbulent burners using natural draft are designed to operate at a draft loss of approximately 0.3 in. of water.

In burning fuels in suspension with natural draft burners, the possibility should be eliminated of too much draft at low steaming rates, with consequent high quantities of excess air. Stack height, however, must be sufficient to provide draft necessary for operation at maximum capacity.

MECHANICAL DRAFT.—When the natural draft provided by a stack is insufficient to overcome the losses attending the admission of air for combustion and the subsequent travel of the products of combustion over the heat transfer surfaces, other means must be used to make up the deficiency. In general, this situation exists when the total draft required exceeds 1.5 in. of water.

When natural draft will carry the normal operating load, and peak loads are infrequent and of short duration, expensive auxiliary equipment is not justifiable. In such cases, steam jets may be placed in air-tight ashpits under the grates, to overcome the resistance of the fuel bed. The burning of coals which have a tendency to mat on the grate is materially aided by the presence of the steam. Jets also may be located in the breeching or in the base of the stack to impart kinetic energy to the products of combustion. When the stack is of moderate or of large diameter, one or more rings of jets is necessary. Because of the large quantities of steam required, this method is uneconomical for creating pressures of over 1 in. of water, or suctions above 0.75 in., in excess of that given by the stack alone. A more flexible method of economically meeting draft requirements for wide variations of load is the use of forced or induced draft fans, or both.

Forced draft fans supply the air for combustion at sufficient pressure to overcome the resistances encountered by the air in its passage to the furnace. Such resistances may be the friction losses through air heaters, ducts, heat exchangers, burners or fuel beds.

A combination of forced and natural draft is used when the stack is capable of maintaining a sufficient suction in the furnace to remove the products of combustion from the apparatus. Induced draft alone is used, when the resistance of the fuel burning equipment is not excessive, and the fan is capable of supplying sufficient suction to overcome this, in addition to supplying draft to the rest of the apparatus.

As equipment is added behind the boiler to increase the efficiency of the unit, greater suction is required to overcome the resistance of these auxiliaries. The lower exit gas temperature reduces the stack draft and induced draft fans are installed to supply the necessary suction. Since these fans handle dust-laden gases, especially in waste heat work, they are subject to erosion and consequent maintenance costs. Their service conditions are more severe than those of forced draft fans for other reasons: the temperature of the gas is higher and the weight handled is greater. They are the only practical means of increasing the capacity of waste heat units.

Forced and induced draft fans are so operated as to give a balanced draft, or better, a slight suction in the furnace at all times. This usually is accomplished by some means of automatic control, based either on steam pressure or furnace suction. In some large central stations, forced and induced draft fans are furnished to deliver static pressures and suctions, respectively, of from 12 to 14 in. of water.

For data on selection of fans see pp. 1–57 to 1–79. See also the chapters on Mechanical Draft in Air Conditioning and Engineering, published by American Blower Corp., and in Fan Engineering, published by Buffalo Forge Co.

2. DESIGN AND CONSTRUCTION OF CHIMNEYS

General classes of chimneys are: 1. Steel, guyed. 2. Steel, self-supporting. 3. Masonry. 4. Reinforced concrete. In recent years a large number of steel chimneys have been installed due to their advantages over masonry construction. These include less weight and lower cost for a given size, ease of construction, and less surface exposed to wind. Steel stacks are economical where space limitations necessitate or warrant erection of the stack over the boiler. A special breeching then usually forms the stack foundation, both breeching and stack being supported by the boiler structural work.

WIND PRESSURE.—The relation between wind velocity and wind pressure per sq. ft. of flat surface usually is expressed by $P = KV^2$, where P = wind pressure, lb. per sq. ft. of flat surface; V = wind velocity, miles per hr.; K = an experimental coefficient,

whose value ranges from 0.0029 to 0.005; 0.004 generally is used. Practically all tests to determine K have been made on small flat surfaces. Later tests show that average pressure on large surfaces is less than on small surfaces. Prof. Kernot states that it does not exceed approximately $^2/_3$ of that upon surfaces of 1 or 2 sq. ft.

In stack design, total wind pressure is considered as the product of pressure per sq. ft. and projected area. Effective pressure per sq. ft. of projected area (flat surface = 1.00) is 0.80 for hexagonal, 0.71 for octagonal, and 0.67 for round stacks. Ordinarily, in stack calculations, 100 miles per hr. is used as maximum wind velocity. This corresponds to a pressure of 40–50 lb. per sq. ft. of flat surface, depending on value of K used, or a 26–33 lb. per sq. ft. of projected area for a round stack. In the U. S., 25–30 lb. per sq. ft. of projected area is commonly used for round stacks. Most cities set, by building ordinance, the allowable wind pressure to be used in stack design. European practice usually considers the variation in pressure between base and top of stack.

STABILITY OF CHIMNEYS.—Stresses in chimneys result from wind pressure and weight of the chimney, whose combined effect tends to overturn the chimney. Pressure due to weight of the stack, with no wind movement, is distributed uniformly over the bearing surface at its base. Wind pressure tends to overturn the stack. The resultant pressure at the base, due to wind pressure and weight, decreases on the windward side and increases on the leeward side. This, in effect, moves the center of pressure away from the center of gravity, toward the leeward side. Distance e, or eccentricity, from center of gravity to center of pressure is $e = M/W$, where e = eccentricity, ft.; M = wind moment, ft.-lb. (total wind pressure, lb. × distance from section under consideration to center of wind pressure); W = weight of stack above section under consideration, lb.

The condition of least stability is when resultant pressure at the windward side becomes zero. The distance q from center of gravity to center of pressure for the condition of least stability is the *radius of statical moment* or *radius of the kern*. For stability, assuming that tension is not allowable, center of pressure must fall within the area of the kern and $e \lesseqgtr q$ or $M \lesseqgtr Wq$. Values of q for sections of stacks and bases encountered in chimney design, as given by G. F. Gebhardt (Steam Power Plant Engineering, 6th Ed. 1928) are

Section	q	Section	
Solid Circular	$D/8$	Solid Square (min.)	$0.118L$
Hollow Circle	$(D^2 + d^2)/8D$	Hollow Square (max.)	$(L^2 + l^2)/6L$
Solid Square (max.)	$L/6$	Hollow Square (min.)	$0.118(L^2 + l$

D, d = outside and inside diam., respectively, ft.; L, l = length of outer and inner sides, ft.

SELF-SUPPORTING STEEL STACKS usually are made with a straight conical flare at the base, the apex of the cone being at the top of the stack, and the height of the frustum approximately one-fourth the height of the stack. The conical section is built of plates of uniform thickness with a base diameter about $^4/_3$ × diam. of straight portion of the stack. By flaring out the base, a larger diameter is provided for receiving the flue opening, resulting in a better flow of gases and necessitating less reinforcement than would be necessary with a straight stack. In fabricating the stack, each section may be made a truncated cone overlapping the previous section enough to allow for riveting. Better practice uses a true cylindrical section, with one end provided with an overlap for riveting. Stacks may be built with or without linings. Vitrified asbestos linings, seldom over 4 $^1/_2$ in. thick and supported by the stack, have replaced the earlier independent brick linings.

In design, due to the liberal factor of safety, and because a cylinder of large diameter with thin walls will probably fail by flattening or buckling on the leeward side, stack weight is neglected and the stack treated as a hollow cylindrical cantilever beam subjected only to the wind pressure. Therefore, at any section, $S = M \div (I/c)$, where S = stress in outer fiber due to wind pressure, lb. per sq. in.; I/c = section modulus; M = wind moment, in.-lb. The section modulus for a hollow cylinder $= \pi(d_1^4 - d_2^4)/32\, d_1$, but since the thickness of the wall is a small fraction of the diameter, the modulus becomes approximately $0.785\, d_1^2\, t$, and $S = M/0.785\, d_1^2\, t$, where d_1 = outside diam. of stack, in.; t = wall thickness, in. Considering stress per linear inch instead of per square inch, stress, S_2, per inch of circumference is $S_2 = M/0.785\, d_1^2 = M/A$, where A = the cross-sectional area of stack, sq. in. Maximum stress due to wind pressure per linear inch of circumference is determined at each joint. Table 7 gives the values for use in design of self-supporting steel stacks. Calculations are based on a wind pressure of 25 lb. per sq. ft. of projected area, weight of stack is neglected, and there should be no danger of failure should corrosion reduce plate thickness $^1/_{16}$ in. Stack weight will increase stress in the compression, or leeward side, by approximately 5 to 12% and decrease stress in the tension, or windward side, the same amount.

When calculating total wind pressures, wind moments, section moduli and bearing pressures, the stack is divided into sections from the top downward, all calculations being made above the lower edge of the section under consideration. In designing self-supporting stacks, if stack weight be neglected, stress per sq. in. or per linear inch of circumference can be calculated as described above.

In order to simplify the formula used to determine stress S_2 per inch of circumference, a wind pressure of 25 lb. per sq. ft. of projected area is assumed. Then $S_2 = M/A = 150\,H^2\,D/A$, where H = height, ft., from section considered to top of stack; D = diam. of stack, ft.; A = cross-sectional area, sq. in. The self-supporting steel stacks in Table 8 were calculated on the basis of Table 7. The stacks are unlined, but if a lining is to be used, calculations should be made for bearing and shear of rivets, limits of which should not exceed those given in Table 7.

STACK FOUNDATIONS are designed in accordance with the type of stack supported. Foundations for masonry stacks have practically no effect upon stability, and are designed to provide support only, with no particular attention to weight or distribution of weight. Steel and reinforced concrete stacks are anchored to the foundation, the stack and foundation forming an integral structure. Weight and shape of foundation affect stability of the stack, and foundations must be designed to meet conditions of flexure and stability.

Maximum Pressure on the Soil is the sum of pressures due to weight and to wind moment. Foundation sizes in Table 8 are based on pressures due to weight, of 160 lb. per sq. ft. per ft. of depth of foundation, and 160 lb. due to wind, or a total of 320 lb. per sq. ft. per ft. depth of foundation.

Table 7.- -Basis of Calculations of Self-supporting Steel Stacks

(Babcock & Wilcox Company)

Pull per in. of Circumference, lb.	Plate Thickness, in.	Diameter of Rivets, in.	Pitch of Rivets, in.	Stress at Rivets, lb. per sq. in.			
				Bearing	Shear	Tension	Tension in body of plate
600	3/16	3/8	2	15,800	9,270	4,020	3200
1100	1/4	1/2	2	16,600	9,950	5,980	4400
1700	5/16	5/8	2	16,600	10,000	8,080	5440
2360	3/8	3/4	(2 rows) 4	16,100	9,860	7,830	6295
3100	7/16	3/4	(2 rows) 3	13,600	9,750	9,600	7090
3930	1/2	7/8	(2 rows) 3	13,000	9,150	11,400	7690
4800	9/16	1	(2 rows) 3 1/2	14,480	10,050	12,100	8545

Table 8.—Data for Construction of Self-supporting Steel Stacks

(Babcock & Wilcox Company, 1935)

			Concrete Foundation (Square)			Foundation Bolts		Bottom Section Including Flare	2nd Section		3rd Section		4th Section		5th Section		6th Section		7th Section		Flare Conical
D				Height, ft.	Earth Press lb. per sq.			Materi				Material,									
48			15.0	5.0	1600			5/16	15	1/4	40	3/16									4' 7''
54			17.2	5.7	1840			5/16	15	1/4	45	3/16									6' 0''
60			19.8	6.6	2110		1 1/8	3/8	15	5/16	20	1/4	45	3/16							6' 7''
66			20.3	6.8	2180	22	1 3/8	5/16	20	1/4	0	3/16									8' 1''
72	150	1023	22.8	7.6	2430		1 7/8	3/8	15	5/16	20		50	3/16							9' 0'
78	150	1212	23.3	7.8	2500		1 7/8	3/8		5/16	20	1/4	55	3/16							
78	175	1310	25.1	8.4	2690	20	2 1/8	7/16	15	3/		/16	20	1/4	55	3 16					9' 6''
84	175	1531	25.6	8.5	2720		2 1/8	7/16					20	1/4	55	3 16					9' 10''
90	175	1770	26.0	8.7	2780		2 1/8	7/16					20	1/4	60	3/16					10' 1''
90	200	1893	27.8	9.3	2980	20	2 1/2	1/2							20	1/4					10' 4'
96	200	2167	28.3	9.4	3010	20	2 1/2	1/2						5/16	20	1/4	60	3 16			10' 8'
96	225	2298	30.0	10.0	3200	20	2 3/4	9/16	15	1/2		7/16	20	3/8	20		20	1/4	60	3/16	10' 11''
108	225	2939	30.9	10.3	3240	24	2 1/2	1/2	20	7/16	20	3	20	5/16	25	1/4					12' 3''
120	225	3657	31.	10.6	3300	24	2 1/2	1/2	20	7/16	20	3/8	20					16			13' 3'
120	250	3855	33.4	11.1	3550	26	2 3/4	9/16	20	1/2	20	7/16					25	1/4	65		13' 2''
132	225	4455	32.5	10.8	3460	32	2 1/8	7/16	20	3/		5/16		1/4							15' 5'
132	250	4696	34.2	11.			2 1/2	1/2	20	/16	20	3/8	25	5/16	95						15' 3'
144	250	5618	34.9	11.6	3710	30	2 1/2	1/2			20			5/16	100	1/4				...	16' 3''
144	275	5890	35.	11.9	3800	28	2 3/4	9/16	20	1/2		7/16	20	3/8	25	5/16	100	1/4		...	16' 1''

GUYED STEEL STACKS are used, primarily, because of their relative cheapness. Heavy foundations are unnecessary, and usually stacks are carried by boiler structural supports. Guyed stacks seldom exceed 72 in. diameter and 100 ft. height. They generally are built of lighter material than self-supporting steel stacks. The material must be heavy enough to support its own weight, to prevent buckling under stresses due to wind pressure (assumed 25 lb. per sq. ft. of projected area), and initial tension in guy wires, and to allow a liberal margin for corrosion. Thickness of material ordinarily is based on rules determined and set by practice. See Table 9.

The overturning moment of the stack, due to its weight and wind pressure is resisted by the guy wires. These usually are supplied in one to three sets, each set consisting of 3 to 6 strands. Let B = number of guy bands; G = total number of guy wires; α = angle between stack and guy wire, usually 60°; θ = angle between guy wires of a set or band, 120° for set of 3; L = height of stack, ft.; D = diam. of stack, ft.; h_1, h_2, h_3 = height of guy bands, ft. (if only two sets of bands are used, $h_3 = 0$); W = vertical load due to weight of stack, lb.; wind pressure = 25 lb. per sq. ft. of projected area. Then calculations for maximum stress in each guy wire are:

$$\text{Total wind pressure} = 25\,DL;\quad \text{overturning moment} = 12.5\,DL^2$$

The entire overturning moment is assumed to be resisted by one strand, or guy wire, in each set of guys; thus,

Horizontal pull in each guy $= 12.5\,DL^2/(h_1 + h_2 + h_3)$ . . . . [14]

Direct pull on each guy due to wind $= 12.5\,DL^2 \operatorname{cosec} \alpha/(h_1 + h_2 + h_3)$ . . . [15]

Initial stress on each guy = 1/2 of direct pull due to wind

Maximum stress in each guy due to wind and initial stress

$$= 1.5 \operatorname{cosec} \alpha\{12.5\,DL^2/(h_1 + h_2 + h_3)\} \quad [16]$$

Calculations for vertical load on base, due to weight of stack, and maximum stress in guys when wind blows between two guy wires are:

Vertical load due to wind $= B \cot \alpha \sec (\theta/2)\{12.5\,DL^2/(h_1 + h_2 + h_3)\}$ . . . [17]

Vertical load due to initial stress $= (G/2) \cot \alpha\{12.5\,DL^2/(h_1 + h_2 + h_3)\}$ . . . [18]

Vertical load due to maximum stress in guys

$$= \{12.5\,DL^2/(h_1 + h_2 + h_3)\}\{B \sec (\theta/2) + (G/2)\} \cot \alpha \quad [19]$$

Total vertical load in base

$$= W + \{12.5\,DL^2/(h_1 + h_2 + h_3)\}\{B \sec (\theta/2) + (G/2)\} \cot \alpha \quad [20]$$

For working conditions allow a safety factor of at least 2.

MASONRY STACKS usually are of circular, octagonal, hexagonal or square section, circular stacks being most common. The use of radial brick in circular stacks is almost universal. Octagonal and hexagonal stacks require special-shaped brick for best construction.

Stacks are built either with a single or double shell. Single-shell stacks are used where the bricks are not affected by the heat. Double-shell stacks are more common, and consist of a masonry outer shell with an inner lining extending partly or all the way up the stack. The lining is independent of the outer shell. Procedure for determining strength is practically the same for both single- and double-shell stacks.

The thickness of the walls, as ordinarily built, decreases in a series of steps from bottom to top of the stack. The thickness of wall at any section is determined by a considera-

Table 9.—Approximate Weight of Guyed Steel Stacks per Foot of Height

Stack Diameter, in.	Thickness of Material				
	No. 12 B.W.G.	No. 10 B.W.G.	No. 8 B.W.G.	3/16 in.	1/4 in.
	Weight of Stack per foot of Height, lb.*				
30	41.2	50.6	.....	.....	.....
33	45.2	55.5	.....	.....	.....
36	49.3	60.6	74.5	.....	.
39	53.2	65.4	80.5	91.4	.....
42	57.1	70.4	85.4	97.0	129.3
48	65.2	80.2	97.2	111.1	150.0
54	....	91.1	110.9	124.6	168.2
60	....	101.0	122.7	139.9	183.8
66	....	....	134.8	153.8	202.3
72	....	....	146.7	167.2	219.7

* Weight includes laps, rivets and manufacturer's maximum allowance for overweight—10% should be added to these figures for guy wires, bands, clips, turnbuckles, etc.

tion of the resultant stress, at that section, due to wind and weight of stack. This stress should not put the masonry in excessive compression on the leeward side and, in general, should not result in tension on the windward side. At any section, compressive stress, S_c, due to weight of stack above that section is $S_c = W/A$, where S_c = compressive stress due to weight of stack, above section, lb. per sq. in.; W = weight of stack above section, lb.; A = area of bearing surface, sq. in.

Stress due to the wind, $S_w = M \div I/c$, where S_w = stress in outer fiber due to wind pressure, lb. per sq. in.; M = wind moment, in.-lb.; I/c = section modulus. Total compressive stress on leeward side is $S_1 = S_c + S_w$; total stress on the windward side is $S_2 = S_c - S_w$. If S_2 is positive, the stack is subjected to compression throughout the section, but if S_2 is negative, the masonry will be in tension on the windward side. Prof. Lang states that the compression on leeward side should not exceed

$$P = 71 + 0.65\,L \text{ (single shell)};\ P = 85 + 0.65\,L \text{ (double shell)}.$$

Tension on windward side should not exceed

$$P = 18.5 + 0.056\,L \text{ (single shell)};\ P = 21.3 + 0.056\,L \text{ (double shell)},$$

where P = pressure, lb. per sq. in.; L = distance, ft., from section under consideration to top of stack.

Custodis and Wiederholt Chimneys.—The Custodis chimney is constructed of a number of sizes of specially-molded, radial bricks, which conform to the circular and radial lines of each part of the chimney. The bricks contain several vertical holes, and after being set in position in the chimney the holes are filled with mortar. This forms an excellent bond, and together with the use of different lengths of radial bricks results in a thoroughly interlocked structure.

In the Wiederholt chimney, inner and outer surfaces are formed by specially designed tiles of vitrified fire-clay. The annular space between inner and outer surfaces is filled with concrete and steel reinforcing bars. This chimney combines features of both masonry and reinforced concrete chimneys.

THE REINFORCED CONCRETE CHIMNEY together with its base forms an integral structure. Wall thickness decreases progressively to the top of the stack. Less area is required than for a masonry or self-supporting steel stack because of the relatively thin walls as compared with masonry stacks and the elimination of the conical flare of the self-supporting steel stack. Reinforced concrete stacks usually are lined, either partly or to top of the stack. They can be erected rapidly, and contour easily may be changed as erection proceeds. The success of the reinforced concrete stack depends to a great extent upon the care with which material is selected, mixed and poured.

SMOKE

The Chicago Assoc. of Commerce Committee (Report on Smoke Abatement and Electrification of Railway Terminals in Chicago) defines smoke as " the gaseous and solid products of combustion, visible and invisible, including . . . mineral and other substances carried into the atmosphere with the products of combustion." Smoke from all fuels, solid, liquid or gaseous, results from non-combustion, or incomplete combustion, of volatile and heavy-hydrocarbon constituents, which are rapidly distilled and are unstable at furnace temperatures. Carbon or soot particles in smoke from solid fuels is due to incomplete combustion of the fixed carbon of the fuel.

The color of smoke, imparted to the gases by the particles of carbon, does not give a true indication of the stack loss. A small amount of carbon or soot will color large volumes of invisible or practically colorless gases, which may represent a combustion loss many times as great as that due to the actual carbon present in the gases. Gases also may be colored by particles of ash and mineral matter, neither of which represents a combustion loss.

With very dense smoke, the loss due to unconsumed carbon, passing from the stack as soot, seldom exceeds 1% of the total burned. However, the loss due to unburned, or partly burned, volatile hydrocarbons, although not indicated by the appearance of the stack, may represent an appreciable percentage of the heat value of the total fuel. Soot deposited on boiler tubes may result in a much greater loss of efficiency than that due to unburned fuel in visible smoke by reducing the conductance to the heating surface.

A plant, whose stack discharges large volumes of dense smoke, may be more economical than one with a smokeless stack. A furnace operating with a small percentage of excess air may cause considerable smoke, and yet lead to a higher evaporation rate than a similar furnace made smokeless by a large percentage of excess air.

SMOKE PREVENTION.—Smokelessness depends largely on the intelligence of the operating force, except when the furnace is wholly unsuitable for burning smoke-producing fuels. Many plants with hand-fired furnaces operate without smoke under ordinary conditions when the design provides for proper mixing of air and combustible gases and temperatures are maintained above the ignition point of the gases. If fires must be brought up quickly to maintain steam pressure, frequent use of the slice bar is necessary. This working of the fire will result in smoke.

In hand-fired furnaces, to prevent smoke, fires should be worked as little as possible. A combination of spreading and alternate firing should be used, coal being fired evenly, quickly, lightly, and often. Long flame travel of gases before striking boiler heating surfaces diminishes smoke. An extension furnace increases length of gas travel, and is particularly desirable with high-volatile coals and low boiler settings. Air introduced over the fire, heating arches, etc., to mingle air with gases distilled from the coal will diminish smoke. To prevent smoke, gases should be distilled from the fuel at a uniform rate, brought into intimate mixture with sufficient air for combustion, and have adequate temperature space and time to completely mix and burn before meeting the relatively cold boiler surfaces.

Stoker-fired furnaces under usual conditions are more nearly smokeless than hand-fired furnaces. With chain-grate stokers, ignition and mixing arches often are used to lengthen gas travel, and to permit rich gases from the front of the fuel bed to mix with excess air passing through the rear of the grate. Zoning of air supplied to different portions of the fuel bed, and proper control of fuel bed thickness and uniformity across the furnace, are important in preventing smoke. With pulverized coal, and fuels burned in suspension, burner design must insure turbulent mixing of fuel and combustion air as they enter the furnace. Fineness of particle size, or degree of atomization, also are critical factors in correct operation.

SMOKE ORDINANCES of cities and communities vary widely. They include such features as: 1. Organization of a department of smoke inspection. 2. Necessity of a permit, issued by the smoke inspector, before a new plant can be built, or an old one remodeled. 3. Necessity of a permit, issued by the inspector, before a new, or remodeled plant, can be placed in operation. 4. Regulation of emission of smoke, and penalties for violation of the ordinance.

SMOKE DETERMINATION.—The most widely known, and at one time the only method used in the qualitative determination of smoke is the Ringelmann chart. This consists of four cards ruled with horizontal and vertical lines, forming squares. Each card is 14 squares wide by 24 squares long. The width of the lines and spacing are as follows:

No. of chart	1	2	3	4
Thickness of lines, mm	1.0	2.3	3.7	5.5
Distance between lines, mm. (Length of one side of square)	9.0	7.7	6.3	4.5

At a distance of 50 ft. the lines are invisible and the cards appear to be different shades of gray, ranging from very light gray to almost black. The observer places the four cards, together with a white card and a solid black card, at a distance of 50 ft. from, and on a level with, his eyes, and in line with the stack. He rapidly compares the color of the smoke, emitted by the stack, with the cards, and judges which one corresponds with the color and density of the smoke. This method now is regarded as inaccurate, because it depends on the judgment of the observer, the angle of observation, the thickness of the gas stream and the background against which observation is made.

Various types of smoke indicators have been developed to permit the fireman to check his operation. One device resembles a periscope, one end of which is connected to the stack or breeching, while the other end, equipped with a glass observation window, is at a convenient position in the boiler room. In the stack, directly opposite the periscope opening, an incandescent lamp with a reflector projects a beam of light through the gas stream. Intensity of the light beam is affected by the amount of visible smoke in the gases, and variations may be instantly seen at the observation point in the boiler room. Automatic smoke recorders, actuated by variations in intensity of a light beam passing through a predetermined thickness of smoke layer, and falling on a photo-electric cell, also have been developed.

The method of determining the quantity of smoke adopted by the Chicago Assoc. of Commerce, consists of withdrawing a continuous sample of stack gases by means of a special pitot tube and exhauster, and entrapping the solid particles in a filter. Rate of flow through the apparatus is maintained the same as in the stack. Since area of the tube openings is fixed in relation to stack area, the weight of solid particles in the filter represents a definite proportion of the total weight emitted from the stack.

Section 7

THE STEAM ENGINE

W. Trinks

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THE STEAM ENGINE

By W. Trinks

General References.—Heck, Steam Engine and Turbine, 1911; Foster, Manchester Association Engineers, Jan. 23, 1915; Ninde, Design and Construction of Heat Engines, 1920; Ripper, Steam Engine Theory and Practice, 1914; Hütte, Vol. II, 23rd Edition, 1920; Dalby, Steam Power, 1915; Ewing, Steam Engine and Other Heat Engines, 1927. Dubbel, Kolbendampfmaschinen, 1923. Gutermuth, Die Dampfmaschinen, 4 vol., 1928. Allen, Uniflow, Back Pressure and Steam Extraction Engines, 1931.

The displacement-type steam engine, made practical by the inventions of James Watt between the years 1768 and 1790, revolutionized manufacturing, and for over a hundred years was the principal prime mover. To-day it is being supplanted by the large steam turbine, working in conjunction with electrical transmission of power, but continues to be built for a number of specific uses. In sizes up to 500 or 600 kw., the non-condensing engine is found to be more economical than the turbine. Combined heating and power plants, needing the most economical prime mover during hot weather, give preference to steam engines.

Certain types of pumping machinery, including water pumps, air compressors and gas exhausters, require wide variations of speed. For driving these, the steam engine is in great demand, both for the foregoing reasons and because the compression efficiency of the displacement-type pump is, as a rule, higher than that of the centrifugal or velocity-type pump.

For many years, economical steam engines will replace less economical units, because the arrangement of boilers and piping in many existing mills and factories favors the continued use of steam engines rather than a complete change to centralization and electrification.

CHARACTERISTIC PROPERTIES.—The steam engine is a "displacement" machine in which work is done by pressure acting on a moving piston. The whole cycle is carried on in a given space, whose walls are exposed alternately to high and to low temperatures. These conditions result in:

1. Condensation of part of the entering steam on the walls of the cylinder, and re-evaporation of the film of water near the end of expansion and during exhaust, involving a transfer of heat energy into the exhaust instead of its conversion into mechanical energy. 2. Variable torque, and cyclical speed fluctuations, necessitating either a flywheel or a multi-cylinder engine with several cranks. 3. Packing between cylinder walls and moving parts, requiring lubrication, which contaminates the exhaust steam with lubricating oil. 4. Valve gearing to alternately admit and exhaust steam.

1. CLASSIFICATION OF ENGINES

I. CLASSIFICATION AS REGARDS CONSTRUCTION.—*1a.* Horizontal Engine, Fig. 1. The motion of the piston is in a horizontal plane. For explanation of running "over" or "under," see illustration. A horizontal engine is *right hand*, if the flywheel is on the right-hand side of a person standing back of the cylinder and looking toward the shaft.

1b. Vertical Engine.—The piston moves in a vertical direction. A vertical engine may have the crank-shaft either below, Fig. 2, or above the cylinder, Fig. 3.

1c. Angle-type or Horizontal-vertical Engine, Fig. 4, has one vertical and one horizontal cylinder. Usually they act on one and the same crank.

1d. Some hoisting engines have cylinders at 45° (inverted V, Fig. 5).

2a. Single-acting Engine, Fig. 6. The steam acts against one side only of the piston and does work during only one stroke, or half a revolution.

2b. Double-acting Engine, Fig. 1. Steam acts alternately on opposite sides of the piston, doing work during the whole revolution.

3a. Reciprocating Engine, Fig. 1. That type in which the piston moves always in a straight line, but alternately in opposite directions, reversing its direction at fixed points.

3b. Rotary Engine, Fig. 7. The piston moves continuously, in a circular or other

curved, closed path, never reversing its direction of motion. The term applies only to displacement machines and not to turbines.

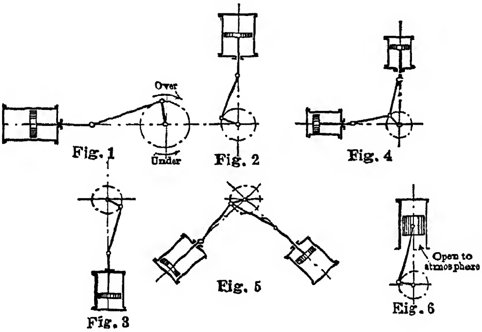
II. CLASSIFICATION AS REGARDS CONDITIONS OF OPERATION.—**1*a*. A Condensing Engine** discharges its exhaust steam into a condenser in which a pressure lower than atmospheric is maintained by condensation of the steam and exclusion of atmospheric pressure.

1*b*. A Non-condensing Engine discharges its steam either into the atmosphere or against pressure higher than atmospheric.

1*c*. A Bleeder-type or Extraction-type Engine is one from whose cylinder a part of the steam is extracted, during either expansion or compression, at a pressure that is higher than the back pressure acting during the exhaust stroke of the engine.

2*a*. Simple Engine, Figs. 1 to 6. The complete expansion of the steam from boiler pressure to exhaust pressure is carried out in one cylinder, or in each of several cylinders.

2*b*. Multi-stage Engine.—One in which the expansion of the steam is divided up into stages. The steam expands in a high-pressure cylinder from boiler pressure to an intermediate pressure; it then flows into another cylinder and expands still further, and so on.



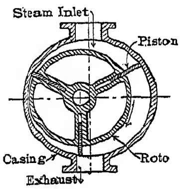


Fig. 7.
Rotary Engine

Figs. 1–6. Simple Engines

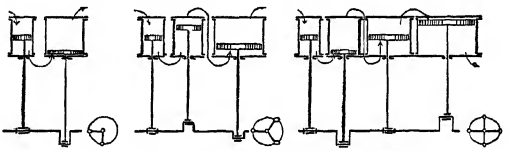


Fig. 8 Compound | Fig. 9 Triple-expansion | Fig. 10 Quadruple-expansion

Depending upon whether the expansion is divided into two, three or four stages, the engine is classified, respectively, as *compound*, Fig. 8, *triple-expansion*, Fig. 9, or *quadruple-expansion*, Fig. 10. A compound engine is a *tandem-compound* when the cylinders are arranged in line, one behind the other and both acting on the same piston rod. See Fig. 11. If the cylinders are side by side, and act upon cranks at right angles to each other, it is a *cross-compound*. See Fig. 8. If there are two low-pressure cylinders, the engine is a *three-cylinder* compound. See Fig. 12.

Twin Engines, Fig. 13, are simple engines having two cylinders side by side with cranks at 90°, as reversing mill-engines and locomotive engines.

3*a*. Uniflow Engine, Fig. 16. The steam flows in one direction only, from the ends to the center of the cylinder. The admission valves are in or near the cylinder heads; the exhaust ports are uncovered by the piston near the center of the cylinder. (See also Figs. 86 and 87.)

3*b*. Duoflow or Counterflow Engine, Figs. 14 and 15.—Steam flows alternately in opposite directions in each end of the cylinder, being first admitted and later exhausted near the ends. The term covers all engines other than the uniflow type.

III. CLASSIFICATION AS REGARDS TYPE OF VALVE AND VALVE-GEAR.—(See Valve Gears.)—1. *Slide valve* engines. 2. *Piston valve* engines. 3. *Poppet valve*

engines. 4. *Corliss* or *rocking valve* engines. 5. *Automatic* engines; the valves are always mechanically connected to the driving crank or eccentric, and the cut-off is varied automatically by a governor. 6. *Releasing engines*; the governor causes the valve to be periodically disconnected from the eccentric. The valve is closed by another force.

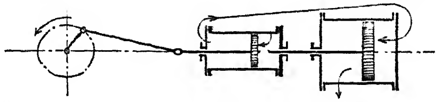


FIG. 11. Tandem-compound.

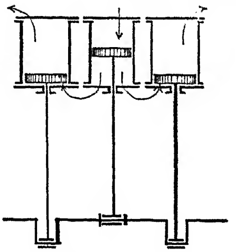


FIG. 12. Three-cylinder Compound

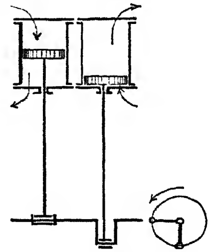
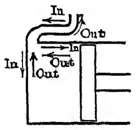


FIG. 13. Cross-compound

IV. CLASSIFICATION AS REGARDS USE.—1. Power engines drive electric generators or deliver power to machinery through a belt, rope, or shaft drive. They are, as a rule, constant-speed engines.

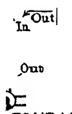
2. Blowing engines drive compressors for supplying air under pressure to blast furnaces or converters, or to compress gas for transmission through long pipe lines.

3. Hoisting engines drive hoists for elevating solid materials, or liquids in containers.



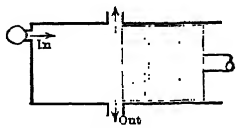
SINGLE VALVE

FIG. 14



FOUR VALVE

FIG. 15



UNIFLOW

FIG. 16

4. Pumping engines are used to elevate water or other liquids, or to supply it under pressure.

5. Reversing mill-engines drive reversing blooming-mills in steel plants.

6. Marine engines propel ocean-going or lake ships. (Engines for paddle-wheel river steamers are of an entirely distinct type from other boat engines.)

7. Automotive engines propel vehicles on land, as tractors, locomotives, and automobiles.

2. CAPACITY OF STEAM ENGINES

HORSEPOWER OF STEAM ENGINES.—The rate at which steam does work upon the engine piston = *Indicated horsepower*,

I.Hp. = (Average total pressure on piston, lb.,
× distance moved by piston in ft. per min.) ÷ 33,000. . [1]

Also, I.Hp. = (Average effective pressure on piston, lb. per sq. in.
× piston displacement, cu. in. per min.) ÷ 396,000. . [2]

. . . . . . [3]

where P = mean indicated pressure, lb. per sq. in.; L = length of stroke, ft.; A = effec-

tive area of piston, sq. in., after deducting area of piston rod or tail rod; N = number of working strokes per minute.

Also,

$$\text{I.Hp.} = \frac{\text{M.E.P., lb. per sq. in.} \times \text{piston area, sq. in.} \times \text{piston speed, ft. per min.}}{33{,}000}. \qquad [4]$$

The mean effective pressure is the average pressure shown by the indicator diagram (see Fig. 17) to be acting on the piston. The *indicated horsepower* is the power actually given up by the steam. Friction causes some loss, hence the *brake horsepower*, or the power available at the engine crank-shaft, is less than the indicated horsepower. The ratio of brake horsepower to indicated horsepower is called the *mechanical efficiency* of the engine. (For values, see Table 8.)

MEAN EFFECTIVE PRESSURE depends on: initial steam pressure; relative length of steam admission (cut-off); back pressure; compression; clearance volume; area of ports; type of valve gear; tightness of valves and pistons, and degree of superheat.

Initial Steam Pressure is determined by the cost of fuel *vs.* the first cost, or the price one will pay for fuel saving. From that standpoint, high steam pressures pay good dividends in localities where fuel is expensive, but the state of the art, including methods of lubrication and packing, sets limitations to high-pressure equipment. Pressures in excess of the heretofore common standard of 250 lb. per sq. in. make the first-cost high, not only of the engine but also of boilers and piping. Few engine builders have drawings

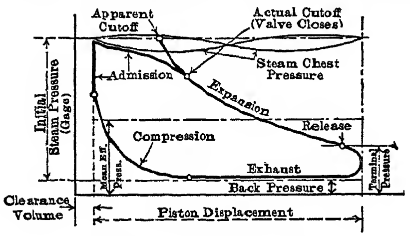


FIG. 17. Indicator Diagram

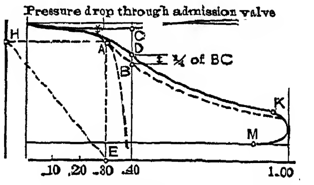


FIG. 18. Approximation of Indicator Diagram

and patterns of engines suitable for pressures in excess of 250 lb. per sq. in., and equipment for higher pressures must be designed and built specially. A number of installations for 450 lb. per sq. in. however, have been made, and in one plant triple-expansion engines operate regularly at 1400 lb. per sq. in.

Clearance volume depends on the type of valve gear and piston speed. High piston speed necessitates large valves, which means more clearance space.

Cut-off depends on the relation of economy to cost. Economy requires a short cut-off. Low engine cost (or high power from a given engine) requires a longer cut-off, with consequently higher steam consumption.

Back pressure depends on the possible use of exhaust steam from non-condensing engines, or on the temperature and quantity of cooling water available for condensing engines.

Compression must be high enough to insure quiet running of the engine, but must not be so high as to seriously decrease the engine economy. (See page 7–23.)

Area of ports depends upon piston speed, and grade of engine (whether expensive or cheap).

Valve gear depends on the use for which the engine is intended, on the rotative speed, and also on the grade of the engine.

TO FIND MEAN EFFECTIVE PRESSURE FOR GIVEN CONDITIONS.—Mean effective pressure (M.E.P.) from indicator cards of an existing engine is found by planimetering or by addition of ordinates.

$$\text{M.E.P.} = \frac{\text{area of card, sq. in.}}{\text{length of card, in.}} \times \text{spring scale.}$$

The spring scale is the pounds per square inch required to produce 1 inch of deflection, corrected by calibration.

For predicting the mean effective pressure of a new engine, there are three methods: 1. Comparison with indicator cards of existing similar engines. 2. Design of the probable indicator card and measurement of its area. 3. Estimation from the ideal indicator card, by means of a diagram factor.

The first is the best method and should be used wherever indicator cards from similar engines are available. (See Table 2.) The second method is desirable, but involves a large amount of numerical work. The third method is the most convenient, but it has led to great errors in calculation of the greatest possible horsepower, the error consisting in assuming that the apparent cut-off of the indicator card will be equal to the actual cut-off of the valve gear.

Design of Probable Indicator Card

ADMISSION.—The pressure in the cylinder is always less than boiler pressure. This pressure loss is due to four causes: 1. Friction in pipe line from boiler to engine. This part of total pressure drop can be computed from pipe friction formulas (see pages 5–20 to 5–23). Average values are from 2 to 6 lb. per sq. in., but with long pipe lines, may amount to 10 lb. per sq. in. In some instances it is unduly large because of the resistance to flow through throttle valves and quick closing safety stop valves. It is advisable to keep the steam velocity in pipes below 6000 ft. per min., except in very short lines or for very highly superheated steam; and in throttle valves it should not exceed 5000 ft. per min. The steam velocity through throttle valves may be calculated from the formula

$$\frac{\text{Piston area, sq. in.} \times \text{piston velocity at cut-off, ft. per min.}}{\text{Area through valves, sq. in.}} \qquad [5]$$

For pressure drop through valves, see p. 5–22.

2. Inertia of steam nearest engine. Pressure is required for its acceleration. Accurate calculation is almost impossible on account of superposition of impressed vibration and natural vibration. It seldom exceeds 4 lb. per sq. in.

Both (1) and (2) are reduced, and vibrations of the pipe line are eliminated by a large receiver-separator, with a comparatively small inlet, designed for a high or average velocity of steam flow (7000 to 8000 ft. per min.) and a larger outlet designed for low instantaneous velocity of flow to the engine (5000 to 6000 ft. per min.) The volume of the receiver-separator should be $1^3/_4$ to $2^1/_4$ times piston displacement, the higher value applying to high rotative speeds.

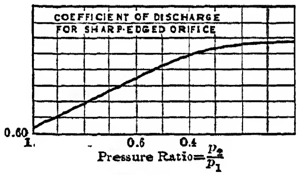


FIG. 19

3. Velocity head or pressure drop required to produce steam flow through the steam admission valve. It causes the apparent cut-off (Fig. 17) to be 6 to 15% earlier in the stroke than the actual cut-off of valve. The correct amount is difficult of exact calculation; increase of volume of the steam in the cylinder due to the motion of the piston, and condensation on the cylinder walls both tend to decrease the pressure in the cylinder, while the inflow of steam as well as expansion tend to keep up the pressure.

Referring to Fig. 18, a close approximation to the correct indicator diagram is obtained by calculating, for various piston positions, the pressure drop due to throttling by the steam admission valve, just as if the steam were inexpansible; merging the admission line thus found into the hyperbolic expansion line ($pv = k$) where the two have a common tangent, at A, and then correcting by raising the pressure at cut-off, *i.e.*, at actual valve closure, by one-fourth of the difference between initial steam pressure and the pressure found from the expansion curve from A; and starting the final expansion curve from D. In making this calculation it is advisable to tabulate as in the following example, which applies to a 10 × 14-in. slide-valve engine, 250 r.p.m., 40% valve cut-off.

1. Piston travel, % of stroke	10	20	25	30	35	40
2. Piston speed, ft. per sec.	9.1	12.2	13.2	14.1	14.5	
3. Area of valve opening, sq. in. = port width × valve opening	6.50	5.08	4.05	2.90	1.40	
4. Ideal steam velocity, ft. per sec. = (Item 2)/(Item 3) × piston area	110	189	256	382	802	
5. Coefficient of discharge (see Fig. 19)	0.62	0.62	0.63	0.64	0.72	
6. Corrected steam velocity, ft. per sec. = (Item 4) ÷ (Item 5)	178	305	410	600	1115	
7. Corresponding pressure drop, lb. per sq. in.	1.05	3.10	5.6	12.0	41.5	

Points representing piston positions should be taken close together near cut-off, and much farther apart at other positions. The angular velocity of the engine is assumed to

be practically constant, hence piston speed is represented by the ordinate of the semicircle. as AC, Fig. 20, for a mean between head end and crank end. Area of port uncovered by the valve at each piston position should be plotted as at AB in the upper part of figure. It is most conveniently found from a valve diagram; see page 7–33. Item 4, ideal steam velocity (as DE, Fig. 20) is (piston area ÷ area of valve opening) × piston velocity, ft. per sec. Coefficient of discharge for sharp-edged orifices varies with the ratio $p_2 \div p_1$, in this case (pressure in cylinder) ÷ (pressure in steam chest), as shown in Fig. 19. For low steam velocities, Item 6, use the expression $p' = c^2/9274\, v$, where p' = pressure drop, lb. per sq. in.; c = steam velocity, ft. per sec.; v = specific volume of the steam, cu. ft. per lb. (see Steam Tables, p. 5–04), for the pressure and temperature existing in the valve chest. For higher steam velocities, find the corresponding pressure drop from the Total-heat-Entropy chart, page 5–19.

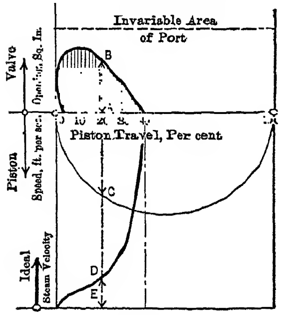


FIG. 20

A quick method of locating the point of tangency of the expansion hyperbola to the admission line is shown in Fig. 18. Draw vertical and horizontal lines from trial point A, to the intersections E and H with the lines of zero pressure and zero volume, and draw a line connecting E and H. If the tangent to the admission curve at A is parallel to EH, it also will be tangent to an hyperbola through A; hence the latter is the desired point. The horizontal axis must be the zero of absolute pressure, and the vertical axis the zero of cylinder volume, including clearance volume.

4. Friction through steam port. As a rule this is negligible, compared to pressure drop through the valve, except for long cut-off (approaching 50% of stroke). It can be computed from the formula for pressure drop through pipes and ducts. (See page 5–20). See also Bonin, *Zeits. f. ang. Math. u. Mech.*, v. 6, p. 491.

EXPANSION LINE.—The curve does not follow a simple law, but is complicated by partial re-evaporation (continuing as the pressure falls) of the film of water on the cylinder walls. It varies to some extent with all of the elements which affect cylinder condensation (see below under Steam Engine Economy). For all practical purposes, the curve pv = *constant* is sufficiently accurate for saturated or moderately superheated steam, while $pv^{1.06}$ = *constant* holds for steam superheated as much as 150° F. above saturation temperature. In duoflow engines, curves departing very far from these two indicate

In uniflow engines, condensation and subsequent re-evaporation are reduced, and $pv^{1.1}$ = *constant* is more nearly correct for saturated or slightly superheated steam. With steam superheated more than 150° F., there is very little cylinder condensation in either type of engine and the expansion curve can be taken directly from the Total-heat-Entropy chart (page 5–19) and the Pressure-Volume-Entropy charts or else $pv^{1.3}$ = *constant* can be used.

EXAMPLE.—To find the expansion curve of steam at 190 lb. per sq. in. absolute pressure, 300° F. superheat. The specific volume of steam at the initial temperature and pressure is 3.55 cu. ft. per lb. These conditions are represented on the Total-heat-Entropy chart by the intersection of the 300° F. superheat line and the 190 lb. pressure line. To find volume at, say, 110 lb. per sq. in. pressure, a line is dropped vertically from the initial point on the Total-heat-Entropy chart to 110 lb. pressure line, which gives the superheat, 205° F. By referring to the Pressure-Volume-Entropy chart, for that superheat and pressure, the specific volume is found to be 5.35 cu. ft. per lb. Then if the cylinder contains at cut-off a certain volume of steam, V_1 (including clearance volume), at 190 lb. pressure, that volume must expand to $5.35/3.55 = 1.507V_1$ when the pressure decreases to 110 lb. per sq. in. absolute.

EXHAUST TOE.—The curve KM (Fig. 18), represents free expansion (throttling) through the exhaust valve opening, but is complicated by evaporation of the film of water on cylinder walls. As long as the pressure in the cylinder is in excess of 1.5 × exhaust pressure, it may, with sufficient approximation, be found from the equation:

$$\left(\frac{P_r}{P}\right) = \log_{10}\left(\frac{c+x}{c+R}\right) , \quad \frac{0.892\, P_r V_r}{s}\int a\, dt$$

in which P = absolute pressure, lb. per sq. in., at any piston position x (fraction of stroke, see Fig. 21); P_r = pressure at release, lb. per sq. in., absolute; c = ratio (clearance volume ÷ displacement volume) and R = release ratio, both fractions of the stroke; P_r and V_r are, respectively, absolute pressure, lb. per sq. in. and specific volume, cu. ft. per lb., at point of release; s = stroke, ft.

The factor a is a function of the area of port opening, which varies with the piston position and is found by the method indicated in Fig. 22. The area of the exhaust valve opening (most conveniently obtained by means of the valve diagrams, Fig. 70-73), is plotted against time, zero of time being the point of release. This area can be determined by use of a planimeter or by averaging ordinates. The curve A (Fig. 22) of port openings thus obtained is distorted by multiplication of each value with the corresponding value of the expression, $1/(c + x)P$, where P = piston area, sq. in., if valve opening is taken in sq. in. Area under distorted curve a, for any piston position, is the value of $\int a\,dt$.

The numerical value of $(0.892\ P_rV_r)$ usually can be taken as 350, when time is taken in seconds. c, x and R are fractions of s.

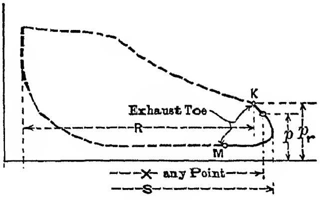


FIG. 21

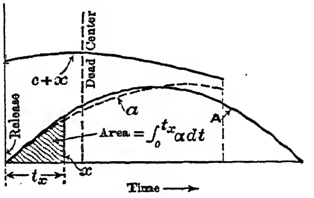


FIG. 22

This method assumes that all of the pressure loss is velocity-head, and none friction head. As the port opens widely, friction in the ducts becomes a noticeable proportion of the total pressure drop, and may be allowed for roughly, by somewhat decreasing the height of curve A near its maximum point.

For the section of curve in which the pressure in the cylinder drops from 1.5 × back pressure down to back pressure, no simple formula or graphical construction exists, but the curve easily can be sketched in from the shape of toe merging into the back-pressure curve (exhaust line), when the steam no longer flows out of the cylinder by its own free expansion, but is moving out under the displacing action of piston motion.

THE EXHAUST LINE is determined: 1. By back pressure in exhaust chest. 2. By pressure drop through exhaust valves and ports. Pressure in the exhaust chest depends upon the type of engine (compound, condensing, or non-condensing). For the receiver pressure of multi-stage engines, see the latter heading; for most suitable vacuum, see Steam Engine Economy and Condensation (page 7-22). The pressure in the cylinder must be higher than that in the exhaust chest, because of pressure drop due to flow of steam through exhaust valves and ports. Methods of finding excess pressure are identical with those described above under "Admission." Throttling through the valve when nearly closed toward beginning of compression causes compression to begin in the indicator card before the exhaust valve actually closes, the average point being 5 to 6% of stroke.

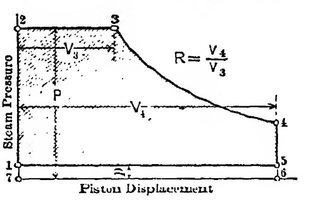


FIG. 23

THE COMPRESSION CURVE is not a simple one. As a rule, it first rises faster than the adiabatic, because the walls are hotter than exhaust steam, and then drops below the adiabatic at higher pressures, because compressed steam becomes hotter than the cylinder walls and begins to condense on them. For most cases $pv = constant$ is sufficiently accurate and is most commonly used; $pv^{1.1} = constant$ is more accurate under average conditions. See also Schüle, *Zeit.*, V. d. I., Nov. 24 and Dec. 8, 1906; Schneider, *Zeit.*, V. d. I., Feb. 9, 1907; Heinrich, *Zeit.*, V. d. I., 1912, page 1191.

MEAN EFFECTIVE PRESSURE FROM IDEAL CARD AND DIAGRAM FACTOR. —(Method 3). The ideal hyperbolic diagram without clearance and compression is represented by 1-2-3-4-5-1, in Fig. 23. The average ordinate, or mean effective pressure, lb. per sq. in., of this diagram is

$$\text{M.E.P.} = P\{(1 + \log_e R)/R\} - p, \qquad [7]$$

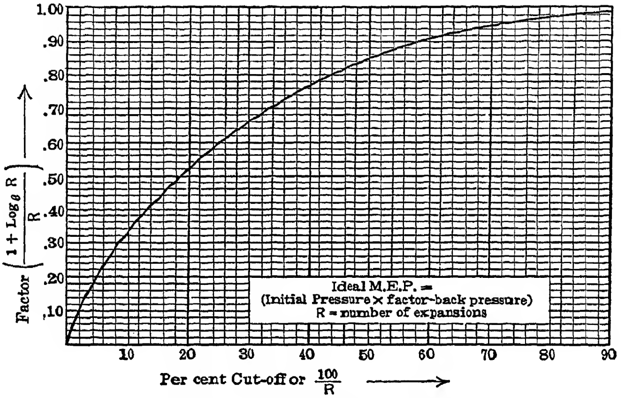
where P = initial steam pressure, and p = back pressure, both lb. per sq. in., abs.; $R = V_4/V_3$, which is called the ratio of expansion, or the number of expansions. For

this form of card it is the reciprocal of the cut-off ratio. The chart, Fig. 24, gives the ratio, (ideal M.E.P./P) for various values of R and for zero back pressure.

Example.—What is the ideal mean effective pressure of an engine with steam at 150 lb. per sq. in., gage, 25% cut-off, 24 in. vacuum, disregarding clearance and compression?

At 25% cut-off, the number of expansions is $1/0.25 = 4$. From Fig. 24, M.E.P./P for perfect vacuum = 0.597; then $0.597 \times (150 + 14.7) = 98.3$. Back pressure = 6 in. of mercury, or 2.95 lb. per sq. in. Hence, ideal M.E.P. = $98.3 - 2.9 = 95.4$ lb. per sq. in.

Fig. 25, shows the *ideal* mean effective pressure directly, for various expansion ratios and initial pressures, and for the two conditions of a back pressure of 15 lb. per sq. in., absolute, and of perfect vacuum. For the reasons explained above, the *actual* M.E.P.



. Factors for Calculating Mean Effective Pressure of Indicator Card

is less than that calculated by the formula and chart. The factor by which the ideal M.E.P. is multiplied to obtain the actual M.E.P. is called the *diagram factor*. Values which commonly apply (all referring to the ideal card without clearance or compression) are shown in Table 1.

This method of finding the M.E.P. is to be used for quick, approximate calculations only, and is not recommended for accurate work. Commercial mean effective pressures and overload capacities for different types of engines are shown in Tables 2 and 3. See also Foster, Manchester Association of Engineers, Jan. 23, 1915.

Table 1.—Diagram Factors

Average values for usual operating conditions, referred to ideal unmodified indicator diagram without clearance or compression (Fig. 23).

Type of Engine	Diagram Factor at Rated Load	Diagram Factor at Maximum Overload
Power Engines and Mill Engines: High Speed		
Single-valve engine, small size, simple	0.80	0.70
Piston-valve engine, simple	.82	.74
Piston-valve engine, compound	.70	...
Automatic four-valve engine, non-releasing valve gear, simple	.86	.82
Automatic four-valve engine, non-releasing valve gear, compound	.74	...
Releasing gear engine, simple	.90	.88
Releasing gear engine, compound	.76	.74
Releasing gear engine, triple-expansion	.70	.68
Uniflow poppet-valve engine, condensing	.78	.75
Uniflow, non-condensing, with large clearance	.62	.60
Uniflow, non-condensing with small clearance and auxiliary exhaust valv	.76	.75
Pumping Engines: Slow Speed		
Releasing gear compound, without jackets	.82	.81
Releasing gear compound, with jackets	.93	.92
Releasing gear triple, without jackets	.73	.72
Releasing gear triple, with jackets and reheaters	.85	.84

The factors are slightly less for non-condensing than for condensing engines of given type.

Influences which *reduce* diagram factors are: throttling due to small valve openings and steam passages or to very short cut-off; inter-related valve movements (single valve), which lengthen compression for early cut-off; late opening of exhaust valve; high piston speed; use of superheated steam.

Influences which *increase* diagram factor are jacketing and use of reheaters.

The *modified* ideal diagram, including clearance and compression, has, in general, an area somewhat less than that of the unmodified diagram, in approximately the following ratios for usual conditions: Releasing gear engine, 0.98; piston-valve engine, 0.96;

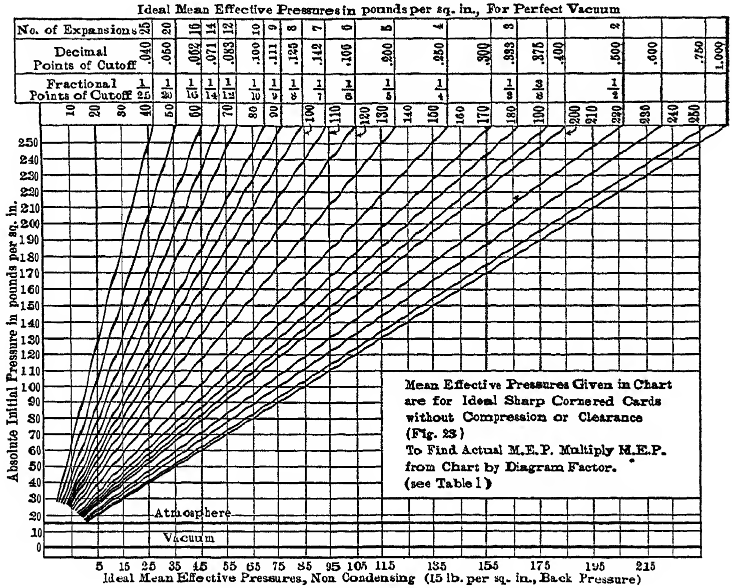


FIG. 25. Ideal Mean Effective Pressure Chart

Table 2.—Commercial Mean Indicated Pressures on Which Engine Ratings Are Based

Initial Steam Pressure, lb. per sq. in., gage	100	125	150	175	200
Engine Type and Operation	Rated Mean Effective Pressure, lb. per sq. in.				
Simple Engines					
Single-valve engine, condensing	45	49	53	57	62
Single-valve engine, non-condensing	51	58	64	71	77
Four-valve engine, condensing	42	46	50	54	58
Four-valve engine, non-condensing	48	54	60	66	72
Uniflow engine, condensing	35	42	48	52	55
Uniflow, non-condensing large clearance type	32	38	44	49	53
Uniflow, non-condensing, small clearance with auxiliary exhaust valves		44	51	56	59
Multi-stage Engines					
Compound condensing	25	28	30	32	34
Compound non-condensing	30	35	40	45	50
Triple-expansion, condensing		20	21	22	24

NOTE.—All engines for pressures above 200 lb. per sq. in. are special.

uniflow condensing, 0.91; uniflow non-condensing, large clearance type, 0.75, small clearance type, 0.95. Diagram factors applying to this form of diagram are the values of the table divided by the corresponding ratios.

EXAMPLE.—The diagram factor which should be used in calculating mean effective pressure from the modified diagram, for large clearance type of uniflow engines = 0.62/0.75 = 0.83.

Mean effective pressures in multi-stage engines are those referred to the low-pressure cylinder.

Most Economical Mean Effective Pressure is usually 65% to 75% of *rated* mean effective pressure (65% to 70% non-condensing uniflow, 70% to 75% in condensing uniflow).

The figures given in the table hold good irrespective of superheat, effect of which is discussed under Steam Engine Economy (p. 7–22).

Usual cut-off at rated load, for power engines under average operating conditions will range about as follows: *Simple engines*; four-valve engine, condensing, 17%, non-condensing, 25%; single-valve engine, condensing, 20%, non-condensing, 30%; uniflow, condensing, 15%, non-condensing, 20%. Compound engine, condensing, 33%, non-condensing, 40%. Triple-expansion, condensing, 35%. Cut-off refers to high-pressure cylinder and is the point of actual valve closing. Rolling-mill engines and blowing engines often work with longer cut-off than above.

CLEARANCE SPACE (as distinguished from linear clearance) includes all volume enclosed between piston and valves at one end of cylinder, when piston is at dead center at that end. Clearance is kept small as possible for the sake of steam economy, except in non-condensing uniflow engines in which it must be large (unless auxiliary exhaust valves are used) so that compression pressure will not rise above initial steam pressure. Increasing the ratio of cylinder diameter to stroke means larger clearance in percent of piston displacement. Low piston speeds permit use of small ports, and clearance can be reduced accordingly, while very high speeds mean large ports and valves and consequently large clearances (see also pages 7–13 and 7–29). See Table 4.

Table 3.—Overload Factors

The rated mean effective pressures of Table 2 are to be multiplied by the factors in this table in order to find the *maximum* M.E.P. which can be obtained.

Simple Engines	Initial Steam Pressure, lb. per sq. in.	Overload Factors: Condensing Operation	Overload Factors: Non-condensing Operation
Automatic engine; or single-eccentric releasing gear engine	100	1.86	1.45
	125	2.05	1.60
	150	2.25	1.75
	175	2.46	1.90
	200	2.67	2.06
Double-eccentric releasing gear engine	100	2.18	1.70
	125	2.43	1.87
	150	2.65	2.02
	175	2.84	2.15
Uniflow engine	100	1.63	1.30
	125	1.70	1.36 Large
	150	1.76	1.41 clearance
	175	1.81	1.44 type
	200	1.87	1.48

Compound Engines

Cylinder Ratio......	....	3	3 1/2	4	4 1/2	5	2	2 1/2	3	3 1/2
Automatic engine.....	100	1.48	1.34	1.20	1.12	1.02	1.25	1.07	......	.....
	125	1.59	1.47	1.32	1.22	1.12	1.39	1.20	1.02	.....
	150	1.75	1.62	1.45	1.35	1.25	1.46	1.27	1.10	1.0
	200	2.21	1.97	1.79	1.66	1.55	1.60	1.42	1.27	1.17
Double-eccentric releasing gear engine	100	1.88	1.70	1.58	1.46	1.34	1.46	1.27	1.10	1.0
	125	2.05	1.86	1.71	1.59	1.48	1.81	1.54	1.37	1.21
	150	2.26	2.06	1.88	1.75	1.63	1.91	1.63	1.46	1.31
	175	2.49	2.28	2.07	1.92	1.79	2.00	1.71	1.54	1.40

Triple Expansion

Releasing gear engine, with cylinder ratios shown in Table 5	150	1.30
	175	1.44
	200	1.60

MODIFIED HYPERBOLIC DIAGRAM.—Fig. 26 shows the sharp-cornered diagram 1–2–3–4–5–6–1, which includes the influence of clearance and compression, and hence more closely approximates the actual indicator card than does Fig. 23. The mean effective pressure is:

$$P\,(1 + c - cR) + \frac{P}{R}(1 + c)\log_e R - p\,(1 + c - cr) - pcr\log_e r. \quad [8]$$

where P = initial steam pressure and p = back pressure, both in lb. per sq. in., absolute; R = true ratio of expansion (*not* the reciprocal of the cut-off ratio, for this form of card), or $R = V_4/V_3$; r = compression ratio $= V_6/V_1 = p_1/p$; c = ratio of clearance $= V_1/(V_4 - V_1)$. V_1, V_3, V_6, etc., represent the volumes at the corresponding points on the card, Fig. 26.

For the actual engine with clearance, the *true* number of expansions $R = V_4/V_3 = P/P_4$, Fig. 26; whereas *apparent* number of expansions,

$$R_a = \text{reciprocal of cut-off ratio} = (V_4 - V_1)/(V_3 - V_1), \quad . \quad . \quad . \quad [9]$$

$$R = \frac{(c + 1)}{(cR_a + 1)} R_a. \quad . \quad . \quad . \quad . \quad . \quad . \quad . \quad . \quad . \quad [10]$$

The true number of expansions is always less than the apparent number of expansions.

Table 4 shows average clearance volumes for various types of engines in percent of piston displacement.

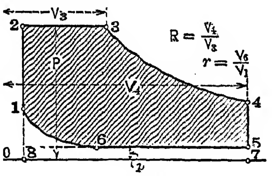


FIG. 26

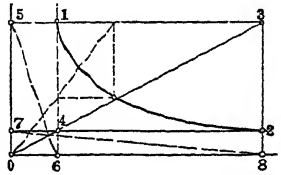


FIG. 27

LENGTH OF COMPRESSION AND FINAL PRESSURE OF COMPRESSION, p_1, depend on the clearance and the speed of the engine (see Table 4 and p. 7–14). For duoflow engines non-condensing, pressure p_1 (gage) varies from 70% *of initial gage pressure* in high-speed engines, down to 15% *of initial* gage *pressure* in low-speed engines; in condensing engines, p_1 seldom exceeds 15 lb. per sq. in. gage, unless the vacuum is poor or the clearance extremely small. For uniflow engines, the pressure p_1 is about 10 to 20 lb. per sq. in. less than the initial steam pressure in non-condensing, and 20 to 30 lb. per sq. in. less in condensing engines. Compression extends over 90% to 94% of the stroke in all uniflow engines, except in the non-condensing type with auxiliary exhaust valves, in which it extends over about 35% of the stroke.

QUICK CONSTRUCTION OF THE EXPANSION CURVE.—In Fig. 27 let 1 be the starting point; draw through it the vertical 1–6 and horizontal 1–3; to locate any other point, 2, draw vertical 2–3; through its intersection at 3 with the horizontal 1–3, draw diagonal to origin 0; through point 4 in which diagonal cuts vertical 1–6, draw horizontal

Table 4.—Clearance Volume of Steam Engines

Type of Valve	Percent of Piston Displacement: Average value for 800 ft. per min. piston speed. Stroke = cylinder diameter × 1.5	Percent of Piston Displacement: Range for varying conditions
Flat slide valve at side of cylinder................	9	5 to 10
Piston valve, usual design, at side of the cylinder..	12	7 to 17
Rocking valve (Corliss or 4-valve automatic engine)	5	3 to 8
Poppet inlet and exhaust valves in cylinder heads...	4	3 to 6
Poppet inlet and exhaust valves at side of cylinder..	9	7 to 12
Poppet inlet valves, uniflow condensing engine.....	2	1 1/2 to 2 1/2
Poppet inlet and auxiliary exhaust valves, uniflow non-condensing engine.....................	3	2 1/2 to 3 1/2
Poppet inlet valves, *without auxiliary exhaust*, uniflow non-condensing engine..................	Depends upon initial steam pressure, smaller for high than for low steam pressures	10 to 17
Valves in cylinder heads, compound pumping engines, slow speed...........................	....	3/4 to 3 1/2

4–2 to intersection with vertical 2–3, thus locating point 2. The area of rectangle 1–6–0–5 equals the area of rectangle 2–8–0–7; also the tangent to curve at 1 is parallel to diagonal 5–6, and the tangent at 2 is parallel to 7–8. The curve also can be constructed by the intercept method, as in Fig. 28, in which 1 is the starting point, and 1–3 = 2–4, 1–6 = 5–7, etc.

Construction of $pv^{1.06}$ = constant, is best accomplished by use of the log log slide rule. In the absence of such an instrument the following method may be used:

$$pv^{1.06} = pv \times v^{0.06} = pv \sqrt[16]{v} = constant. \quad . \quad . \quad . \quad . \quad . \quad . \quad . \quad [11]$$

The 16th root is found by extracting the square root four times. All other expansion

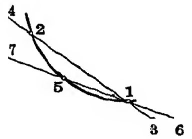


Fig. 28

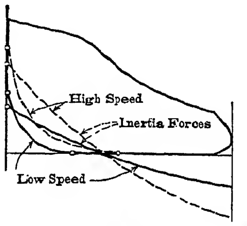


Fig. 29.—Effect of Speed on Compression

curves should be constructed by use of the Total-heat-Entropy chart or by use of the log log slide rule.

PISTON SPEED AND ROTATIVE SPEED.—The term piston speed commonly is understood to mean *average* piston speed, or 2 × stroke in ft. × r.p.m. High piston speeds are desirable for reducing floor space, weight, and cost of engine; high rotative speeds are desirable for engines direct-coupled to generators because size, weight, and cost of electric generators thereby are reduced. High piston speed, obtained by lengthening the stroke, rather than by increasing the revolutions per minute, does not cheapen the engine.

Economical piston speeds range between 500 and 1000 ft. per min., varying with the size of engine. (See Figs. 30 and 31.) Piston speeds up to 2000 ft. per min. are possible, but not economical for the following reasons: 1. High piston speeds cause excessive pressure drop through inlet and exhaust valves and ports. This either means increased steam consumption per horsepower-hour, or else large valves and ports, with resulting

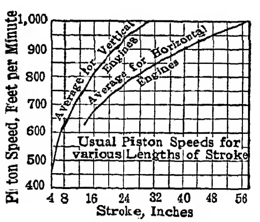


Fig. 30

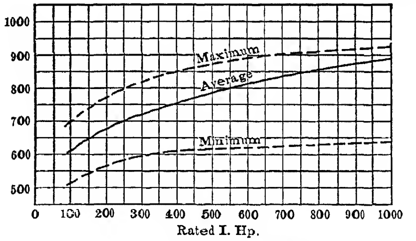


Fig. 31.—Piston Speeds of Engines

large clearances and waste spaces and, consequently, greater steam consumption (see Steam Engine Economy).

2. High rotative speeds produce great inertia forces. To prevent pounding in the main pins, the force must change slowly from tension to compression in the connecting-rod. This requires that change of direction of force shall occur before the steam valve opens. As a result, with high rotative speeds, longer compression is needed for a given clearance. (See Fig. 29.) Longer compression means somewhat greater steam consumption (see Steam Engine Economy). On the other hand, very low piston speeds cause increased cylinder condensation, because there is excessive time for such condensation. The two conflicting influences cause piston speeds to range as above given, and as shown in Figs. 30 and 31.

Valve gears impose limitations on rotative speeds. Releasing gears can be operated at speeds up to 150 r.p.m., but they offer no advantages above 100 r.p.m., and introduce difficulties above 120 r.p.m.

Direct coupling to compressors for air, gas, or ammonia limits the piston speed of engines.

References.—Davidson, *Power*, Feb. 9, 1909. Polster, *Mitteil. u. Forsch.*, Heft 172–173, 1915. Ripper, Steam Engine Theory and Practice, 1914. Radinger, Dampfmaschinen mit höher Kolbengeschwindigkeit.

Figs. 30 to 35 show piston speeds, weight of engines per horsepower, floor space, volume of foundation required, and (weight × diameter[2]) of flywheel, for various sizes

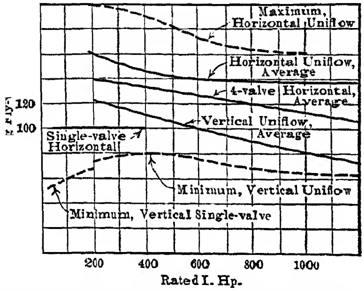


Fig. 32.—Weight of Engines per Horsepower, Including Fly-wheel but not Generator; Latter Weighs about 12 lb. per kw.

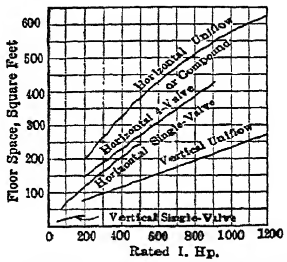


Fig. 33.—Floor Space Required by Engines (Area of rectangle enclosing engine; area of L-shaped top of foundation is about 2/3 as large.)

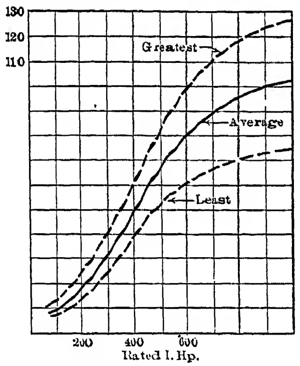


Fig. 34.—Volume of Engine Foundation Required

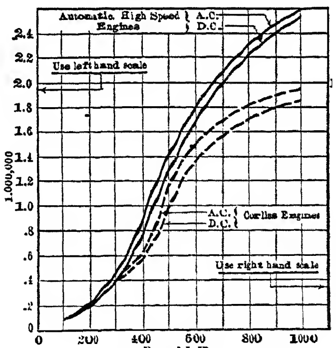


Fig. 35.—Weight of Fly-wheel Required

of engines. The floor space is the area of the rectangle enclosing engine and generator. These curves were plotted from data furnished by a number of builders of modern power engines, and show American practice in 1932.

Capacity of Compound Engines

The capacity of compound engines usually is figured by using mean effective pressure referred to the low-pressure cylinder, or *equivalent mean effective pressure.* The latter equals (M.E.P. of low-pressure cylinder + M.E.P. of high-pressure cylinder × ratio of high-pressure piston displacement to low-pressure piston displacement).

The best mean effective pressure is determined as a compromise between too great free expansion loss (high M.E.P.) and too great cylinder condensation loss (low M.E.P.). Commercial mean effective pressures are shown in Table 2.

The ratio of high-pressure piston displacement to low-pressure piston displacement is

influenced by various considerations, as: 1. Steam economy (meaning minimum heat interchange between cylinder walls and steam), would require that the expression (area of h.p. piston × steam temperature drop in h.p. cylinder + area of l.p. piston × steam temperature drop in l.p. cylinder) be made a minimum.

2. Equalization of forces in high-pressure and low-pressure rods and cranks (cross-compound only), would require that (h.p. piston area × pressure range in h.p. cylinder) and (l.p. piston area × pressure range in l. p. cylinder) be made equal.

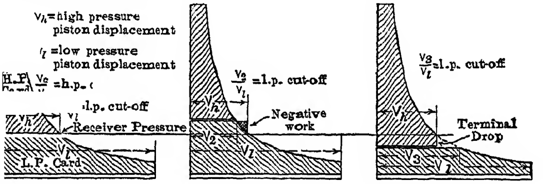
3. Equalization of work would require equality of work done per revolution in high-pressure and low-pressure cylinders.

4. Overload capacity would require the high-pressure cylinder to be of sufficient size to give ample overload capacity within the limits of cut-off in that cylinder.

Requirements (1) and (4) are contradictory. As a rule, a compromise is necessary in view of standard cylinder sizes being used.

Commercial cylinder ratios are shown in Table 5. For superheated steam somewhat lower ratios are used.

A rough rule for the size of non-condensing compound engine to give the same power as a simple engine of the same stroke and speed, is: High-pressure cylinder $0.75D$; low-pressure cylinder $1.3D$; where D = cylinder diameter of the equivalent simple engine. For condensing engines, high-pressure cylinder = $0.7D$; low-pressure cylinder = $1.4D$.



CORRECT L. P. CUT-OFF　　SHORT L.P. CUT-OFF　　LONG L. P. CUT-OFF

FIG. 36

RECEIVER PRESSURE can be varied by changing the low-pressure cut-off. The receiver is the space or container into which steam from the high-pressure cylinder passes before it enters the low-pressure cylinder. In many engines there is no separate receiver tank, the pipe connecting the high-pressure to low-pressure cylinder being made of sufficient diameter to give the required volume. In the Woolf type tandem-compound (now obsolete) the receiver was dispensed with entirely.

Fig. 36 is an ideal indicator diagram for an engine with a very large receiver (ideal case of uniform receiver presure) neglecting clearance and compression. For this ideal case, receiver pressure, $P_r = RP/n$, where P = initial steam pressure; R = cylinder ratio or V_l/V_h; n = number of expansions = V_l/V_c. Hence, neglecting clearance, P_r = initial steam pressure × cut-off (fraction of stroke in high-pressure cylinder). Pressures are absolute.

Table 5.—Commercial Cylinder Ratios of Multi-stage Engines

Cylinder ratios represent compromise between steam economy and good overload capacity.

	Condensing Engines					Non-condensing Engines				
Initial Steam Pressure, lb. per sq. in., gage	100	125	150	175	200	100	125	150	175	200
Type of Engine:										
Automatic compound for electric power generation, or blowing engines	3.5	3.9	4.3	4.7	5.1	2.3	2.6	2.9	3.3	3.7
Automatic compound for large overload capacity (mill engines)	3.0	3.3	3.6	4.0	4.4	2.0	2.2	2.5	2.8	3.1
Compound engine with releasing valve gear (rocking or poppet valves)	4.0	4.4	4.8	5.2	.....	2.6	2.9	3.2	3.6	
Triple-expansion engine: h.p. int. l.p.	....	1: 3.2: 7.0	1: 3.3: 7.7	1: 3.5: 8.4	1: 3.7: 9.0					

NOTE.—For higher pressures, cylinder ratios are not standardized, since engines for these pressures are built only to special order.

For the actual case, receiver pressure is less than that given by the above expression, since some *terminal drop* usually is allowed for the purpose of equalizing the work done by the two cylinders. Average receiver pressure may be found by constructing the indicator diagram, Figs. 39 and 40, if high-pressure and low-pressure cut-off are known. The following empirical expressions give approximate results, which may vary 10% from actual.

$$P_r = 0.58P \qquad [12], \qquad\qquad 0.9P \qquad [13],$$

in which P_r = receiver pressure, and P = initial steam pressure, both in lb. per sq. in., absolute; C_h and C_l = the cut-off in high- and low-pressure cylinders, respectively; R = ratio of low-pressure to high-pressure piston displacement; n = true number of expansions, considering clearance. Formula [13] may be used with the usual cylinder ratios and valve settings. Average values of receiver pressure may be taken at about 22 lb. gage for condensing, and 40 lb. gage for non-condensing engines with 150 lb. per sq. in. initial pressure (gage).

EFFECT OF CHANGING LOW-PRESSURE CUT-OFF.—If the high-pressure cut-off remains constant, lengthening the low-pressure cut-off reduces the work done by the low-pressure cylinder, and increases the work done by the high-pressure cylinder, because lengthening the low-pressure cut-off reduces the receiver pressure. (See Fig. 36.)

ACTUAL INDICATOR CARD.—Methods of constructing the ideal card and diagram-factors are similar to those used for simple engines. Features peculiar to the compound engine card are to be gained by study of the combined indicator card, in which both high-pressure and low-pressure diagrams are reduced to the same scale of pressure and volume. There are two methods of combining (also called "Rankinizing"), *viz.*, 1. By putting the clearance lines together as in Fig. 37. 2. By making the high-pressure and low-pressure

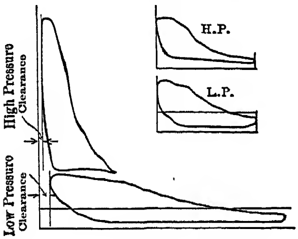


Fig. 37

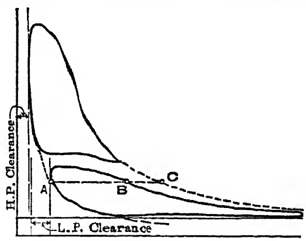


Fig. 38

Methods of Combining Indicator Cards

compression lines extensions of each other, as in Fig. 38. As a rule AB is only 80% to 85% of AC, thus showing the effect of cylinder condensation in the low-pressure cylinder. AB can be made equal to AC by reheating receivers and jackets on the low-pressure cylinder, but this is seldom done. (See Steam Engine Economy.)

Volume of Receiver.—The shape of the card is affected by the size of the receiver. As a rule, its volume should be $1\,^1/_2$ times the high-pressure piston displacement. It can be less for tandem than for cross-compound engines. Small receiver volume increases the temperature range in each cylinder, thereby reducing the economy. A very large receiver volume interferes with the regulation of the engine, if the high-pressure valve gear only is controlled by the governor.

Figs. 39 and 40 show a method of constructing the ideal combined indicator card for cross-compound engines and tandem-compound engines, respectively, and illustrate the effect of small receiver capacity. In these figures, abscissas represent volumes; ordinates, in the upper part of the figures, are portions of one revolution, and steam pressures in the lower part; V_h and V_l are high-pressure and low-pressure piston displacements, and C_h and C_l are high-pressure and low-pressure clearance volumes, respectively, while R is the volume of the receiver. The helices at the top show simultaneous positions of the high-pressure and low-pressure pistons. In cross-compound engines, the low-pressure crank is usually 90° (sometimes as much as 105°) ahead of the high-pressure crank in the direction of rotation. The effect of finite connecting-rod length is not shown in the diagrams, but can be taken care of by drawing arcs as shown under Valve Diagrams, Figs. 70 and 73. The variation introduced is unimportant, unless the cranks are 180° apart. At the right in Fig. 39, the high-pressure diagram is shown in dotted lines above

the low-pressure diagram. In the following discussion p_1, p_2, V_1, V_2, etc., refer to pressures and volumes at the points 1, 2, etc., on the diagram.

To lay out the diagram of Fig. 39, which is for a cross-compound engine, draw from the point of high-pressure cut-off, with point 18 as origin, the hyperbola 13–9 (volume 1–15 = volume 16–13) and locate point of low-pressure cut-off, 8, which is shown in this case as 50% of the stroke. This gives also pressure p_8 of the receiver into which the high-pressure cylinder later discharges its steam at 2. From the known point of beginning of low-pressure compression, draw hyperbola 11–12, which determines the pressure p_{12}, of the clearance space into which steam from the receiver at pressure p_4 flows at low-pressure admission, point 7.

Having drawn expansion hyperbola 1–2 with point 17 as origin, find pressure at 3 from the equation $p_2(V_2 + C_h) + p_8R = p_3(V_2 + C_h + R)$. Both V_2 and V_3 are the same as V_h. From 3 to 4, the high-pressure cylinder is exhausting into receiver, and $p_3(V_3 + C_h + R) = p_4(V_4 + C_h + R)$. At 4, steam from the receiver is admitted to the low-pressure cylinder, in which the clearance steam is at pressure p_{12}. Find the pressure at 5 from $p_4(V_4 + C_h + R) + p_{12}\, C_l = p_5(V_5 + C_h + R + C_l)$ and $p_5 = p_7$.

The line 5-6 of the high-pressure diagram corresponds to 7–14 on the low-pressure diagram, steam flowing into the receiver from the high-pressure cylinder, and out of it into the low-pressure cylinder, $p_5(V_5 + C_h + R + C_l) = p_6(V_6 + C_h + R + C_l + V_{14})$; also $p_6 = p_{14}$. At point 6 (determined from valve diagram), compression begins in the

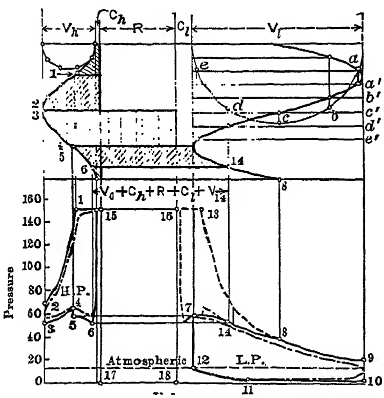


Fig. 39. Cross Compound Engines

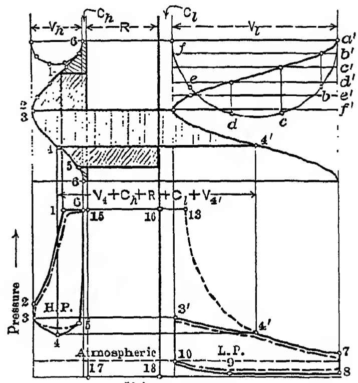


Fig. 40. Tandem Compound Engines

Methods of Constructing Ideal Combined Indicator Cards for Compound Engines

high-pressure cylinder. From 14 to 8 steam is expanding from the receiver into the low-pressure cylinder, and $p_{14}(R + C_l + V_{14}) = p_8(R + C_l + V_8)$. Volumes are represented by the horizontal lines in upper part of figures and may be scaled off directly from them for all points. The actual diagram will differ from the ideal diagram, due chiefly to the rounding of corners by throttling through valves and ports, and to cylinder condensation. The actual form is shown by the dot-dash lines.

Diagrams for tandem-compound engines can be constructed by the same methods. They have a form entirely different from those of the cross-compound, as shown by Fig. 40.

References.—Foster, Manchester Association of Engineers, Jan. 23, 1915; Ninde, Design and Construction of Heat Engines, 1920; Hütte, Vol. II, [illegible], 1920; Ripper, Steam Engine Theory and Practice, 1914; Unwin, Machine Design, Part 2; *Power*, Aug. 6, 1912, and Dec. 21, 1909; Low, Compound Engines.

3. TRIPLE-EXPANSION ENGINES

The field of the triple-expansion engine is very limited. It is essentially a constant-load machine, as the small size of the high-pressure cylinder does not allow much overload capacity.

In marine service, the geared steam turbine and the turbo-electric drive are competing with the triple-expansion engine, with the result that the principal use of the latter is

to-day, in water works pumping engines. The overall efficiency of the triple-expansion engined pump is considerably better than that of a turbo-driven centrifugal pump, but the first cost of the reciprocating set is much higher, with the result that relative first costs and fuel costs decide which type is to be used.

4. EXTRACTION OR BLEEDER-TYPE ENGINES

In plants with engines operating condensing, where steam at 5 to 15 lb. per sq. in. gage pressure is needed for heating or chemical processes, it may be obtained from the receiver of compound engines, or tapped from the engine cylinder of simple engines, through a check valve and port which is uncovered by the piston some distance before the end of the stroke. In uniflow non-condensing engines, steam can be extracted during the compression stroke through a separate valve in the piston. This has advantage of decreased compression, permitting the use of a smaller clearance volume. (*Zeit.*, V. d. I., Dec. 20, 1913; Stumpf, Uniflow Steam Engine, 1922.)

5. STEAM ENGINE ECONOMY

References.—Heck, Steam Engine and Turbine, 1911; Foster, Manchester Association of Engineers, Jan. 23, 1915; Heilmann, *Zeit.*, V. d. I., June 10, 1911, and Oct. 31, 1911; Barrus, Engine Tests; Stumpf, Uniflow Steam Engine, 1922; Gebhardt, Power Plant Engineering, 1917; Fernald and Orrok, Engineering of Power Plants, 1916.

The economy of steam engines is expressed in pounds of steam per indicated horsepower-hour; pounds of steam per brake horsepower-hour; British thermal units per indicated horsepower-hour; B.t.u. per brake horsepower-hour; or by these same values per kilowatt-hour; or by the Rankine Cycle Ratio.

STEAM CONSUMPTION GUARANTEES.—In practice, it is customary to base steam consumption guarantees on the performance of similar engines working under similar conditions of steam pressure, superheat and back pressure. If such data are not available, an analysis of the following type becomes necessary.

FACTORS DETERMINING EFFICIENCY.—The ideal cycle for both steam engines and turbines is the Rankine cycle, Fig. 41. See also Steam Cycles, p. 5–14. The highest useful steam pressure is determined by the cost of engine and boiler plant in relation to cost of fuel. Back pressure is determined by the use of exhaust steam for heating purposes or chemical processes, or in condensing operation, by the relation of the gain in steam economy by increasing vacuum, to the cost of vacuum (see Fig. 46 and pages 7–22 to 7–25).

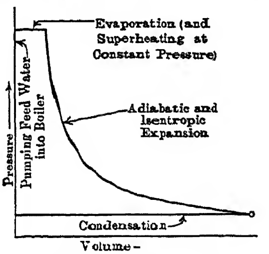


FIG. 41. Rankine Cycle

The steam consumption of a perfect engine following the Rankine Cycle, in lb. of steam per Hp.-hr. is (2546 ÷ B.t.u. available per lb. of steam). The divisor is found directly from the total-heat-entropy chart. The ratio

$$\frac{\text{Steam per brake-Hp.-hr. consumed by perfect Rankine Cycle engine}}{\text{Steam per brake-Hp.-hr. consumed by actual engine}}$$

is called the Rankine Cycle Ratio, Cylinder Efficiency or Relative Efficiency. The Rankine Cycle Ratio is depressed below 100% by the following factors: 1. Friction of steam in passages and ports; 2. Friction of moving parts of engine; 3. Cylinder condensation; 4. Leakage; 5. Losses by free expansion. Items 3 and 5 are the most influential factors. For engines with back pressures of 5 lb. per sq. in. or more above atmospheric pressure, expansion can be carried to back-pressure, and yet the loss by cylinder condensation is small. While steam consumption per Hp.-hr. is high, and the thermal efficiency, therefore, low, the Rankine cycle ratio is very high (about 90% if referred to indicated Hp.-hr.).

The Rankine Cycle Ratio of simple condensing engines is very low when saturated steam is used, because of excessive cylinder condensation, if expansion is carried to or near back-pressure; this also means a very large and expensive engine. Consequently, considerable free expansion loss is allowed. The proper size of cylinder is a compromise between excessive condensation loss (large engine for a given power) and excessive free expansion loss (small engine for a given power).

AVERAGE RANKINE CYCLE RATIOS.—Table 6 gives typical Rankine Cycle Ratios based on indicated Hp.; steam consumption per I.Hp.-hr., and overall thermal efficiencies based on brake Hp., for various types of engines working at or near full load, and with various steam and back pressures. Table 7 gives results of economy tests of various types of steam engines. The *water rate obtained under test conditions cannot be duplicated under ordinary continuous operating conditions* and Table 6 rather than Table 7 should be used for estimating steam consumption for every-day operation.

EXAMPLE.—To find the Rankine Cycle Ratio of uniflow engine, condensing, of 1200 Hp., working with 150 lb. saturated steam. From Table 6 the efficiency of a 500-Hp. engine under the same conditions = 64%. Substituting in formula to find the value of A, $64 = A + 11\sqrt[6]{500}$. $A = 33$. Efficiency of 1200-Hp. engine $= 33 + 11\sqrt[6]{1200} = 68\%$. The sixth root may be found by taking the cube root of the square root or by use of the log log slide rule.

FACTORS INFLUENCING CYLINDER CONDENSATION.—1. Temperature range of steam cycle in cylinder. 2. Area of surface on which steam can condense, in relation to volume of steam admitted. 3. Physical condition of entering steam (wet, dry satu-

Table 6.—Rankine Cycle Ratio and Steam Consumption

Average values obtained under ordinary operating conditions. Show tests give higher values.

Type and Approximate Size of Engine	Steam Pressure (Gage)	Steam Conditions	Vacuum *	Rankine Cycle Ratio, % Based on I.Hp.	Pounds of Steam Used per I.Hp. Hour	Thermal Efficiency, % Based on Brake Hp.
Slide valve, simple, 50 Hp.	125	Saturated	N.C.	39.0	40.0	5.5
Piston valve, simple (using same port for inlet and exhaust)	125	Saturated	N.C.	58.0	27.0	8.5
	125	150° superheat	N.C.	62.0	22.5	9.3
Piston valve, compound, 300 Hp.	150	Saturated	N.C.	66.0	22.0	10.3
	150	150° superheat	N.C.	70.0	18.5	11.2
	150	Saturated	26 in.	50.0	17.5	11.8
	150	150° superheat	26 in.	58.0	13.9	14.0
Four-valve, non-releasing compound, 300 to 500 Hp.	150	Saturated	N.C.	74.0	19.5	11.6
	200	"	N.C.	75.0	17.2	12.9
	150	"	26 in.	61.0	14.5	14.3
	200	"	26 in.	63.0	13.1	15.8
Corliss, simple, 300 Hp.	125	Saturated	N.C.	69.0	22.5	10.0
	125	"	26 in.	50.0	18.5	11.3
Corliss, compound, 500 Hp.	150	Saturated	N.C.	76.0	19.0	11.9
	150	"	26 in.	64.0	13.7	15.1
	150	50° superheat	26 in.	66.0	13.2	15.5
Triple expan. marine engine	200	Saturated	26 in.	64.0	13.2	15.1
" " power engine	175	150° superheat	26 in.	75.0	10.4	19.1
" " pumping engine, 500 Hp.	175	150° "	26 in.	81.0	9.7	20.5
Uniflow (simple), 500 to 1000 Hp.	125	Saturated	N.C.	70	21.7	10.2
	150	"	N.C.	71	20.3	11.1
	200	"	N.C.	74	17.5	12.7
	125	150° superheat	N.C.	74	18.9	11.2
	150	150° "	N.C.	75	17.5	11.9
	200	150° "	N.C.	77	16.0	13.2
	125	Saturated	26 in.	63	14.5	14.3
	150	"	26 in.	64	13.7	15.0
	200	"	27 in.	64	12.7	16.2
	125	150° superheat	26 in.	67	12.5	15.6
	150	150° "	26 in.	68	11.8	16.4
	200	150° "	27 in.	68	11.0	17.7
	250	150° "	26 in.	69	10.5	18.2
	150	Saturated	5 lb.†	87		

* N.C. = Non-condensing. † Back pressure (gage).

NOTE.—For larger sizes and for higher superheats, the values of Rankine Cycle Ratio become greater than those given in the table. Variation of Rankine Cycle Ratio with size between the limits of 100 Hp. and 1500 Hp. is shown with moderate accuracy by the empirical formula:

$$E = A + 11\sqrt[6]{\text{I.Hp.}}, \text{ for condensing engines, and } E = A + 8\sqrt[6]{\text{I.Hp.}},$$

for non-condensing engines; value of factor A changes with the type of engine.

rated, or superheated). 4. Time during which steam can condense. 5. Space relation between entering and discharging steam (wiping heat on and off the walls of cylinder and ports). 6. Length of time of compression during which steam temperature exceeds temperature of cylinder walls. 7. Temperature of cylinder walls (steam jacketing).

All the above factors are at present beyond the range of any rational formula. It is necessary, then, to pile up a great mass of test data, or else to use an empirical formula, such as Heck's, given below, which, by a number of coefficients, furnishes sufficiently accurate results for most practical purposes.

References.—Heck, Steam Engine and Turbine; Callendar and Nicholson, *Proc.*, Inst. Civ. Eng., 1897, Vol. 131; Clayton, University of Ill. Bulletin No. 58; Duchesne, *Revue de Mécanique*, Vol. 19, 1906; Barrus, Engine Tests; Nusselt and Nagel, Forsch. Arb. V. d. I., Heft 300.

DETERMINATION OF STEAM CONSUMPTION FROM INDICATOR CARDS.—The initial steam consumption per indicated horsepower-hour may be determined from an actual indicator card, or from a probable indicator card as developed on p. 7–07, by means of the formula

$$W = \frac{\text{density at } E \times e}{\text{M.E.P.} \times L}, \quad . \quad . \quad . \quad [14]$$

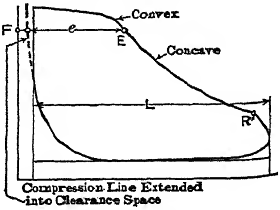


Fig. 42

where W = ideal steam consumption, lb. per I.Hp.-hr.; M.E.P. = mean effective pressure, lb. per sq. in.; L = length of stroke, ft. The value of e and position of E are given in Fig. 42. Point E is located near the cut-off a little below the inflection point from convex to concave. The quality of steam at E is unknown and therefore, it is customary to assume dry saturated steam at E, and take care of the variation of steam density by an initial condensation formula. (See below.)

FORMULA FOR INITIAL CONDENSATION.—The following formula was devised by R. C. Heck (Steam Engine and Turbine, paragraph 22),

$$m = (C/\sqrt[3]{N}) \times (\sqrt{sT/pe'}), \quad . \quad . \quad . \quad . \quad . \quad . \quad . \quad . \quad [15]$$

in which m = fraction of initial condensation, or ratio of missing quantity to (ideal steam + missing quantity); N = rev. per min.; s = area of nominal cylinder surface, sq. ft., divided by the piston displacement, cu. ft., or

$$s = (24/D) \times \{(D/L) + 2\}, \quad . \quad . \quad . \quad . \quad . \quad . \quad . \quad . \quad [16]$$

where D = diameter, in., and L = stroke of piston, in.; hence s is inversely a measure of the cylinder size; T is an empirical function of the temperature range in the cylinder, and is found from Fig. 43, by reading off T_1, corresponding to the highest pressure in the cylinder, and T_2 corresponding to the lowest pressure; then $T = T_1 - T_2$; p = absolute pressure at point E, Fig. 42, just at cut-off; e' is the ratio

$$\left(\frac{\text{cut-off, \%} + \text{clearance, \%}}{100\%}\right).$$

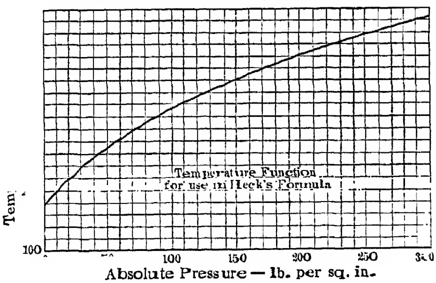


Fig. 43

In Fig. 42 it is represented by $FE'L$. C is a coefficient which takes care, in a measure, of the space relation between ingoing and outgoing steam, and hence must be somewhat with the type of See Figs. 14 to 16. A good average for C, for 4-valve using saturated steam, is 0.27. For single-valve engines, C is about 0.33. the speed exceeds 125 rev. per min., instead of $C/\sqrt[3]{N}$ use $^2/_3\,(C/\sqrt[4]{N})$. For effect of changing clearance volume, see p. 7–23.

The formula does not apply to low-pressure cylinders of compound engines, unless the water is separated from the steam in a large receiver and drained out, nor to en using superheated steam or steam jackets. It refers to engines which show inapprecia steam leakage when tested standing still.

To extend the range of usefulness of Heck's formula to the uniflow engine and to

use of superheated steam, the coefficient C must be modified. For duoflow engines, the value $C - \sqrt{\dfrac{\text{Superheat, deg. F.}}{4500}}$ may be substituted, and for uniflow engines, the value $0.45C - \sqrt{\dfrac{\text{Superheat, deg. F.}}{10{,}000}}$.

This modification necessarily must be only approximate, for the reason that the amount of cylinder condensation depends somewhat upon the point of cut-off. Condensation may for a given degree of superheat, be wholly absent at long cut-off and still be present at short cut-off.

EFFECT OF CHANGE OF INITIAL STEAM PRESSURE on engine economy is more important for high than for low ratios of expansion. For power engines operating with saturated steam, and with the usual expansion ratios, the relative steam consumption at various initial pressures is shown in Fig. 44, for both condensing and non-condensing operation.

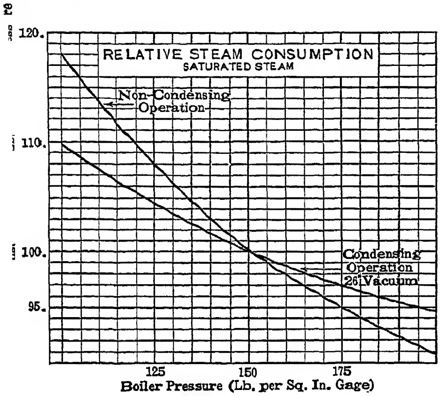


Fig. 44.

EFFECT OF SUPERHEAT on engine economy is double: (1) It reduces the weight of steam per unit volume; (2) It reduces cylinder condensation. If superheat is high enough, the missing quantity as calculated by the above given method, becomes negative, because the weight of steam admitted was computed upon basis of saturated steam. The worse the engine with regard to cylinder condensation, the more beneficial is the effect of superheat. Engines operating with high superheat show a flat load-steam-consumption curve (similar to that of the uniflow engine, Fig. 51). A rough-and-ready rule for single stage expansion engines is that each 10° F. of superheat, reduces steam consumption 1%. Fig. 45 shows the percentage saving in steam consumption due to superheat in various types of engines. Difficulties with packing and lubrication limit the total steam temperature, at present (1935) to 750°, or at most, to 800° F.

References.—Heilmann, *Zeit.*, V. d. I., June 10 and Oct. 31, 1911; Barrus, Engine Tests; Hanszel, *Zeit.*, V. d. I., Jan. 13, 1912; Doederlein, *Zeit.*, V. d. I., Oct. 7, and 21, 1911; Goss, University of Ill., Bulletin No. 57, April, 1912; *Blast Furnace and Steel Plant*, June, 1920.

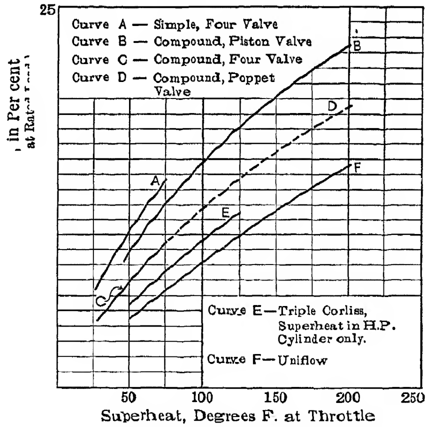


Fig. 45. Saving Effected by Superheat in Percent of Saturated Steam Consumption

EFFECT OF BACK PRESSURE OR VACUUM.—The harmful effect of back pressure is best seen from the lowered thermal efficiency of the Rankine cycle. It is not harmful if all of the exhaust steam is required at the high back pressure for heating or chemical processes. The engine then acts simply as a reducing valve and the power is a by-product.

The effect of vacuum varies with the type of engine. It is least effective in simple engines using saturated steam, with long cut-off, and most effective in multi-stage engines using superheated steam with short cut-off, provided compression need not be high. To make high vacuum effective, release must occur

comparatively early in the stroke (at R, in Fig. 42) and valve passages must be large. Under average conditions each added inch of vacuum reduces the steam consumption 1%. This percentage does not grow with increasing vacuum, but rather tends to decrease. In this respect the engine differs materially from the turbine. Fig. 46 shows the saving due to vacuum, for various types of engines.

References.—Heilmann, *Zeit.*, V. d. I., June 10, 1911; Barrus, Engine Tests; Fothergill, *Power*, Jan. 16, 1912.

CLEARANCE VOLUME is harmful to economy for two reasons: 1. Because it directly increases the surface on which steam condenses; 2. Because it increases free expansion loss, since the true expansion ratio for a given piston displacement at cut-off is less than if no clearance space were present. The effect of surface of clearance space on condensation cannot be expressed in a simple manner. A rough approximation may be gained by increasing C in Heck's formula (see above), in the proportion of

$$\sqrt{(\text{Clearance surface} + 2 \times \text{piston area}) \div (2 \times \text{piston area})}$$

The effect of clearance surfaces evidently varies with the type of engine, temperature range, superheat, etc.; that is, with all those factors which influence cylinder condensation.

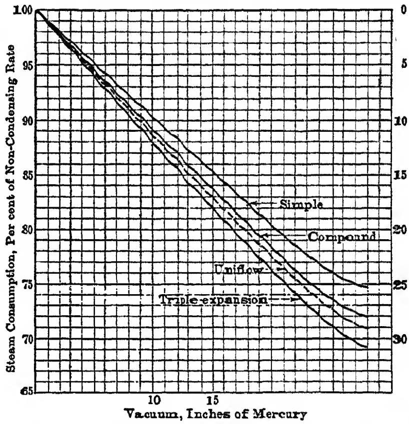


FIG. 46. Saving in Steam Consumption by Increase of Vacuum

The clearance volume as such is harmful, since it causes a given quantity of steam to do less mechanical work than it would do in an engine without clearance. Thus in Fig. 47 the same quantity of steam a does the work of the indicator card outlined in solid lines, in an engine with clearance, and does the additional work of the section-lined area enclosed in the dotted lines, in an engine without clearance. Fig. 47 is a particularly simple indicator card; conditions usually are more complicated. The effect of clearance space as such (neglecting surface action) can be counteracted by carrying compression to initial steam pressure. See Fig. 48. The card in solid lines (with clearance) and the card in dotted lines (no clearance) have the same area. The larger cylinder required for a given power, the surface of the clearance space, and the additional cylinder condensation during compression, render this method of compensating for clearance space valueless.

Taking into account only the harmful effects of *volume* of clearance space, and not those of its surface, a geometrical construction shows that, on an average, each additional 1% of clearance volume in single expansion engines increases the steam consumption 2%. See references below. Heat interchange between the steam and the surface of the clearance space, as a rule, still further increases this loss. In compound and triple-expansion engines, the effect of clearance is not quite so harmful.

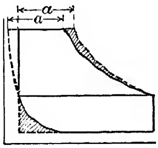


FIG. 47.

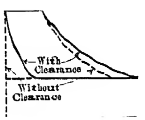


FIG. 48.

The smallest possible clearance space calls for valves in the cylinder head, with consequent reduced accessibility of the interior of the steam cylinder.

Non-condensing uniflow engines, on account of their long compression must have a large clearance volume, to prevent the compression pressure from rising above the steam pressure, or else they must have auxiliary exhaust valves which delay compression until more than half of the return stroke has been completed; see p. 7–39. Of the two, the latter design has been shown by several tests to have the better economy at mean effective pressures above 25 lb. per sq. in.

EFFECT OF COMPRESSION.—Taking into account only the volume of the clearance space, the starting point of that compression which results in greatest area of indicator card for a given quantity of steam, is given by the construction of Fig. 49. Joining

the cut-off point B to point A, extending the line to intersect clearance line at C; joining C with admission point D and extending the line to intersect the back-pressure line, we find point E at which compression should start (J. Stumpf, Uniflow Steam Engine, 1922). This construction results in short compression for long cut-off and long compression for short cut-off, a condition realized in most linkages and shaft governor valve gears. It shows that there is a most favorable compression for any given case.

In reality, matters are much more complicated on account of cylinder condensation during compression. Toward the end of long compression, the steam temperature exceeds the wall temperature, and cylinder condensation is not delayed until fresh steam is admitted, but actually begins during compression; the total time for cylinder condensation is increased and steam consumption rises. Steam-jacketed cylinders can economically carry earlier compression than non-jacketed, because wall temperature is higher. Tests by Heinrich (Mitteil. ü Forsch., 1914) have shown that between 2% and 50% compression, with cut-off constant, the steam economy does not vary more than 4% (Fig. 50). The test engine had 8 2/3% clearance volume.

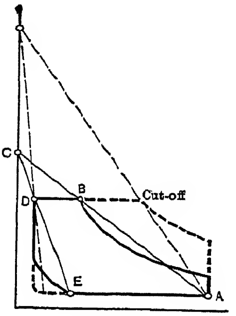


FIG. 49

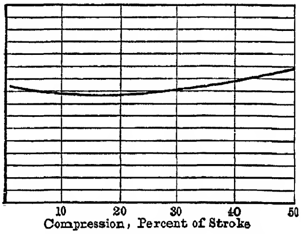


FIG. 50. Effect of Compression on Steam Consumption.

Compression earlier than that indicated by conditions of steam economy may be necessary for quiet operation of engine. See page 7–29.

References.—Heinrich, Mitteilungen über Forschungsarbeiten, *Zeit.*, V. d. I., Heft 146, June, 1914; Klemperer, *Zeit.*, V. d. I., 1905, p. 797.

EFFECT OF LEAKAGE.—Leaky pistons and valves can waste more steam than the best valve gear can save. Unsuspected leakage will readily lose as much as 10% of the total steam. (Barrus, Engine Tests.) No standard methods exist for determining the leakage of engines separately from losses by cylinder condensation. For approximate experimental determination, see A.S.M.E. Power Test Code. A valve may be perfectly tight when standing still, but leak badly in operation. For example, in piston valves, when not moving, steam works behind the rings and sets them out against the bushing, making a tight fit; but when moving, there is not sufficient time for steam to attain much pressure behind the rings, and the fit of the rings against the bushing depends largely upon the elastic pressure of the rings themselves; also on the surfaces of the valve and bushing which are exposed alternately to high and low pressures, steam condenses, the surfaces then slide over each other without removing the water film, and the latter flashes into steam when exposed to the low pressure and thus escapes to the exhaust.

Leakage affects the steam consumption per Hp.-hr. more at light loads than at heavy loads. Leakage through slide valves is never less than 4%, and often exceeds 20% of the total steam consumption at rated loads. Tests of slide valves (Callendar and Nicholson, *Proc.* Inst. C. E., 1897) show that the leakage Q in pounds per hour is given by the formula

$$Q = (C \times p \times B) \div l, \quad . \quad . \quad . \quad . \quad . \quad . \quad . \quad . \quad . \quad [17]$$

in which C = constant (about 0.02 for well fitted valves); p = pressure difference, lb. per sq. in.; B = perimeter of valve, in.; l = average overlap, in. The same value of C seems also to apply fairly well to piston valves without rings.

Tests made by the Penna. Railroad at Altoona, Pa., with a piston valve 12 in. diam., showed the following average leakage at *each end* of the valve, in pounds per hour per inch of circumference (leakage length); with one ring, 16.4 lb.; with two rings, 7.2 lb.; with three rings, 6.0 lb. These figures were obtained with steam at 185 lb. per sq. in. gage pressure, and atmospheric pressure at the other end of the valve. With 100° F. superheat, the leakage was about 15% greater, but with 200° F. superheat, it was only 7% greater than that of saturated steam. The leakage was inde-

pendent of speed and length of stroke. Within the range of usual pressures it is probably proportional, approximately, to the pressure difference.

Leakage is of greatest importance in a single-valve engine, since steam can leak directly into the exhaust without entering the cylinders. It is not so serious in an engine having separate valves for inlet and exhaust, since leakage steam must enter the cylinder, and do some work. It is still less wasteful in the uniflow engine. As it is shut off from the exhaust ports by the piston during 90% of the stroke, nearly all leakage steam must do work. Furthermore, a slight leakage raises the compression pressure considerably, causing the inlet valves to clatter so that leakage cannot occur unnoticed.

References.—*Engg.*, July 4, 1919; *Proc.* Inst. of C. E., 1897, Vol. 131; *Engr.*, Mar. 24, 1905 and Feb. 9, 1912; *Power*, Oct. 11, 1910.

EFFECT OF CHANGE OF LOAD. WILLANS LAW.—The combined influences of cylinder condensation, leakage and free expansion losses (the relative effectiveness of each factor varies with type of engine), result in *variation* of steam consumption *with load* as shown in Fig. 51 (pounds of steam per I.Hp.-hour) and Fig. 52 (total steam consumption per hour). The dot-and-dash lines in Fig. 52 illustrate *Willans Law*, which states that total steam consumption, at any load, of an engine controlled by throttling is $C_1 + C_2 \times$ (indicated horsepower), in which C_1 and C_2 are constants for a given engine; C_1 represents steam consumption at no load, or steam lost by cylinder condensation and leakage; C_2 is the steam used per hour to produce one indicated horsepower. This holds almost exactly for throttling engines and for turbines, but cannot be applied to engines controlled by varying the cut-off.

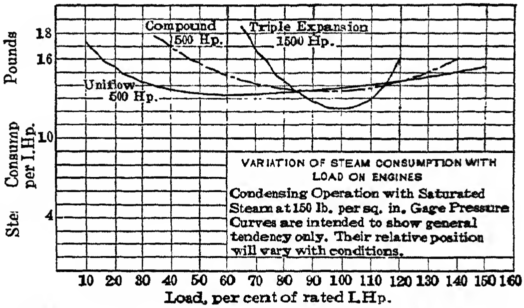


Fig. 51. Variation of Steam Consumption with Load on Engine

EFFECT OF VALVE GEARS AND OF METHOD OF CONTROL.—Narrow, long ports and slowly opening and closing valves result in wire-drawing through such ports and valves. For calculation of this effect, see Capacity of Steam Engines, p. 7–07. The loss of area of the indicator card divided by the area of the ideal card, furnishes directly the loss due to such wire-drawing. The arrangement of valves and ports likewise affects economy through its effect upon cylinder condensation.

EFFECT OF MOISTURE IN STEAM.—Tests by Marks and by Denton show that increase in the amount of moisture in steam at the throttle, up to 40%, has practically no effect upon the dry steam consumption; the water remains inert, passing through the cylinder without helping or hindering. (*Trans.* A.S.M.E., vol. xv., 1893.)

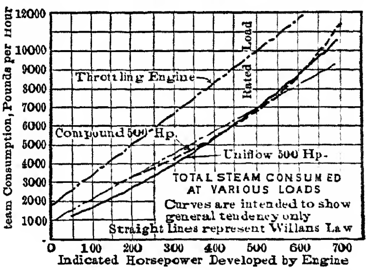


Fig. 52.

EFFECT OF REHEATING RECEIVERS.—Reheating receivers dry or superheat the steam going to the low-pressure cylinders of compound or triple-expansion engines, and also to the intermediate cylinders of triple-expansion engines. Reheating receivers, if made large enough, eliminate cylinder condensation, but require an equivalent amount of live steam in the reheaters. Hence, no gain results from their use alone. In compound engines, however, the *combination* of a reheating receiver having 1/4 sq. ft. of heating-surface per horsepower, and of jackets on the low-pressure cylinder, has shown a slight gain, while each of the two expedients by itself produced a loss. In triple-expansion pumping engines, reheaters are successfully used in combination with jackets on all cylinders.

References.—Barrus, New England Cotton Mfrs. Assoc., April, 1903; *Power*, Sept., 1903.

Table 7.—Steam Consumption, Rankine Cycle Ratio and Thermal Efficiency of Various Types of Engines

Test No.	Make of Engine	Location	Type of Engine *	Authority	Reference
	Simple Engines:				
1	P.R.R.Locomotive 1499		V_1		Loco. Tests & Exhibits, Penna. R.R.
2	Fleming	Cornell University	V_4	Carpenter	Gebhardt, "Steam Pr. Plant Engg.," p. 417
3	Erie City Lentz	Shop test	$P-V_4$	Moyer	*Power*, Dec., 1913, p. 779
4			R	Barrus	Barrus Engine Tests, No. 2
5	Augsburg-Nürnberg	Shop test	P-V-J		*Zeit.*, V. d. I., Aug., 1905, p. 1310
	Compound Non-condensing:				
6	P.R.R.Locos. Nos. 709, 710 & 711		V_1		Loco. Tests & Exhibits, Penna. R.R.
7	Allis-Chalmers	Interborough Subway, New York City	D-X-R	Stott	*Trans.* A.S.M.E., 1910
8	White automobile engine		V_1	Carpenter	*Trans.* A.S.M.E., 1907, p. 579
	Compound Condensing:				
9		Dow Chemical Co., Midland, Mich.	PV & R-X-H		*Power*, Aug. 23, 1927, p. 291
10	Rice & Sargent	Am. Sugar Refinery, Brooklyn, N.Y.	R-H-X	Jacobus	*Trans.* A.S.M.E., xxiv, p. 1275
11	Cooper	Atlantic Mills, Providence	R-X	Barrus	*Eng. Rec.*, Nov. 8, 1902, p. 436
12	Rice & Sargent	Melbourne Mills, Pa.	X-P-V-(h.p.) R (l.p.)	Jacobus	*Trans.* A.S.M.E., xxv, p. 27P
13	Easton & Co.	Brantham-Manningtree	H-N-F-V-(h.p.)R(l.p.)	Ewing	*Am. Elec.*, April, 1903, p. 178
14	Cole, Marchant & Morley	Belfast	$(4V_p)K$	Longridge	*Eng.* (London), June 2, 1905
15	Schmidt		HNPVJ	Hartmann	*Zeit.* V. d. I., vol. 65 (1921), p. 665
	Triple Expansion Condensing:				
16	Coates	Belfast	T-R		*Eng.*, Aug., 1894, p. 230
17	Sulzer Mill Engine	Augsburg			*Zeit.*, V. d. I., May 16, 1896
18	Sulzer	Luisenstrasse, Berlin	4L-T		*Zeit.*, V. d. I., May, 1900, p. 606
	Pumps and Compressors:				
19	Nordberg Pumping	Wildwood Pumping Sta., Pittsburgh	Q-G	Carpenter	*Eng. Rec.*, 1899, I, p. 495
20	Nordberg Compressor	Michigan Copper Mine	Q-G	Hood	*Trans.* A.S.M.E., xxviii, p. 705
21	Worthington Uniflow Pump	Buffalo	M-H-W		*Fire & Water Engg.*, Sept. 22, 1920
22	Allis-Chalmers	Chestnut Hill, Boston	K-T		*Eng. News*, Aug., 1900, p. 125
23	Holly	Louisville, 1909	T		*Bull.* of Holly Mfg. Co.
24	Allis-Chalmers	Cleveland, 1918	T		*Bull.* Allis-Chalmers Co.
	Uniflow Engines:				
25	Ames	Shop test	(NC)-W-PV-K-4L	Barrus	*Mech. Engg.*, 1925, p. 440
26	Skinner	Shop test	U-W-P-V		*Bull.* Skinner Engine Co.
27	Ames	Shop test	(NC)-W-P-V	Low	*Power*, Nov. 17, 1914, p. 701
28	Ames	Shop test	C-W-P-V	Low	*Power*, Nov. 17, 1914, p. 701
29	Görlitz		C-W		*Zeit.*, V. d. I., Oct. 7, 1911
30	Swiderski		C-W		*Zeit.*, V. d. I., Oct. 7, 1911
31	Nordberg	Lakeland, Fla.	C-W-2V		Private communication

* Symbols in this column have meanings as follows: C = condensing; D = duplex; F = lift; G = regenerative; H = horizontal; J = jacketed; K = vertical; L = cylinder; M = simple; N = tandem; (NC) = non-condensing; P = poppet; Q = quadruple-expansion; R = Corliss; S = piston; T = triple-expansion; U = Universal; V = valve; V_p = piston valve; V_1, V_2, V_3, V_4 = 1, 2, 3 or 4 valves, respectively; W = uniflow; X = cross-compound.

Table 7.—Steam Consumption, Rankine Cycle Ratio and Thermal Efficiency of Various Types of Engines—*Continued*

Reference No.	Cylinder Size, in.	R.p.m.	Actual I.Hp.	Steam Pressure at Throttle, lb. per sq. in., gage	Moisture, percent, or Superheat, deg. F.	Back Pressure, in., of Mercury or lb. per sq. in., abs.	Steam Consumption, lb. per I.Hp.-hr.	B.t.u. per I.Hp. per min.	Rankine Cycle Ratio Referred to I.Hp., percent	Thermal Efficiency Referred to I.Hp., percent
1	22 × 28	120	975	196	1.0%	15 lb.	23.6	396	55.6	10.7
2	19 × 19	206	217	120.5	Saturated	Atmos.	22.5	378	70.4	11.2
3	19 × 21	208	282	155.6	142°	Atmos.	15.2	279	84.0	15.2
4	34 7/16 × 60	60	613	82.3	2.5%	2.1 in.	18.5	337	49.5	12.6
5	22.6 × 45	48	262	79.0	....	2.7 in.	15.0	276	62.8	16.4
6	14.2 & 26.1 × 23.6	240	719	188	91°	17.8 lb.	17.6	316	70.7	13.4
7	(2) 42 & 86 × 60	75	9904	180	1.0%	12.1 lb.	16.1	274	79.7	15.5
8	3.66 × 4.5	850	40	426	316°	Atmos.	12.0	244	68.0	17.4
9	17 × 30 & 32 × 40	119	1550	354	145°	16.4 lb.	12.8	238	79.4	17.8
10	20 & 40 × 42	121	627	151	Saturated	1.7 in.	12.1	223	64.3	19.0
11	16 & 40 × 48	80	565	172.2	Saturated	2.3 in.	11.2	223	63.5	19.0
12	16 & 28 × 42	102	420	142.4	375°	4.0 in.	9.6	208	72.2	20.4
13	15 & 24 × 48	140	239	139	400°	2.0 in.	9.0	199	69.3	21.3
14	21 & 36 × 36	101	334	117	403°	2.6 in.	8.7	191	76.4	22.2
15	7.1 & 19.7 × 23.7	....	124.7	612	300°	1.5 in.	6.83	153	73.0	27.7
16	19 & 29 & 46 × 48	76	785	163	1.0%	2.0 in.	11.8	219	66.7	19.4
17	22.9, 44.5, 51.6 × 78.7	56	1823	134	Saturated	3.6 in.	11.3	208	78.5	20.4
18	34.1, 49.2, (2)61.1 × 51.2	82.5	2860	188.4	230°	....	9.6	198	73.2	21.3
19	19.5, 29, 49.5, 57.5 × 42	36.5	712	200	1.3%	2.3 in.	12.4	188	69.3	22.6
20	14.5, 22, 38, 54 × 60	57	990	242	5.7%	2.6 in.	11.9	168	75.3	25.2
21	13 1/2 × 21	152	171	222	173°	2.7 in.	11.9	241	58.3	17.6
22	30, 56, 87 × 66	17.7	801	185	1.4%	1.7 in.	10.5	196	70.5	21.6
23	30, 56.5, 84 × 60	24	926	155	110°	2.3 in.	9.7	187	77.5	22.6
24	36, 68, 108 × 66	21	....	207	136°	....	8.9	180	77.8	23.6
25	14 × 16	300	499.4	124	Saturated	5 lb. gage	22.4	372	78.0	11.4
26	21 × 22	....	300	140	Saturated	Atmos.	18.3	310	80.7	13.7
27	15 × 16	255	92.5	140	130°	Atmos.	16.8	304	80.0	14.0
28	15 × 16	255	131	145	93°	4.5 in.	12.5	238	69.0	17.8
29	523 Hp.	....	516	178	68°	3.0 in.	10.5	203	74.1	20.9
30	99 Hp.	....	100	110	419°	5.0 in.	9.4	199	78.9	21.3
31	20 × 32	150	366	217	47°	3.9 in.	11.63	217	71.6	19.5

EFFECT OF STEAM JACKETING.—Steam jackets raise the average temperature of the cylinder walls and prevent deep-reaching temperature fluctuations in them but they cannot prevent "skin deep" temperature fluctuations. In consequence they are of no effect in high-speed engines, as much steam being lost by condensation in the jackets as is saved by decreased condensation in the cylinder. They are, however, beneficial in slow-speed engines, such as pumping engines, in which the time for cylinder condensation is so great as to produce deep-reaching temperature fluctuations if no jackets are used.

References.—Hanszel, *Zeit.*, V. d. I., Jan. 13, 1912; *Power*, Mar. 18, 1913; Stumpf, Uniflow Steam Engine, 1922; Barrus, Engine Tests; Isherwood, Experimental Researches.

EFFECT OF UNIFLOW PRINCIPLE.—The effect of the uniflow principle has been discussed under Cylinder Condensation. An additional effect, favorable to economy, is caused by the sudden and short exhaust. Heilmann, *Zeit.*, V. d. I., June 10, 1911, and Doerfel (private report) contend that the explosion-like exhaust mechanically sweeps the water off cylinder walls and does not give the heat in the cylinder walls sufficient time to evaporate the water. Hence, the average wall temperature is higher and initial condensation is reduced. Tests demonstrating this action are lacking. The clearance volume can be made very small in the condensing uniflow engine; this favorably influences the efficiency.

EFFECT OF ENGINE FRICTION.—Engine friction appears as the difference between indicated horsepower and brake horsepower. Both are easily determined, but since the power absorbed by friction is, in a given engine at constant speed, almost constant at all loads, increasing only very slightly as load increases, it is more usual to take a "friction card," showing the indicated horsepower when the engine is delivering no power at the shaft, and all of the work shown by indicator card is being absorbed by engine friction. In blowing engines, pumps and compressors, engine friction appears as the difference between indicated steam horsepower and indicated pump horsepower. Thurston makes the following approximate distribution of the total losses due to friction: Main bearings, 47%; piston and rod, 33%; crank-pin, 7%; crosshead and wrist pin, 5%; valve gear, 8%.

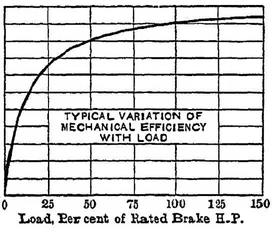


Fig. 53.

The ratio of brake horsepower (or indicated pump horsepower, respectively) to indicated steam horsepower is the *mechanical efficiency*.

Mechanical efficiency $= \frac{\text{I.Hp.} - \text{friction Hp.}}{\text{I.Hp.}} = 1 - \frac{\text{friction Hp.}}{\text{I.Hp.}}$. As friction horsepower is nearly constant, the mechanical efficiency varies with the load. Fig. 53 is a typical curve showing such variation. Mechanical efficiency depends upon the type and effectiveness of lubrication, upon the quality of workmanship and correctness of alignment of the engine, and upon the degree of cleanliness of the place in which the engine is to work.

It becomes lower as the initial pressure and number of cylinders are increased, and rises with increasing ratio of M.E.P. to initial pressure and with increasing size of engine. It is seldom below 85% at full load, while the highest record is about 97 1/2%. The mechanical efficiency of vertical engines is higher than that of horizontal engines of similar

Table 8.—Mechanical Efficiency of Steam Engines

Average values for every-day operation at rated load. Engines in good condition.

		Percent
Simple Engines:	Portable engine, 50 Hp	83
	Horizontal automatic engine, 150 Hp	90
	Horizontal Corliss engine, 300 Hp	91
	Horizontal uniflow engine, 400 Hp	90
	Locomotive	85
Compound Engines:	Horizontal automatic engine, 300 Hp	90
	Horizontal Corliss engine, 500 Hp	89
	Horizontal blowing engine, automatic, 2000 Hp	88
Triple-Expansion Engines:	Vertical power engine	90
	Vertical pumping engine, slow speed, 1500 Hp	94

type. Other factors being equal, the variation of mechanical efficiency with size of engine is expressed by

$$E_m = A + 2\sqrt[4]{\text{I.Hp.}} \qquad [18]$$

For a horizontal simple engine, constant A = about 79, so that for such an engine

$$E_m \text{ (percent)} = 79 + 2\sqrt[4]{\text{I.Hp.}} \qquad [19]$$

Table 8 shows average mechanical efficiencies obtained at full load, under ordinary conditions of operation, with engines in fair shape.

References.—Heilmann, *Zeit.*, V. d. I. June 10, and Oct. 31, 1911; Fernald and Orrok, Engineering of Steam Power Plants.

6. QUIET OPERATION OF STEAM ENGINES

Vertical single-acting steam engines run quietly as long as the forces on the piston act always downward. This condition usually exists at low and medium speeds. At high speeds or low back pressure, the compression of steam in the cylinder is not sufficient to keep the moving parts in compression at all times, because upward inertia exceeds the compression force, and knocking results. Noisy operation in single-acting engines may be caused by piston slaps or by alternating forces in the valve gear. A table of inertia factors is given in Vol. 3.

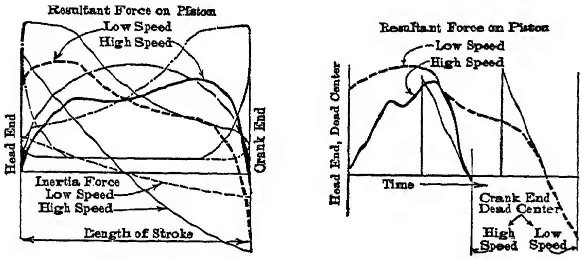


Fig. 54.

In double-acting engines, forces must change sign (tension to compression or *vice versa*) in every stroke. A knock occurs whenever the force changes sign suddenly and there is clearance in the joints. Play in the joints can be taken up, to a certain extent, by adjustment of brasses in main bearings, crank-pin and crosshead-pin. It can never be entirely eliminated, because pins wear oval, and deflect in operation; also because room must be allowed for an oil film and for unequal heat expansion. (For amount of clearance, see section on Machine Design, vol. 3 of this series.) This clearance can be filled with oil under pressure (forced lubrication; highly developed in British "quick revolution" engines), and knock can, thereby, be practically eliminated. Complete quietness is obtained by keeping the *rate of change* of force low, at the time the force changes sign. See Fig. 54. For a given shape of indicator card and a given engine, the rate of change of force increases with the revolutions, which is one reason why high speeds tend to produce pounding.

A second reason is due to the fact that the higher the speed and the greater the inertia forces the more is the position where the force changes sign moved toward the dead center. See Fig. 54. If the force changes sign *after* compression has been completed and when steam is being admitted, a knock occurs, unless steam is admitted gradually by a V-lead, or holes in the valve or valve seat. (See Valve Gears, Fig. 62.)

Long compression, with large clearance space, enables high-speed engines to operate very quietly. For detrimental effect on steam consumption of these features, see Steam Engine Economy, p. 7-23.

Direct-connected crank-and-flywheel pumping engines (steam engines with steam piston and pump piston on one common rod) including blowing engines and compressors, produce knocks in mid-stroke, because the forces change sign in mid-stroke. The knock is made more severe by the cross-heads vibrating from one side to the other.

Loose pistons are occasionally the cause of knocking and pounding. In high-speed releasing gears, the sudden contact of the lift arm with the valve arm produces knocking.

7. VALVE GEARS

References.—Dubbel, Steuerung der Dampfmaschinen, 1913; Furman, Valves, Valve Gears and Valve Diagrams; Zeuner, Treatise on Valve Gears; (see also Ninde, Ripper, Heck, Unwin and Allen).

Steam engines which operate pumps or compressors directly without a shaft, derive their valve motion directly from the motion of the piston. Steam engines turning a shaft derive the valve gear motion from cranks, eccentrics or cams on the main engine shaft or on a "lay" shaft driven by the main shaft. In the selection of valve gears, the possibility of enclosing the mechanism should be given careful consideration. Unenclosed valve gears throw oil, collect dust and dirt, and may injure attendants. In some cases the steam engine has been displaced by the small turbine because the advantage of complete enclosure could be shown for the latter.

VALVE GEARS OF DIRECT-ACTING STEAM ENGINES.—The piston, a short distance before the end of its stroke, operates a small pilot valve which admits steam to operate the main slide valve (Fig. 55). A time lag between the two valve movements allows the piston to complete its stroke. The diameter of the plug valve is usually 0.30 that of the steam piston. The size of slide valves and ports is based on a nominal steam velocity of 6000 ft. per min., and an exhaust velocity of 4000 ft. per min.

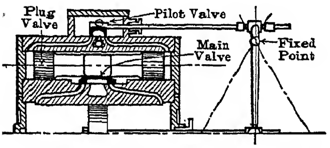


Fig. 55. Valve Gear of Direct-acting Engine

If two cylinders lie side by side as in a duplex pump, each piston in mid-stroke can reverse the motion of its neighbor, it being premised that the second half of the stroke will be completed in less time than the first half. If friction upsets this premise, one of the pistons "short-strokes," that is, does not complete its stroke.

VALVE GEARS OF CRANK-AND-FLY-WHEEL ENGINES.—In engines with revolving shafts, and valves operated by crank or eccentric, the fraction of the crank circle circumference during which the valve is open determines the ratio of opening travel to idle travel of the valve-moving mechanism. See Fig. 56. The idle travel may be reduced by linkages, as in Fig. 57, or eliminated by cams as in Fig. 58. The latter method is necessary on all non-releasing poppet-valve gears.

In Fig. 59, if the valve is to open when the piston stands at (3), and is to close with the piston at (4), both points being the mean between head end and crank end, then the valve opens when the engine crank is at (1) and closes with it at (2). The valve is wide open

Table 9.—Steam Velocities in Ports and through Valve Openings

Higher steam velocities reduce economy by excessive pressure drop. Lower steam velocities reduce economy by necessitating increase in clearance volume.

Type of Valve and Engine	Nominal Steam Velocity, Feet per Second		
	In Ports, Smallest Section	Through Greatest Valve Opening	
		Inlet	Exhaust
Slide-valve, slow-speed engine	70	90	70
Slide-valve, high-speed engine	130	175	130
Piston-valve, locomotive	130 to 165	170 to 220	130 to 165
Piston-valve, power engine, h.p. cyl	100	130	100
Piston-valve, power engine, l.p. cyl	120	150	120
Piston-valve, hoisting or mill engine, h.p. cyl	150	200	150
Piston-valve, hoisting or mill engine, l.p. cyl	165	215	165
Poppet-valve, power engine, h.p. cyl.*	....	75 to 105	65 to 95
Poppet-valve, power engine, l.p. cyl	....	80 to 130	70 to 120
Rocking-valve, power engine, h.p. cyl	....	120 to 140	80 to 120
Rocking-valve, power engine, l.p. cyl	....	140 to 160	80 to 120
Uniflow engine exhaust ports	....	....	60 to 120

* Velocities for double-beat poppet valves are based on nominal area—that of two circles having average diameter of valve seats. Velocities through greatest free opening, usually twice the above values.

The lower values apply to small engines, the higher values to large engines and superheated steam.

Length of port, for slide valves = 0.6 to 0.8 piston diameter; for rocking valves, 0.8 to 1.1 piston diameter, depending on valve diameter, which is usually 0.25 of the piston diameter when length = piston diam.

when the crank is at (5), half-way between (1) and (2). This condition determines the location of the valve-operating crank or eccentric on the main shaft or lay shaft. In particular, if the valve moves parallel to the piston, the eccentric is $(90 + \beta)$ degrees ahead of the crank or $(90 - \beta)$ degrees behind, depending upon opening direction of valve. The angle β is the angle of advance.

SIZE OF PORTS.—(See also Steam Engine Capacity.)—Port area commonly is determined by the equation *Port area* $\times$ *steam velocity* = *piston area* $\times$ *piston velocity*. In this equation the term "steam velocity" is only a conventional coefficient, as the

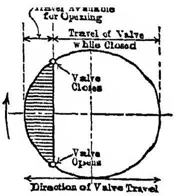


FIG. 56

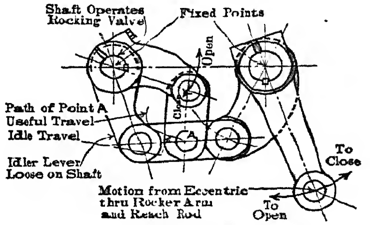


FIG. 57. Valve Gear Linkage

actual steam velocity varies continually because of the variation of piston velocity and of port area. See Fig. 60. In the use of so-called "steam velocities" from tables, care must be exercised because "piston speed" may mean either instantaneous or average piston speed. Table 9 is based upon *average* piston speed, extended over the whole stroke.

TYPES OF VALVES.—There are two broad classes of valves: 1. Sliding valves; 2. Lifting valves.

Slide Valves

DEFINITIONS.—Travel is total distance moved by the valve.

Throw of the Eccentric is eccentricity of the eccentric = distance from the center of the shaft to the center of the eccentric disc.

Lap of the Valve, also called outside lap or steam-lap is distance the outer or steam edge of the valve extends beyond or laps over the steam edge of the port when the valve is in its central position.

Inside Lap, or Exhaust Lap is distance the inner or exhaust edge of the valve extends beyond or laps over the exhaust edge of the port when the valve is in its central position. The inside lap sometimes is made zero, or even negative, in which latter case the distance between the edge of the valve and the edge of the port sometimes is called exhaust clearance, or inside clearance.

Lead of the Valve is the distance the steam-port is opened when the engine is on its center and the piston is at the beginning of the stroke.

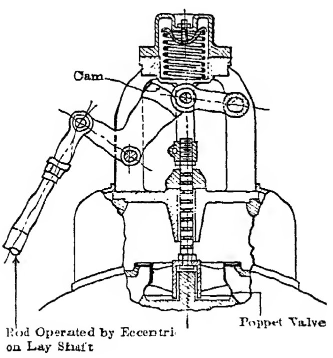


FIG. 58. Double Beat Poppet Valve

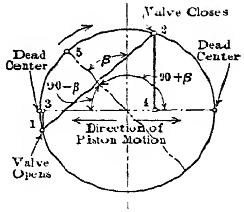


FIG. 59

Lead-angle is the angle between the position of the crank when the valve begins to be opened and its position when the piston is at the beginning of the stroke.

The valve is said to have lead when the steam-port opens before the piston begins its stroke. If the piston begins its stroke before the admission of steam begins, the valve

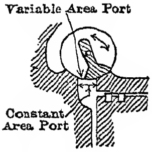
is said to have negative lead, and its amount is the lap of the edge of the valve over the edge of the port at the instant when the piston stroke begins.

Lap-angle is the angle through which the eccentric must be rotated to cause the steam edge to travel from its central position the distance of the lap.

Angular advance of the eccentric is (lap-angle + lead-angle).

Lineal advance is (lap + lead).

EFFECT OF LAP, LEAD, ETC., UPON STEAM DISTRIBUTION.—If a valve had neither lap nor lead, the center of the valve eccentric being 90° ahead of the crank center, then steam would be admitted to the engine throughout the full length of the stroke.



G. 60. Rocking Valve

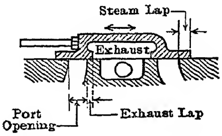


Fig. 61. D-Slide Valve

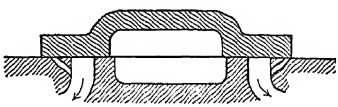


Fig. 62. Valve for Gradual Admission.

To enable steam to be cut off before the end of the stroke, and thereby allow expansion, lap is added to the valve. To enable steam to be admitted at the beginning of the stroke, as before lap was added, the eccentric is advanced on the shaft an amount equal to the lap angle. Advancing it still further by an amount called the lead angle causes admission to begin before the beginning of the stroke.

The four events, admission, cut-off, release or exhaust-opening, and compression or exhaust-closure, take place as follows: Admission, when the crank lacks the lead-angle of having reached the center; cut-off, when the crank lacks two lap-angles and one lead-angle of having reached the center. The greatest port opening is equal to half the travel of the valve less the lap. Therefore, in order not to reduce the port opening the valve travel must be increased if lap is increased. Adding exhaust lap to the valve delays release and hastens compression by an angle of rotation equal to the exhaust-lap angle, which is the angle through which the eccentric rotates from its middle position while the exhaust edge of the valve uncovers its lap. The above discussion of the relative position of crank, piston, and rod for the different points of the stroke is accurate only for a connecting-rod of infinite length. In practice, rods have a finite length, but the use of the infinite rod in laying out the valve diagram has the advantage that a mean between head and crank ends is obtained.

For actual connecting-rods the angular position of the rod causes a distortion of the position of the valve, causing the events to take place too late in the forward stroke and too early in the return. The correction of this distortion may be accomplished to some extent by setting the valve so as to give equal lead on both forward and return stroke, and by altering the exhaust-lap on one end so as to equalize the release and compression. F. A. Halsey (Slide-valve Gears) describes a method of equalizing the cut-off without at the same time affecting the equality of the lead. In designing slide-valves the effect of angularity of the connecting-rod should be studied on the drawing-board, and preferably by the use of a model. In the section on Machine Design, vol. 3, of this series, is a table which gives the crank-angles for various fractions of the stroke.

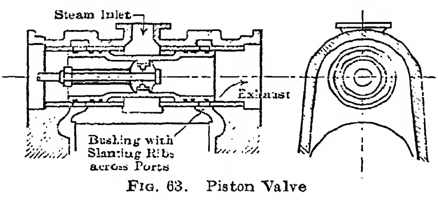


Fig. 63. Piston Valve

SLIDING VALVES.—The simplest form of *sliding valve* is the common flat slide valve, or D-valve, Fig. 61, which controls both steam admission and exhaust. It is much used for small portable engines working with saturated steam, but is not suitable for high steam pressures or for high superheats on account of distortion and rapid wear. It acts as a relief valve if the compression becomes too high or if there is too much water in the steam. It can be balanced by a pressure plate or by exposing part of the top of the valve to exhaust pressure. Steam can be admitted gradually at one end of the stroke by a V-lead or by holes in valve or valve seat as in Fig. 62.

The *piston valve* is perfectly balanced, and can be used with the highest steam pressures

and superheats, if the wearing parts of valves and valve bushings are free from the distorting influence of ribs, as is the valve in Fig. 63, and if the rings are protected against being set out by steam pressure. An example is the lock-ring type. The principal drawback of the piston valve is its large clearance volume. Clearance can be reduced by placing piston valves in cylinder heads (Van den Kerchove type).

The *rocking valve*, also called the Corliss valve, is single ported as in Fig. 60, for the slow speeds of pumping engines and ammonia compressors and double ported (Fig. 64) for piston speeds above 600 ft. per min. Separate valves are used for admission and for exhaust, the form of the latter being shown in Fig. 64. The rocking valve is a desirable

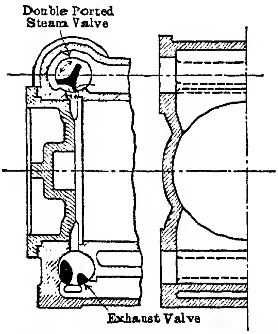


FIG. 64. Double-ported Corliss Valve

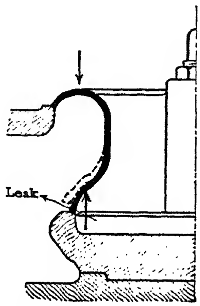


FIG. 65. Leakage of Poppet Valve

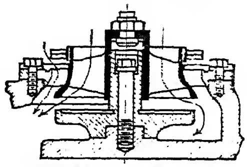


FIG. 66. Stumpf Valve

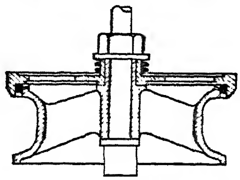


FIG. 67. Skinner Valve

form for use with saturated steam and for pressures up to 160 lb. per sq. in. gage. High steam pressures, or superheats exceeding 50° F., produce much friction, the first by squeezing the oil out of the space between the valve and valve seat, the second by distorting the valve and valve chamber. The general use of the rocking valve in the United States considerably retarded the introduction of high steam pressures and superheat, and paved the way for the steam turbine.

All sliding valves can be made *multi-ported*, to admit and shut off steam more quickly. For effect of multi-porting on economy and on shape of indicator card, see below.

LIFTING VALVES.—Single beat poppet valves, or mushroom valves, are sometimes used in the heads of low-pressure cylinders of triple-expansion pumping engines, to minimize clearance.

Double beat poppet valves, Fig. 58, have two seats of narrow width. They are very nearly balanced and can be entirely balanced if desired.

Steam pressure compresses the valve as indicated in Fig. 65 and tends to produce leakage. Unequal expansion, due to different metal or to differences of temperature (valve and one seat exposed to high temperature steam, other seat exposed only to low temperature steam) also causes leakage. To overcome this trouble, valves must either be ground in while under steam pressure (in which case they are tight for that particular pressure but leak if the pressure or superheat changes), or must be made flexible, as in the Stumpf valve, Fig. 66, or the Skinner valve, Fig. 67. Poppet valves can be used with the highest steam pressures and superheats. They have the disadvantage that the valve must be operated either by cams, rollers, or a pair of wiper cams.

In recent practice, cams and valve operating parts are hardened and ground, enclosed and lubricated by continuous oil streams. For speeds below 100 r.p.m., poppet valves may be operated by a releasing gear without a cam, although drop piston valves are preferred with this type of mechanism because quick closing is obtainable without slamming.

Valve Gear Diagrams

VALVE GEAR DIAGRAMS.—The diagram of valve opening at any piston position, Fig. 68, can be obtained from existing engines by placing the engine piston in different positions and measuring the valve opening for each position with various positions of the cut-off control mechanism, by blocking the governor at different heights. In engine design, the diagram can be taken by doing the same thing with a model, or by finding

various valve positions from a drawing. If the area of the valve opening exceeds the constant port area, the latter determines the steam flow. Constant port area is indicated by line AB, Fig. 68.

For steam admission valves, the foregoing diagram shows the range of cut-off; point

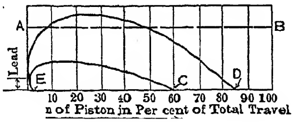


Fig. 68

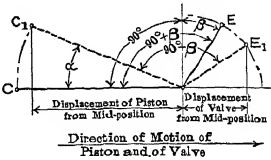


Fig. 69

of admission before dead center; lead, or linear valve opening at dead center, expressed in inches; maximum valve opening at short cut-offs; and wire drawing effect of valve. (See also Steam Engine Capacity, Fig. 20.) For exhaust valves, it shows the point of release, valve opening at dead center, and point of compression.

The interdependence and limitations of admission and exhaust events is shown by a different type of diagram, such as the Sweet, Zeuner, or Bilgram diagram, in which one variable indicates the positions of both valve and piston, for any crank position. The various valve diagrams differ by the choice of that variable. The use of one or the other is largely a matter of personal preference.

SWEET DIAGRAM, Fig. 70. (Also called Mueller diagram or Reuleaux diagram.) —The basis of the diagram is as follows: 1. Crank is turned ahead through angle $(90 + \beta)$. 2. Engine stroke is drawn to a reduced scale so that it coincides with valve travel. Then if the crank has turned through angle α from dead center position, DE indicates distance of valve from mid-position, while AC indicates the distance (average between head end and crank end), which the piston has traveled from end of stroke, expressed as a fraction of the whole stroke AB. By striking arcs EF and EG with radii $HE = IE$ so that HE/AB = ratio of connecting-rod length to engine stroke, we obtain AG = piston travel from head end, AF = piston travel from crank end dead center.* In the same manner, valve displacement DE can be corrected for finite length of connecting-rod, but as a rule the correction is negligibly small. DE is the valve displacement for all gears in which the valve is driven directly from an eccentric, without distortion by linkage or cams. Fig. 71 shows diagram derived from this principle, and the ideal indicator card (mean between head and crank end), with direction of *valve* motion horizontal, while Fig. 72 shows diagram with direction of *piston* motion horizontal and indicator cards for both head and crank ends. In both illustrations, the circle, drawn with any convenient diameter, represents both valve travel and piston stroke. If admission (point A) and cut-off (point B) are selected, the angle of advance is fixed. Either release or compression then can be selected, and the other one of the two is given, because CD is parallel to AB (same angle of advance).

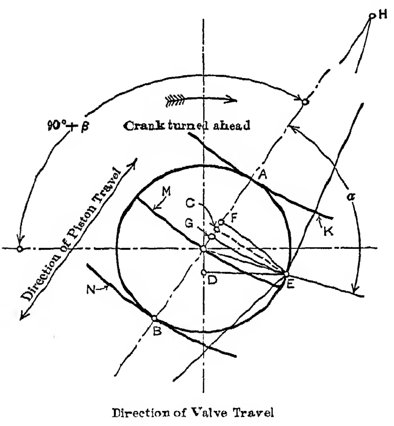


Fig. 70. Sweet Valve Diagram

The maximum port opening A, Fig. 71, is fixed by so-called mean steam velocity (see

* Instead of drawing arcs, as EG or EF, from each crank position, the arcs K, M, and N, drawn with radii = HE, may be used as reference lines representing ends and middle of stroke. Distance of piston from center of stroke when crank [illegible] distance parallel to AB from point E to arc M. With arcs as shown, A is head end dead center.

p. 7–30), and by available length of port. The ratio of travel to width of port is available from the diagram, and the valve travel can be found from the value of A. In using the diagram, the following precautions are to be observed: Admission must be $1/2\%$ to $1\ 1/2\%$ of stroke ahead of dead center, depending upon piston speed, compression, and

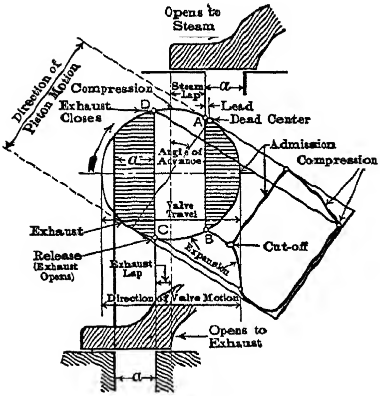


Fig. 71.

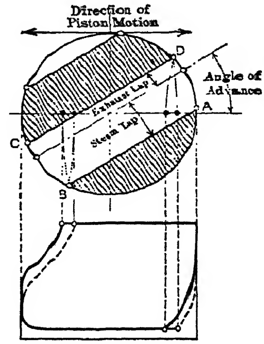


Fig. 72.

clearance volume. Cut-off in the actual indicator card is 10% to 20% earlier in the stroke than that determined from valve diagram, depending upon the so-called average steam velocity. For factors determining release and compression, see pages 7–13 and 7–23. Compression in the actual card begins 4% to 7% earlier than in the valve diagram. See p. 7–09. Release occurs from 5% ahead of dead center for low piston speeds, up to 25% for high piston speeds. If cut-off and compression are to be made more nearly equal for head and crank end, the valve must be moved toward the head end so that the head end lap is greater than the crank end lap. Complete equalization by this method is not desirable, because admission and release become unequal at the two ends.

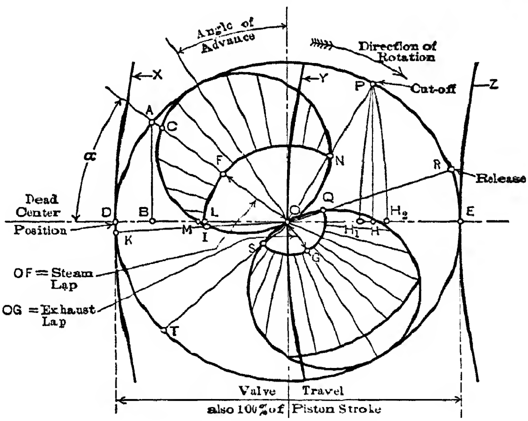


Fig. 73. Zeuner Valve Diagram

Piston valves usually have steam admission at the center, and exhaust at the outer ends of the valve. The valve phase is shifted 180° as compared to the plain slide valve. The valve diagram is the same as for the slide valve, with the following exceptions: Eccentric, instead of being (90° + angle of advance) ahead of crank in direction of motion, is (90° − angle of advance) behind crank. To equalize the cut-off, valve must be shifted toward the crank end.

ZEUNER DIAGRAM.—For finite length of rod, $R \cos \alpha$ = piston displacement from mid-position, if R = crank radius, α = angle through which crank has turned from dead center; $r \cos (\alpha + \beta)$ equals valve displacement from mid-position, if $2r$ = valve travel, and β = angle of advance. Any geometrical construction which, by one variable, gives

value of both $R \cos \alpha$ and $r \cos (\alpha + \beta)$ is a valve diagram. Zeuner, of Dresden, Germany, constructed a diagram, Fig. 73, by using rectangular co-ordinates for the piston travel, and polar co-ordinates for the valve travel.

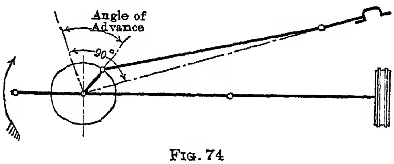


Fig. 74

For any crank-angle α, and crank position A, DB = piston travel (for DE = stroke), and OC = valve displacement from mid-position. At I, valve displacement = steam lap. Hence admission of steam occurs at K. Lead = LM. Cut-off occurs at N and P. Piston stroke at cut-off = DH for an infinitely long rod, = DH_2 at head end, = DH_1 at crank end.*

Release occurs when valve displacement equals exhaust lap (at Q and R). Compression occurs at S and T. The Zeuner diagram, although formerly very popular is now used less than the Sweet diagram.

LIMITATIONS OF VALVE-GEAR DIAGRAMS.—If the center line of the valve motion is not parallel to the center line of piston motion (see Fig. 74), the angle of advance marked in that illustration must be used. If valve motion is in any way distorted by linkages, cams, or skew drives, the simple valve diagrams cannot be used.

References.—Halsey, Slide-valve Gears; Fessenden, Valve Gears; Peabody, Valve Gears for Steam Engine; Spangler, Valve Gears.

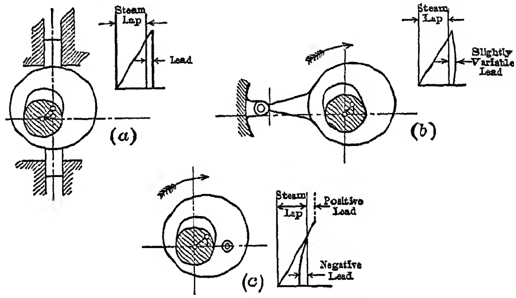


Fig. 75. Movable Eccentrics

SINGLE ECCENTRIC, VARIABLE CUT-OFF GEARS.—If the cut-off is to be variable with a single valve, and if admission and release are to remain correct over a wide range of cut-off, both valve travel and angle of advance must be varied in such a manner that the *lead* remains almost constant. This is accomplished by moving the eccentric " across the shaft," either in a straight line, as in Fig. 75 (a) or in the arc of a circle, as in Fig. 75 (b) and (c). In the former case, lead remains constant, while in the latter it is variable. Variability can be kept within narrow limits by proper location of the pivot. The pivot frequently is put on the center line of the crank [Fig. 75 (c)] to facilitate change of direction of rotation. It is desirable to reduce the lead to zero or to a negative value at minimum valve travel, in order to control the engine without load.

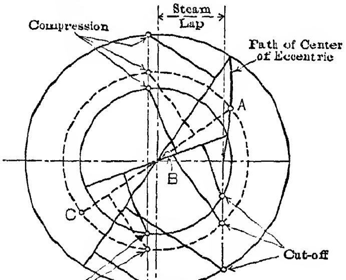


Fig. 76. Sweet Diagram for Slide or Piston Valve

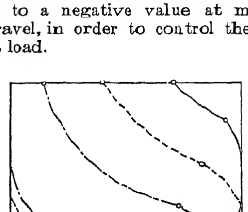


Fig. 77. Ideal Indicator Card for Slide or Piston Valve

* Instead of arcs PH_1, PH_2, etc., arcs XY and Z may be used as reference lines for the ends and middle of stroke, as explained for the Sweet diagram. With arcs as shown, D is the crank end dead center.

The Sweet diagram for a slide or piston valve, operated by a movable eccentric, is shown in Fig. 76, while the resulting ideal indicator cards, neglecting the effect of throttling, appear in Fig. 77. As the eccentric is moved closer to the center of the shaft, the valve travel is reduced, and with it the length of line representing 100% piston travel.

EXAMPLE.—$(AB/AC) \times 100$ = percent cut-off for position A of eccentric. As cut-off is shortened, release occurs earlier and compression becomes longer. Actual indicator card shows considerable throttling at early cut-off, on account of insufficient valve opening. (See Capacity of Steam Engines.)

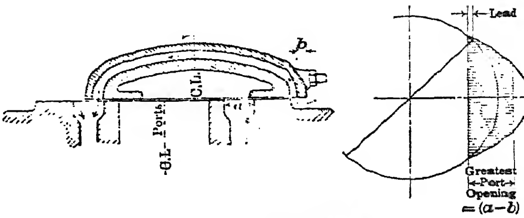


FIG. 78. Allen Valve

Single valves, driven by eccentrics which swing across the shaft, are much used in high-speed engines of office building power plants.

Excessive throttling is avoided by double admission valves, whose principle is shown by the Allen valve, Fig. 78. This valve admits steam directly and also through the valve from the other end. Port openings are doubled throughout, if the inlet port is made wide enough to accommodate the valve port and bridge. The Allen valve increases the clearance during a part of compression.

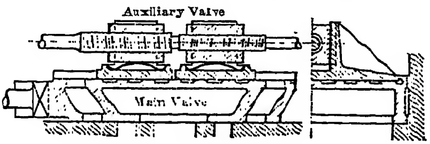


FIG. 79. Meyer Valve

RIDING CUT-OFF VALVES.—A single valve cannot vary the cut-off without change of compression. It is limited in its drive to shaft governors or to valve link-drive. Both of these drawbacks are avoided by *riding cut-off valve gears,* such as the Meyer valve, and others, illustrated in Fig. 79. The main valve controls admission, release and compression, while the auxiliary valve varies the cut-off. The latter is done either by changing the lap of the cut-off valve, or by varying the time of its motion by shifting the eccentric around the shaft, or by a combination of the two.

Relations can be read from the modified Sweet diagram in Fig. 80. The auxiliary or cut-off valve diagram determines cut-off only, the main valve diagram which is the same as for a single valve, determines admission, release, and compression. MA = 100% piston travel for the main valve diagram. EC = 100% piston travel for cut-off valve diagram. With the crank horizontal as shown, OM = main eccentric, OC = cut-off eccentric; then triangle OMC revolves about O, and MC = relative eccentric. The valve travel is horizontal, with CE as the direction of piston travel. The cut-off valve closes after the crank has turned through the angle C–5. $(CF/CE) \times 100$ = cut-off in percent of piston stroke. The greater the negative lap of the cut-off valve, the later in the piston travel does it close. If negative lap almost equals radius of relative eccentric, the cut-off valve reopens before the main valve has closed (with angles, etc., as shown in the diagram)

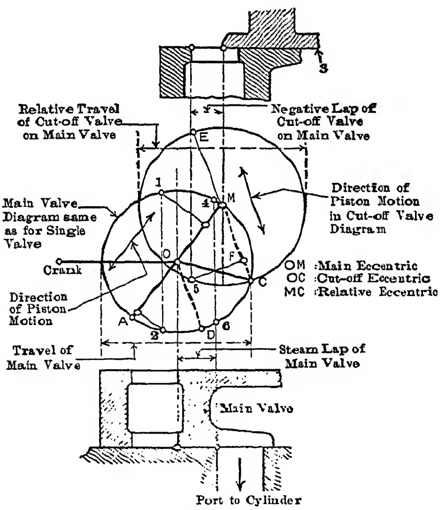


FIG. 80. Valve Diagram for Riding Cut-off Valve

MC must not be taken so large that there is not room on main valve for cut-off valve plus change of lap. Edge 3 of the cut-off valve must not be allowed to open when the main valve is open. The radius of the relative eccentric, *MC*, is usually made 1.6 to 2.0 times port width. The proportions in Fig. 80 give good results for average conditions. The angle of advance of main valve is from 30° to 35°. The distance between cut-off valves

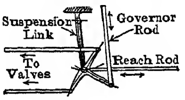
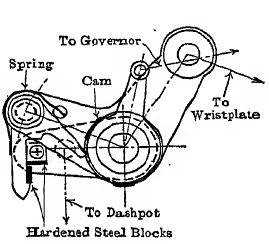


Fig. 81



82. Releasing Gear for Rocking Valves

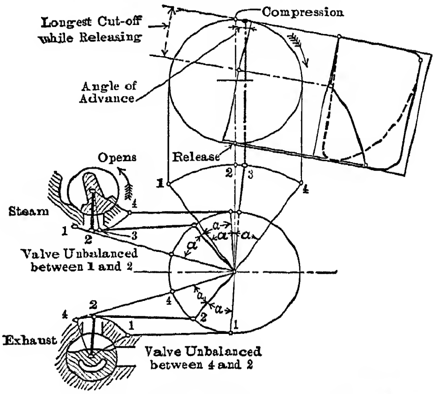


Fig. 83. Wrist Plate Linkage for Rocking Valves.

is adjusted by a right- and left-hand screw, or by a swinging bell crank as in Fig. 81; *MC*, Fig. 80, becomes both cut-off eccentric and relative eccentric, if its motion is superposed on that of the main valve as in the Buckeye engine, or if the cut-off valve operates in a separate fixed valve chamber.

VALVE GEARS WITH SEPARATE ROCKING VALVES FOR STEAM AND EXHAUST.—Rocking valves offer large unbalanced surfaces, if subdivided as in the four-valve type. Wasted friction work and wear are reduced by keeping the valves almost stationary when they are unbalanced. Such motion is obtained by " wrist plate " linkages, Fig. 83. Releasing cut-off is very limited in single eccentric gear as shown by the modified Sweet diagram in the upper part of the illustration. No intermediate cut-off is possible between 45% and 95%, because the valve gear moves away from the releasing cam, Fig. 82, in that interval. To make the releasing cut-off long, the angle of advance is made small (about 5°). This causes late release and short compression, and hence is suitable only for engines with small clearance and slow speed, such as ammonia compressors, slow-speed pumping engines, etc. For higher speeds, the releasing cam is made oscillating to overtake the gear on its return stroke, or else separate eccentrics are used for steam and exhaust, as in Fig. 84, which is a modified Sweet diagram (mean between head end and crank end). With double eccentrics, 3/4 cut-off can be obtained. The steam valve must be released, even with the governor in lowest position, otherwise the steam valve will be open at the same time the exhaust valve is open. The steam valve is moved directly, without distorting of motion by the wrist plate. The exhaust valve is driven through the wrist plate.

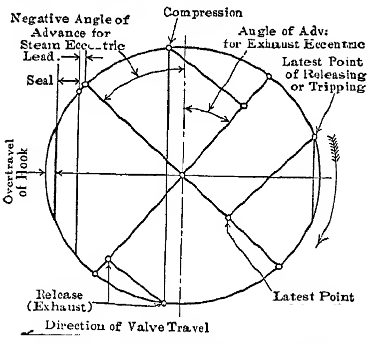


Fig. 84. Sweet Diagram for Double Eccentrics

Four-valve non-releasing gears usually have separate eccentrics for steam and exhaust. Valve diagrams are the same as for single valve gears but the travel of valve after closing is very much reduced by linkage as in Fig. 57.

Reference.—Moss, Layout of Corliss Valve Gears.

POPPET-VALVES AND GEARS.—If D = valve diameter, the free area on account of obstruction by the hub, ribs, and annulus of valve ranges from 0.65 ($\frac{1}{4}\pi D^2$) for small valves, to 0.85 ($\frac{1}{4}\pi D^2$) for large ones. The width of seat ranges from $^3/_{32}$ to $^3/_{16}$ in. For balancing purposes, the difference of diameters of upper and lower seats is small as possible, except in uniflow engines in which the top seat frequently is made considerably larger than the bottom seat for the purpose of making the valve relieve excessive compression in case of the failure of vacuum. In that case compression must be carried up almost to initial steam pressure, to relieve strains on the valve. The maximum valve lift is usually $^1/_8 D$.

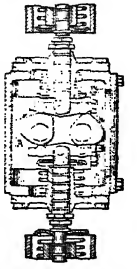


FIG. 85. Cam Mechanism for Double-beat Poppet Valve

In engines up to 500 or 600 Hp., the valves commonly are operated directly from a shaft governor on the main shaft. In larger engines, they are driven from a lay shaft parallel to the cylinder axis. Between 600 and 1000 Hp., releasing gears almost invariably are used; although occasionally, linkage gears with variation of linkage by a spindle governor are built. Poppet-valve gears have been brought to a high state of perfection by European engineers. Their number is so great that they cannot be described in this volume. See Steam Engine Valve Gears by Dubbel (German), published by J. Springer, Berlin.

Poppet-valve gears, as valve gears, are subject to the same limitations as Corliss valve gears, with the added complication that there is no over-travel possible after the valve has closed. Consequently, non-releasing poppet-valves all are moved by cams. The cam, as a rule, only lifts the valve, while a spring returns it to its seat. If double-acting cams are used, a rather stiff, but elastic member must be introduced to keep the stresses from becoming excessive.

All "automatic" valve gears, whether operated by shaft governor or by linkage, result in very small valve openings at short cut-offs, just as in slide valve gears. In releasing gears, separate eccentrics for steam and exhaust are necessary, just as in Corliss valve gears.

A great difficulty of releasing poppet-valve gears lies in the fact that valves either hammer or else leak, correct seating seldom being obtained. Air dash-pots close properly at some lifts, but not at others. Oil dash-pots work well at all lifts, at certain oil temperatures. When the oil is cold, the valves do not seat; when it is hot, the valves slam.

A double-cam mechanism for poppet valves is shown in Fig. 85.

In some European engines, hydraulic (oil-pressure-operated) valve gears have been used with success, also compressed-air-operated valve gears. (See *The Engr.*, 1928, p. 134.)

VALVE GEARS OF UNIFLOW ENGINES.—Exhaust slots must be long enough to allow the exhaust pressure to drop down to condenser pressure with the latest cut-off (30 to 33%). As a rule the length of slots equals 10% of the stroke. The cylinder must

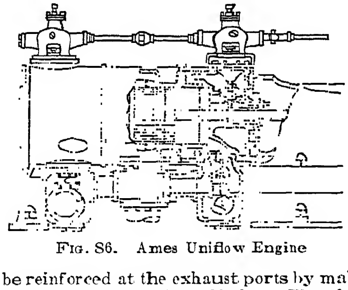


FIG. 86. Ames Uniflow Engine

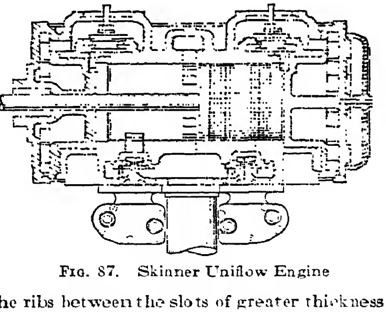


FIG. 87. Skinner Uniflow Engine

be reinforced at the exhaust ports by making the ribs between the slots of greater thickness than the body of the cylinder. The clearance volume should be such that the compression rises to within about 20 lb. per sq. in. of the steam pressure; with good vacuum, however, this is not possible. For the exponent of the polytropic curve, and for increase of compression by throttling, see p. 7-09.

Inlet valves, almost exclusively, are of the double-beat poppet type, in order to allow the use of the engine with high steam pressure and high superheat. For greatest economy, involving high vacuum, the inlet valves are in the cylinder heads; for moderate economy and lower vacuum (24 to 25 in.) valves may be in the top of the cylinder. Uniflow engines, which always operate condensing, except in emergency cases, are equipped with auxiliary clearance volume in the cylinder heads, which is added to the working space by the engine in case of emergency. Engines which always or frequently operate non-

condensing usually are equipped with auxiliary exhaust valves. Fig. 86 represents the exhaust valves of the Ames engine and Fig. 87, those of the Skinner engine. For European designs of auxiliary exhaust valves, see Dubbel, Valve Gears; also, *Power*, Feb. 13, 1923. The auxiliary exhaust is adjustable to adapt the engine to different back pressures with constant clearance volume. For other data on inlet valve gears, see above under

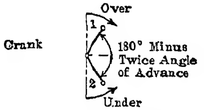
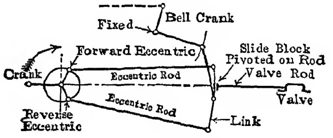


Fig. 88. Two Eccentrics



. 89. Stephenson

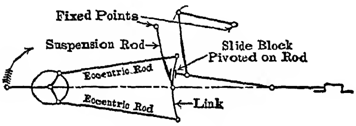


Fig. 90. Gooch

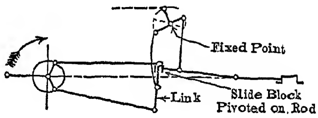


Fig. 91. Allen

Reversing Gears

Poppet-valve Gears. Cut-off usually is limited to 30 or 35%, in order to get sufficient valve lift at normal cut-off. In condensing engines it ranges from 8 to 15%.

References.—Stumpf, Uniflow Steam Engine, 1922. Trinks, Assoc. of Iron and Steel Elec. Engrs., 1916, p. 139; Heilmann, *Zeit.*, V. d. I., Oct., 1911; Freytag, *Zeit.*, V. d. I., May 9, 1914.

REVERSING GEARS.—Reversal of the direction of rotation is obtained in the most perfect manner by operating the valves from an eccentric which is (180° − 2 × angle of advance) out of phase with the other eccentric. See Fig. 88. In a crude manner, the engine can be reversed by interchanging the steam inlet and exhaust or by reversing the motion of the valve, which is equivalent to driving it by an eccentric which is 180° out of phase.

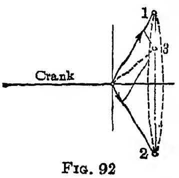


Fig. 92

Changing from eccentric (1) to eccentric (2) can be effected by actually shifting the eccentric, or by using two eccentrics. In the latter case the eccentric rods either can be provided with gab-hooks and can be made to alternately engage the valve rod at the will of the engineer, or they may be connected to a link. The former method is still in use on paddle-wheel steamers. Linkage reversing gears are the preferred form, because the change of position of the block in the link allows various combinations of motion of the two eccentrics. For arrangements of link gears see Figs. 89 (Stephenson), 90 (Gooch) and 91 (Allen). The general effect of a reversing gear link is illustrated by Fig. 92. Shifting the relative position of block and link results in using fractions of the motion of each eccentric and in combining these

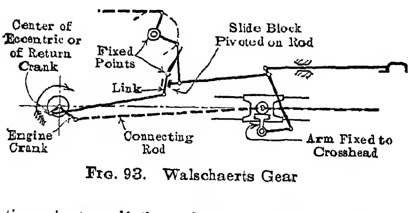


Fig. 93. Walschaerts Gear

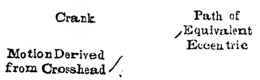


Fig. 94

motions just as if the valve were driven directly from an equivalent eccentric (3). This means that center of the equivalent eccentric "is shifted across the shaft" and that cut-off, compression and release vary as above described under the heading of single-eccentric variable cut-off gears. Fig. 89 shows "open rods." If, with the crank in the position shown, the rods are crossed, the arrangement is known as "crossed-rods."

The path of the equivalent eccentric is, as a rule, curved, as indicated by the dotted lines in Fig. 92, depending on the relative length of link, eccentric throw, eccentric rods, on the suspension and guiding of link as well as on the location of rod pivots and of the block in the link. Accurate determination may be made by means of models or of point-by-point construction on the drawing board. For the effect of the above-mentioned details, see Halsey, Slide Valve Gears; Furman, Valves, Valve Gears and Valve Diagrams.

WALSCHAERTS GEAR (Fig. 93).—If one of the eccentrics is replaced by the main crank of the engine, the Walschaerts gear results. The mechanism utilizes a constant fraction of the crosshead motion, but by means of the linkage the motion, derived from an eccentric keyed on the shaft at right angles to the crank, is made adjustable and reversible. See Fig. 94. The result is a straight line path of the equivalent eccentric, with the exception of the error caused by the obliquity of the connecting-rod. The Walschaerts is a very desirable valve gear for locomotives, despite the large number of joints. The eccentric is replaced by an outside return crank when used on locomotives. Its advantages are: Low friction work; all parts lie outside and are accessible for inspection and repairs. For other details see Halsey, Slide Valve Gears; Furman, Valves, Valve Gears and Valve Diagrams. See also p. 14–38.

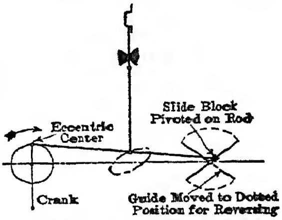


FIG. 95. Radial Valve Gear

Reversing gears are of the greatest importance in locomotives, where only those reversing gears are useful in which the up-and-down motions of the steam cylinder relative to the wheels do not materially affect steam distribution. This consideration is absent in reversing gears for marine engines, which allows the use of so-called "radial" valve gears. Only one eccentric is used, and a point of the eccentric rod is guided in an adjustable direction. Each point of the eccentric rod describes a different curve, which varies with the inclination of the guide point. Motion is taken from the eccentric rod at right angles to its mid-position. The various radial valve gears differ with regard to the relative positions of the eccentric point, guide point and valve operating point, and also with regard to the method of guiding the guide point. See Fig. 95. In the position of the eccentric shown in the illustration, the valve position is independent of the direction of the guide and lead is constant. Radial valve gears are described in detail in books and periodicals on marine engines.

References.—General references for valve gears, in addition to those given above are: Auchincloss, Link and Valve Motions; Durand, Practical Marine Engineering; Dalby, Valves and Valve Gear Mechanisms; Hiscox, Modern Steam Engineering.

8. OPERATING DATA

REGULATION TROUBLES.—See Governors, vol. 3 of this series; also Trinks. Governors, and Governing of Prime Movers. If regulation is not close enough (too much difference between full-load speed and no-load speed), use a smaller fraction of governor travel, or alter the governor, for instance by making the spring more flexible. This can be done by grinding off some of the outer part of the spring wire, making it thinner, or by reducing the width in the case of a leaf spring.

Racing of engine may be due to many causes, such as no oil, or not sufficiently heavy oil in the governor gag pot; excessive friction in the governor; too much steam admission at no load; too much steam volume beyond control of governor; leakage in the pilot valve of relay governor; lost motion of relay mechanism; dull edges of catch hooks in releasing gears; improperly adjusted or worn vacuum pots of releasing gears; too close setting of governor; insufficient capacity; improperly designed or poorly adjusted linkages between governor and control valve; reaction of valve gears upon governor.

HOT BEARINGS result from many different causes, such as insufficient oil; dirty oil, even a powder so fine as not to cause any gritty feeling being sufficient to heat a bearing; water in oil; use of oil that is too light or too heavy; undue tightness of bearing; wrong shape or location of oil-grooves; improper bedding of bearing, or contact in spots; binding, due to lack of alignment; improper bearing material; too high rubbing speed or pressure (product of *bearing pressure* × *revolutions per minute* is excessive); insufficient opportunity to get rid of friction heat; poor finish of journal or box.

In each individual case of a hot bearing, a search must be made for any of the above-mentioned causes.

KNOCKING AND POUNDING.—See Quiet Operation of Engines, p. 7–29. In addition to the reasons therein mentioned, excessive water in the steam will produce severe knocking.

STARTING.—Warming up before starting is advisable. Four-valve engines usually are equipped for hand operation of the valves for warming up purposes. A single valve engine should be started slowly with the drain valves wide open. In engines which have to be started suddenly against full load, relief-valve springs are set rather loosely.

VALVE SETTING should be checked regularly, say, every two weeks, with the indicator. Common defects and their cause are indicated in Fig. 96.

WEAR OF CYLINDERS AND OF PISTON RINGS.—Under average conditions, a set of piston rings travels 330 million feet before removal becomes necessary. Steam cylinders last for 2450 million feet of piston travel, before re-boring becomes necessary. Cylinder walls are made thick enough for two reborings. For day and night operation, this means that a set of rings lasts 14 months, and that the cylinder must be rebored every 8 years, and must be replaced after 25 years. Both figures are influenced by the quality of material of the rings or of the cylinder and by the effectiveness of lubrication. Thus a set of rings sometimes is worn out in two months, but in some cases will last two years. If cylinders are made of ordinary soft cast iron, the wear may be five times as rapid as indicated by the above figures, which apply to castings of hard cylinder iron. After a

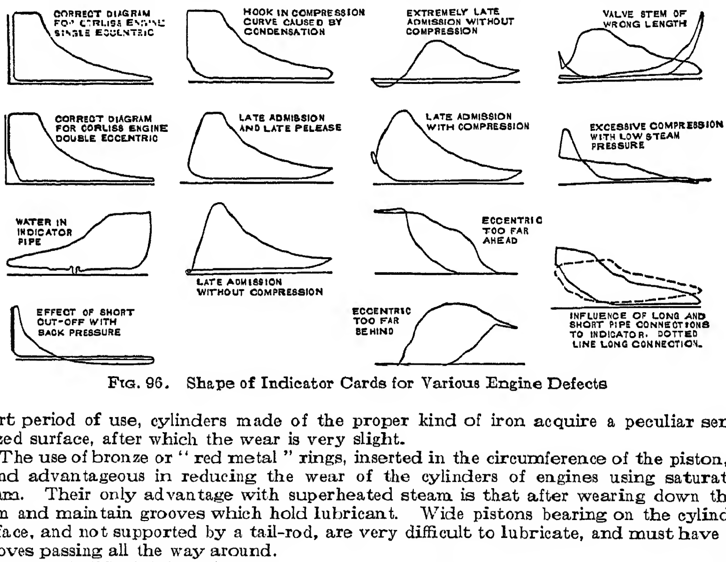


Fig. 96. Shape of Indicator Cards for Various Engine Defects

short period of use, cylinders made of the proper kind of iron acquire a peculiar semi-glazed surface, after which the wear is very slight.

The use of bronze or "red metal" rings, inserted in the circumference of the piston, is found advantageous in reducing the wear of the cylinders of engines using saturated steam. Their only advantage with superheated steam is that after wearing down they form and maintain grooves which hold lubricant. Wide pistons bearing on the cylinder surface, and not supported by a tail-rod, are very difficult to lubricate, and must have oil grooves passing all the way around.

BALANCING OF ENGINES.—In all steam engines, except very slowly moving pumping engines, it is desirable to balance the rotating parts (crank, crank-pin, and about 60% of connecting-rod) by counterweights. The balancing of reciprocating parts by rotating parts causes unbalanced forces at right angles to the inertia force of the reciprocating parts. For that reason the reciprocating parts of vertical engines are very seldom balanced. In horizontal engines, up to 40% of the reciprocating parts are balanced by rotating parts. According to Radinger, if W = weight of reciprocating parts, lb., and A = piston area, sq. in., $W = 3A$ for low-pressure cylinders and $4A$ for high-pressure, both for strokes below 30 in. From 30 in. up, $W = A \times S$ to $1.8A \times S$, where S = stroke, ft. These are average values which can be somewhat reduced by careful design and excellent materials.

In horizontal-vertical engines (angle engines) a counterweight opposite the crank can be made to balance the reciprocating parts of both engines, with the exception of the deviation caused by the finite length of the connecting-rod. That deviation equals $\times a$, where a = the sine harmonic acceleration and n = the ratio of the connecting-

rod length to crank radius. The deviation can be balanced by counterweights on two shafts, rotating in opposite directions at twice the engine speed.

9. STEAM ENGINE PARTS *

PISTON RINGS.—Concentric rings are preferable to eccentric ones on account of strength, if they are made right. Several makers machine them to fit the cylinder and then give them their spring by peening, or by annealing when sprung open. Others machine them while the ends are sprung closed. Or the rings may be cast out of round, in such shape that they exert uniform pressure when sprung in. (Reinhardt, *Zeit.*, V. d. I., 1901, pages 232–373; also *Auto. Engr.*, London, Sept., 1919.) Rings for large low-pressure cylinders are sectional.

The thickness of eccentric snap rings of ordinary cast iron is $^1/_{32}\,D$ at the heavy side, and $^1/_{50}\,D$ at the split, D being the cylinder diameter, in. They are turned $^1/_{72}\,D$ larger than the cylinder. The pressure between ring and cylinder is 2 lb. per sq. in. Concentric snap rings which are to be sprung over, made of high class cast iron of 40,000 lb. per sq. in. tensile strength, may be made $^1/_{30}\,D$ thick, with a width equal to the thickness. Rings must travel only a fraction of their width into the counterbore, otherwise the steam pressure collapses and breaks them.

PISTONS AND TAIL-RODS.—Most pistons work without tail-rods. The clearance between cylinder and piston should be 0.001 in. per inch of diameter up to 20 in. diam. of cylinder. From 20 in. diam. the clearance should be 0.0005 in. for each additional inch of diameter.

Example.—Piston diameter for a 40-in. cylinder

$$= 40 \text{ in.} - (20 \times 0.001) - (20 \times 0.0005) = (40 - 0.030).$$

The bearing surface of pistons, in horizontal engines, is the surface between piston rings only. Any bearing surface outside of the rings is unbalanced and causes excessive wear. Circular oil grooves are desirable between the rings to prevent oil being squeezed out by the weight of the piston.

Pistons in vertical marine engines frequently have tail-rods, to avoid the use of oil in cylinders, without unduly increasing friction. These vertical cylinders (using saturated steam) are first " run-in " with graphite and beeswax, and thereafter are lubricated only by the water which condenses on the cylinder walls. The rubbing parts acquire a hard smooth " water polished " surface.

Pistons of uniflow engines from 500 Hp. up usually are equipped with tail-rods. These long pistons must be loose to avoid sticking in cylinder. They wobble if not guided at both ends. European builders make the tail-rod serve as a guide only, but American builders prefer to make it also carry a large part of the weight of the piston; for high-temperature steam, due to the difficulty of lubricating the piston, the tail-rod and cross-head slides are made to carry the entire weight of the piston, and the only pressure on the cylinder walls is that of the piston rings.

Low-pressure pistons of large engines, such as blowing engines, are made of the cone type (umbrella type, Swedish type, or marine type), and their weight is carried outside the cylinder, on slides. The clearance between the cylinder and piston should be $^1/_{16}$ in. to $^3/_{32}$ in. all around, because contact between a steel piston and a cast-iron cylinder must be avoided. The piston and tail-rod usually are in one piece, and frequently are made hollow. They are turned about two sets of centers, with the axis of the tail-rod portion at a slight angle to the axis of the piston-rod portion, so that the rod is bent when not loaded, but becomes approximately straight, when supported on the slides at the two ends and loaded with the weight of the piston. Tail-rods also can be turned " cambered " by being loaded with a weight equal to the piston, and fixed at the ends while a revolving tool turns them. The rod also can be sprung in a revolving steady rest in the lathe, and thus be turned cambered.

The clearance between piston and head is ($^1/_{32}$ in. + $^1/_{150}\,S$) for slow-speed engines and ($^1/_{16}$ in. + $^1/_{150}\,S$) for high-speed engines, S being the stroke, in.

CRANK-PINS AND CROSSHEAD-PINS.—In some side-crank engines, in order to obtain the advantages of a cast-iron (semi-steel) wearing surface, the pins are cast integral with the crank disc or crosshead, but are reinforced by a steel pin through the middle. In most modern engines, however, the pins are of forged carbon steel, case-hardened to eliminate wear.

* Only those parts peculiar to steam engines, and not discussed in the general subject of Machine Design (Vol. 3 of this series) are treated under this heading.

STANDARD PROPORTIONS OF STEAM ENGINE PARTS.—Proportions found by long usage to give satisfactory results were compiled by Trooien (*Bull.* Univ. of Wis., No. 252; *Am. Mach.*, April 22, 1909) and are expressed by the formulas given below in standard type. They are correct for a steam pressure of 125 lb. per sq. in. above exhaust. As operating pressures have steadily risen since the time of compilation, Trooien's proportions must be adapted to the increased steam pressure. The formulas given in italics may be used for this purpose.

The wide variation in the " constant " in some of the parts is doubtless due to variations in design which cannot be explained in detail in a brief tabulation.

Notation.—D = diameter of piston. A = area of piston. L = length of stroke. L_c = length of connecting-rod. Hp. = rated horsepower. N = rev. per min. C and K = constants, and d = diam., l = length of element under consideration. H.S. = high-speed engines, L.S. = low-speed or long-stroke engines. All linear dimensions are in inches. p = initial pressure, absolute, minus back-pressure, absolute, lb. per sq. in.; p_1 = initial gage pressure, lb. per sq. in.

Piston-rod.— $d = C\sqrt{DL}$. H.S.: C = 0.15 (min. 0.125, max. 0.187); L.S.: C = 0.114 (min. 0.1, max. 0.156). *Uniflow Horizontal, C = 0.22; vertical, C = 0.17 (150 lb. per sq. in. steam pressure).* $d = C\sqrt{DL}\sqrt{\frac{p}{125}}$. *Occasionally, the diameter of the body of the rod is determined by the character of the fastening, either at crosshead or at piston end.*

Cylinder.—Thickness of wall, $t = CD + 0.28$. C = 0.054 (min. 0.035, max. 0.072). $t = (p_1CD/125) + 0.28$. Thickness of flanges = 1.12 to 1.20 × wall thickness. No. of stud bolts = 0.72 D for H.S.; 0.65 D for L.S. Diam. of stud, $d = 0.04\,D + 0.375 > 3/4$ in. $d = (0.04D\sqrt{p_1/125}) + 0.375$. *Spacing* $< 6\,d$. *Values given by these formulas are very safe, being based on considerations, not only of strength, but of tightness of joint. Other authorities recommend that the nominal stress in the studs should not exceed*

Ratio (C) of Stroke to Cylinder Diameter (L/D).— C = L/D; for N > 200, C = 1.07 (min. 0.82, max. 1.55); for N = 110 to 200, C = 1.36 (min. 1.03, max. 1.88); for N =< 110, C = (L − 8)/D = 1.63 (min. 1.15, max. 2.4). *These ratios do not hold for uniflow engines.*

Piston.—Width of face, f = CD + 1 in., C = 0.32 for H.S. (0.26 for Corliss). Thickness of shell, t = thickness of cylinder wall × 0.6 for H.S. (0.7 for L.S.).

$$f = C(\sqrt[3]{p/125})\,D + 1 \text{ in.}$$

Crosshead.—Area of shoes, sq. in. = CA (C = 0.53 mean; min. 0.37, max. 0.72). Maximum pressure, based on full steam pressure at mid-stroke, 40 lb. per sq. in. H.S. (min. 28, max. 57), 43 lb. per sq. in. Corliss (min. 32, max. 61). *Area* = $CAp/125$.

Crosshead-pin or Wrist-pin.—Diameter = CD (C = 0.25 mean, min. 0.17, max. 0.28). Length of bearing surface = 1.25 × diam. (min. 1.0, max. 1.5) for H.S.; 1.43 × diam. (min. 1.0, max. 1.9) for Corliss. *1.8 for Uniflow. Diam.* = $CD\sqrt{p/125}$.

Connecting-rod.—For high-speed engines, rod of rectangular section, thickness at middle = $C\sqrt{L_c \cdot D}$. (C = 0.073 mean; min. 0.055, max. 0.094). Width at middle = thickness × 2.28 (min. 1.85, max. 3.0). For low-speed engines, rod of circular section, diam. = $C'\sqrt{L_c \cdot D}$. (C′ = 0.092 mean; min. 0.081, max. 0.104). L_c = length, center to center of bearings.

Thickness, H.S., = $C\sqrt{L_c \cdot D}\,\sqrt[4]{p/125}$; *diameter, L.S.* = $C'\sqrt{L_c \cdot D}\,\sqrt[4]{p/125}$. *Area of smallest section (neck), 0.70 × area at middle.*

Modern experience teaches that finished connecting-rods are safer than rough-forged or cast-steel rods, on account of the stress-concentrating action in the latter of nicks, laps, cracks, seams, or other flaws.

Crank-pin.—Diam. for H.S. = CD. (C = 0.40 mean, min. 0.28, max. 0.52); diam. for side-crank Corliss = C′D. (C′ = 0.27 mean; min. 0.21, max. 0.32). *For Uniflow at 150 lb. per sq. in. pressure, C = 0.40.* Length for H.S. = diam. × K. (K = 0.87 mean; min. 0.66, max. 1.25); for Corliss = diam. × K′. (K′ = 1.14 mean; min. 1.0, max. 1.3). *Diam., H.S.* = $CD\sqrt{p/125}$. *Diam., Corliss* = $C'D\sqrt{p/125}$.

Main Bearings of Crank-shaft.—For H.S. center-crank engines, diam. = $C\sqrt[3]{\text{Hp./N}}$. (C = 6.6 mean; min. 5.4, max. 8.2). For Corliss side-crank engines, diam. = $C'\sqrt[3]{\text{Hp./N}} - 0.3$. (C′ = 7.2 mean; min. 6.4, max. 8.0). Length, H.S. = 2.1 × diam.;

length, Corliss = 1.9 × diam. Diam. × length = 0.48A, for H.S.; = 0.60A for Corliss. *In engines carrying armatures of electric generators on the shaft, the diameter of the latter in the main bearing is determined primarily by rigidity requirements. Bearing pressure due to combined weight and steam force seldom exceeds 250 lb. per sq. in. In recent high-speed engines, use of force-feed lubrication, oil filtering and cooling has permitted bearing pressures to be increased and size of crank-pin and crosshead-pin to be reduced.*

Fly-wheel.—Diam., in., = 4.4 × length of stroke (min. 3.4, max. 5.0). Rim speed, for H.S. = 70 ft. per sec. (min. 48, max. 70); for Corliss = 68 ft. per sec. (min. 40, max. 68). *For (weight × diameter²), see Fig. 35. In recent practice, fly-wheel weights have been increased, particularly for driving a.c. generators, so that no trouble is now encountered in synchronizing.*

Weight of Reciprocating Parts.—(Piston + piston-rod + crosshead + $^1/_2$ connecting-rod.) W = weight, pounds = $(D^2/LN^2) \times 2{,}000{,}000$. (min. 1,370,000, max. 3,400,000). Balance weight opposite crank-pin = 0.75W.

This formula gives the weight of reciprocating parts which should not be exceeded, to ensure quiet and safe operation of the engine. Average values of reciprocating weights are given by Unwin as 2 to 3 lb. per sq. in. of piston area, for simple engines; 5 to 7 lb. per sq. in., for compound engines, high-pressure side; 2.1 to 2.5 lb. per sq. in., for low-pressure side.

THICKNESS OF STEAM CYLINDERS.—If t = thickness of cylinder, in.; D = cylinder diam. in.; p = admission pressure, pounds per sq. in.; $t = 0.0004\,Dp + 0.3$. Large horizontal low-pressure cylinders should be made somewhat thicker to prevent them becoming oval on account of their own weight. Recently, special mixtures of cast iron, which resist wear and "growth" under temperature, have been used for cylinders, also nickel-cast-iron for valves.

The following proportions and allowable stresses were taken from various authorities, the selection in each case being that which agrees best with modern conditions.

CYLINDER HEADS.—Cylinder heads are usually either the box form, or the ribbed type. For the latter, the thickness = 1.0 to 1.2 × cylinder wall thickness t; for the box type, thickness of each wall = 0.85 t. Flange thickness = 1.12 to 1.25 t. Ribs are used to give the necessary stiffness, but in the single-wall type, if improperly designed, they may actually weaken the head by cooling strains which result in cracks. Overall depth of head = $0.04\,D\sqrt[3]{p_1} + 1.5$. This dimension varies considerably, depending upon the type and location of valves.

PISTON-ROD ENDS.—Nominal stresses in lb. per sq. in., based on the maximum steam pressure acting on the piston under usual operating conditions, should not exceed the following values, unless the parts are made of high quality steel. At section through key slot, 5500 lb. per sq. in.; shear in key, 5000 lb. per sq. in.; bearing pressure of key, 15,000 lb. per sq. in. Bearing pressure of rod on piston or crosshead should not exceed 15,000 lb. per sq. in., steel on cast iron; 20,000 lb. per sq. in., steel on steel. Tensile stress at root of thread, 7000 lb. per sq. in. Care must be taken to avoid sharp corners, which cause extreme magnification of stress. At a change of section from diameter d_1 to a larger diameter d_2, least radius of fillet = $0.15(d_2 - d_1)$. Fine threads, or threads with rounded corners frequently are used.

CONNECTING-ROD ENDS.—Average tensile stress in side members of connecting-rod ends between 3500 and 6000 lb. per sq. in. The high value is safe, if the end member and bearing brasses are rigid. *Closed-end type:* If d_p = diam. of crank-pin (or wrist-pin, respectively), width of eye = $1.15\,d_p + {}^1/_8$ in. Width of side = $0.30\,d + 0.1$ in. Width at end of eye = $0.33\,d_p + 0.1$. Thickness of sides and end = $0.9\,d_p$. These proportions are based on correctly-selected dimensions of crank-pin. *Marine type:* Allowable stress in bolts, based on nominal diameter d_b, is $\left(9000 - \dfrac{3200}{d_b} - \dfrac{600}{d_b^2}\right)$. One bolt is taken as carrying 60% of the total load. In rods with forked ends (4 bolts), the allowable stress is 20% less. Stresses 15% to 20% higher are permissible with high-class material, or in bolts whose average cross-section is reduced below the section at root of the thread. For high-speed engines, connecting-rod bolts are now (1935) made of heat treated alloy steel of high tensile strength, as are valve rods.

ECCENTRIC.—Diameter depends on shaft diameter and on throw. Width, in., = (average force, lb. × r.p.m.) ÷ 70,000. So-called "eccentric strap" should be designed as a beam rather than as a strap, to avoid locally concentrated pressures.

ENGINE FRAMES are, almost without exception, of cast iron and of the bored guide type. The moving parts should be wholly enclosed by the frame, leaving covered holes which can be opened for inspection. It is desirable to have a partition near the

stuffing box for separating the drips from lubricating oil. Frames must take up the engine forces without harmful deformations.

The average static stress is 600 to 800 lb. per sq. in. in designs where bending moments can be kept small. Where the moments are likely to be large, the average tensile stress in the smallest section should not exceed 350 lb. per sq. in. In general, rigidity rather than strength determines the dimensions, and the permissible stress becomes less in large engines.

Fig. 97 is a Corliss engine frame, and represents the typical American design for horizontal engines. Fig. 98 is a typical European design, in which the oil shield is entirely separate from the bedplate. Fig. 99 is a vertical power engine frame, and Fig. 100 a

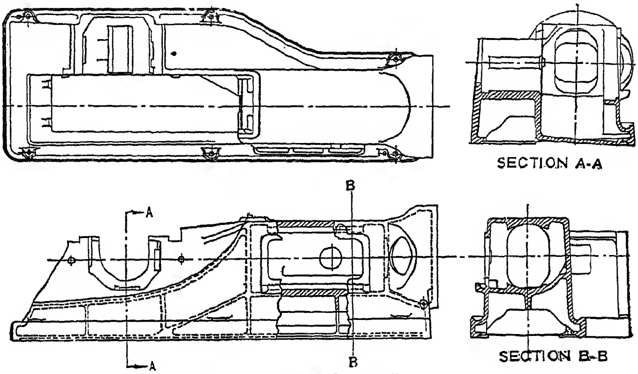


Fig. 97. Corliss Engine Frame

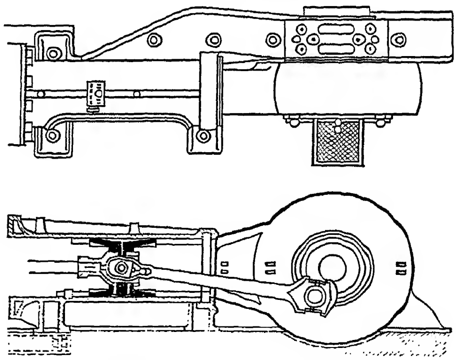


Fig. 98. European Engine Frame

marine engine frame. In the latter, the two front rods can be removed, leaving the cylinder supported on the back frame, so that the crank-shaft easily can be removed and replaced.

10. LUBRICATION

The lubrication of reciprocating engines divides itself into three types: 1. Drip lubrication; 2. Stream lubrication; 3. Forced lubrication. If oil is allowed to be splashed around or to run into the foundation, only drop-by-drop lubrication is possible. Types 2 and 3 require catching and filtration of the oil. Some engines are equipped with self-contained lubricating systems, using no other filter than a strainer in the crank case.

Wear is inevitable in such a system. Wear can be wholly avoided by the use of stream lubrication or forced lubrication in conjunction with a close-mesh cloth filter. See Vol. 3 of this book for additional information.

On engines with complicated valve gears, such as Corliss or poppet-valve gears, automatic lubrication is very difficult. In purchasing engines, the lubrication problem should be carefully considered.

In engines operating with steam at high temperature (either at high pressure or highly superheated), proper lubrication of cylinders is difficult. In the present state of the art (1935), the limiting steam temperature is about 700 to 750° F. although one installation is stated to operate at 800° F. For these conditions, the "hydrostatic" lubricator is replaced by mechanically-operated lubricators injecting oil into the steam line near the throttle, but closer to the inlet valves as the steam temperature increases. For very high temperatures, oil is injected directly above each inlet valve and also into the cylinder itself, at points about 1/3 of stroke from each end and at top or at several points around the circumference, with an auxiliary feed to the piston-rod packing case. Injection into the cylinder increases the oil consumption but is especially necessary in designs of uniflow engines in which the jackets trap out the oil in the steam. For extreme temperatures the

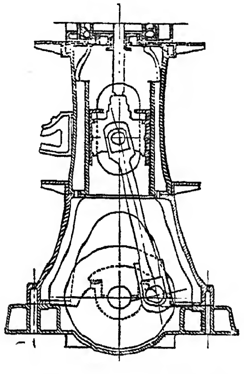


FIG. 99. Vertical Power Engine Frame

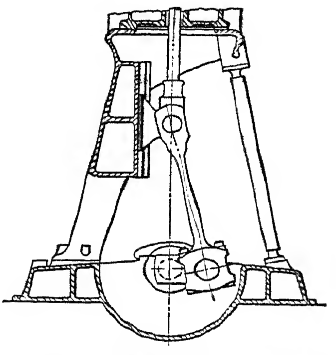


FIG. 100. Vertical Marine Engine Frame

injection into the cylinder is *timed* to occur when the piston reaches a given point in the stroke.

The lubrication of steam cylinders and sliding valves (including piston valves and rocking valves) never can be perfect, as the rubbing surfaces cannot be completely separated by a film of oil. The contact will be partly metallic, and wear cannot be entirely avoided.

In operation with saturated steam, the film of water on the walls washes off any mineral oils, unless they are blended with grease, usually 5% tallow. With superheated steam, the oils used are straight mineral oils (uncompounded), or else have only a very small amount of animal oil.

The cylinder oil consumption ranges from 0.1 pint to 4 pints per million sq. ft. of surface swept over by the piston (perimeter of piston × distance traveled), with an average of 0.8 pint per million sq. ft. The average consumption per million sq. ft. will be less in large poppet-valve engines than in small slide-valve engines. Even in engines of the same type and size, there is a wide variation in oil consumption, due partly to waste, to different methods of handling, and in steam and operating conditions, but chiefly to the difference in texture of cylinder and piston castings. In some cylinders, oil adheres strongly to the walls, in others it does not cling to the surface but is wiped off by the piston and by the moisture, and requires continual replacement. See also *Power*, Feb. 15, 1910.

In engines using high-temperature steam (700° to 800° F.), the cylinder oil actually is vaporized by the heat of the steam, but if the viscosity is high enough, some oil remains on the surfaces until a fresh supply is injected. If *too much* oil is supplied, it "cracks" and forms carbon deposits which cause increased heating and wear. In large engines, with steam at 700° F., 7 drops of cylinder oil per minute to each end of each cylinder.

6 drops in each valve, and 5 drops in each packing case have been found to give good lubrication while avoiding carbon.

From carefully recorded data furnished by the Vacuum Oil Co., the average cylinder oil consumption for usual steam conditions is found to vary with the size of engine according to the equation

$$Q = K\sqrt{\text{Rated Hp.}},$$

where Q = gallons per year of 3000 hours. The factor K ranges from a minimum of 1.2 to a maximum of 12, with an average of 4.8 in power plants, and from a minimum of 8 to a maximum of 53, with an average of 24 in steel plants where the consumption always is high.

The consumption of bearing oil varies over an even wider range, but averages 1 1/2 to 2 times the cylinder oil consumption.

Marine engines using superheated steam at 700° F., after initial treatment as described on p. 7–43, have been lubricated by injecting a spray of cold water into the steam near the throttle, together with a very small amount of oil.

SEPARATION OF OIL FROM EXHAUST STEAM.—Where exhaust steam from reciprocating engines is to be used for heating feedwater or in process heating, the oil carried in the steam must be removed. The oil may be removed either before or after condensation of the steam. In the former case, the steam is passed through oil separators, usually of the baffle type but occasionally of the centrifugal type; or where extreme freedom from oil is necessary, it is passed through a baffle separator in series with a closed feedwater heater, beyond which a cold water spray is injected into the exhaust pipe, and is finally passed through a second separator. To separate the oil from the water after condensation, skimming tanks, or sponge or towel filters may be used. The filters must be cleaned regularly.

Compounded oils are more difficult to remove than straight mineral oils.

On Mississippi River steamboats, aluminum sulphate and caustic soda are used to coagulate the oil, after which it is removed from the water by sand filters. In Europe, charcoal oil-separators have been used successfully.

11. COST OF ENGINES

Prices vary with the size, type of engine, and market conditions. Average costs in 1931, when basic pig iron cost $15.50 at Pittsburgh and common labor 39 cents per hr., are shown in Fig. 101. For comparison, the average prices in 1915, when basic pig iron cost $12.50 per ton at Pittsburgh and common labor 20 cents per hour, are also shown, in dotted lines.

These prices do not include generator or piping. The rated I.Hp. of an engine is usually 1 2/3 times the rated kw. of the generator which it is to drive. Compound engines average about 70% higher in cost than simple engines of the same type.

The cost of an engine also can be derived from the pound prices and from the weights of engines as shown in Fig. 32, bearing in mind, that, in general, the weight of the fly-wheel varies from about 25% of the weight of the engine without fly-wheel in 100-Hp. engines, to 30% in 800-Hp. engines. (In late practice, the fly-wheels of engines driving alternators are made considerably heavier.)

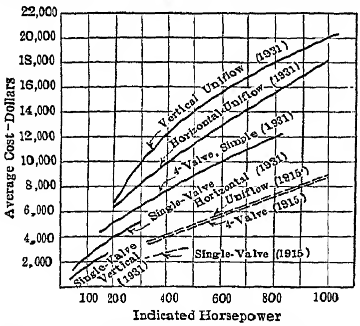


Fig. 101. Comparison of Engine Prices, Not Including Generator

The pound price, cents per pound, in 1931, averaged:

For vertical uniflow engines..........	18.4 + 640/rated I.Hp.
For horizontal uniflow engines........	9.1 + 1370/rated I.Hp.
For 4-valve and single-valve engines....	8.9 + 1370/rated I.Hp.

The cost of installation, varying with local conditions, averages about 17% of engine cost.

DEPRECIATION OF ENGINES ranges from 3% to 5% per year with slow-speed engines, to 5% to 10% with high-speed engines.

12. SELECTION OF TYPE OF ENGINE

The first question to be decided in selecting an engine usually is, whether to install a cheap engine of simple form or a more expensive highly developed type. For small sizes, driving auxiliaries, etc., where steam economy is not so important as reliable operation over long periods without attention, the simplest form of engine, such as the single-valve type, is installed, usually of the vertical enclosed, self-oiling type. Vertical single-acting engines are much used for driving stokers, as they need very little attention (see p. 7–29, Quietness of Operation). For the main engines of power plants, simplicity usually is less important than economy, and the 4-valve or uniflow engines are installed, except where more steam is needed for heating than for power.

Non-condensing engines are installed where the exhaust steam can be used for heating during a large part of the year, as in the power plants of office buildings, hotels, and hospitals. For such use, they should be of the most economical type, to save fuel in those periods in which all or part of the exhaust steam is not required for heating. If the steam needed for heating is considerably less than required for power, the bleeder or extraction type of engine can be used. This works condensing but takes steam for heating from the receiver, if of the compound type. In this case the high-pressure cylinder is made larger in proportion to the low-pressure cylinder than in the usual compound; or two smaller engines can be installed, one working non-condensing and furnishing steam for heating, and the other condensing, furnishing additional power. Heating systems now installed produce a back-pressure at the engine of 1 to 10 lb. per sq. in. above atmosphere; the recent tendency is to decrease this back pressure, due to increased use of vacuum heating systems in which the back pressure does not exceed 1 lb. per sq. in. gage.

The compound engine has been displaced by the uniflow, except, *a*, where the load is quite constant, and *b*, for compressors and blowing engines. For the latter, the distribution of forces during the stroke, in the uniflow type, is particularly unsuited to the requirements of the compressor, and as the load is reasonably constant, compound engines are almost universally used for this service. On account of its flat economy-load curve (p. 7–25) the uniflow engine is especially suited to the average plant where the load varies widely from day to day and throughout the day. In office building and hotel power plants, uniflow engines now are installed almost exclusively, usually operating with superheated steam. For high back pressure (ranging in process work from 20 to as high as 75 lb. per sq. in. above atmosphere) the uniflow is not suitable, on account of its long compression, unless the initial steam pressure is raised in proportion, and simple 4-valve or single valve engines usually are preferred. The compound principle is used in small marine engines and in large direct-acting boiler feed pumps. In locomotives, high superheat in simple engines has displaced the compound engine.

Triple- and quadruple-expansion engines are limited almost entirely to marine engines and pumping engines with absolutely constant load.

Slow-speed engines have longer life, are more reliable and usually somewhat more economical than high-speed engines, but the latter usually cost less for the unit including generator, and take up less floor space. For selection of piston speed and rotative speed, see p. 7-14.

Vertical engines have not been popular in the U. S., except in comparatively small sizes, where the handling of crosshead and connecting-rod in case of repairs causes no hardship. Large vertical engines are inaccessible to crane service, unless the steam cylinder is taken off the frame or the lower head can be pulled up through the cylinder. Tall vertical engines must have very broad bed plates and rigid housings, because of horizontal vibrations which are caused by pulsations in the steam pipes. Vertical engines are disliked by operators on account of the frequent climbing of stairs necessary in attendance and oiling. They are, however, necessary in marine work and in other places where the floor space is very limited. Recently, for engines driving generators, the trend has been to high-speed multi-cylinder vertical engines, on account of great reduction of floor space occupied, without excessive headroom requirement.

Horizontal engines are preferred by all operators, on account of their accessibility for attendance and repair. Side-crank engines are preferred to center-crank engines on account of the care required to keep three bearings in line. In general, single-crank engines are preferred to multi-crank engines, because of the smaller number of parts needing attention, but in many cases greater uniformity of torque is needed or there is a necessity for quick starting from any position, which requires more than one crank. Locomotives, marine engines, automotive engines, rolling mill reversing engines, etc., all require more than one crank. Reciprocating pumping engines are built with several

cranks, to give more uniform water delivery. The reasons for the trend toward multi-crank vertical engines already have been mentioned.

Angle engines occasionally are installed where space is limited or vibrations must be reduced to a minimum.

Corliss valves should not be used with superheated or high-pressure steam. Inlet valves of the poppet or drop piston types are used instead, in the h.p. cylinder.

For selection of valve gear for steam engines, see pp. 7–30 to 7–41.

13. TESTING OF STEAM ENGINES

Tests of steam engines are for the purpose of verifying guarantees given by the manufacturer of the engine or to determine its performance as regards: 1. Thermal-economy characteristics (heat value or steam rate or both). 2. Capacity in indicated horsepower, kilowatt output, or brake horsepower. Tests of engines also are made to determine special data or to verify particular guarantees.

All steam engine tests should be made in accordance with the A.S.M.E. Test Code for Reciprocating Steam Engines. An abstract of the 1935 Code is given on p. 16–26.

Section 8

THE STEAM TURBINE

By A. G. Christie

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THE STEAM TURBINE

By A. G. Christie

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A steam turbine is a form of heat-engine in which two distinct changes of energy take place. The available heat energy of the steam first is converted into kinetic energy by the expansion of the steam in a suitably shaped passage, or nozzle, from which it issues as a jet. A portion of this kinetic energy then is converted into mechanical energy by (1) directing the jet, at a proper angle, against curved blades mounted on a revolving disc or cylinder; or (2) by the reaction of the jet itself if the expanding channel can revolve.

The pressure on the blades, causing rotary motion, is purely dynamical and is due solely to the change of momentum of the steam jet during its passage through these blades.

Radiation and condensation losses in turbines are of a small order. Leakage losses occur through clearances over the ends of Parsons blades and through labyrinths and

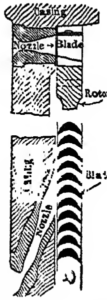


Fig. 1. Impulse

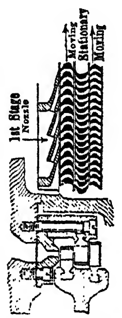


Fig. 2. Curtis.

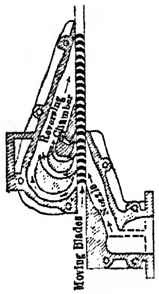


Fig. 3. Re-entry

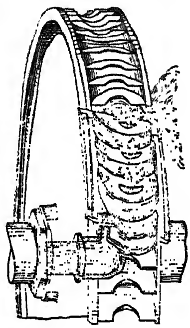


Fig. 4. Helical Flow

Types of Steam Turbines

glands. Friction of high-velocity steam jets through passages and across blades, together with the friction losses of high-speed revolving discs and idle blades in steam-filled chambers, have considerable effect on the ultimate efficiency of the turbine. The requirements of safe mechanical construction must be considered.

1. TYPES OF TURBINES

THE SIMPLE-IMPULSE TURBINE consists essentially of one or more nozzles supplied with high-pressure steam, with the discharge jet impinging, at a suitable angle, on a single row of blades on a revolving disc. The steam expands in the nozzle to exhaust pressure, its velocity increasing during expansion. The resulting kinetic energy is partly converted into mechanical energy during the passage of steam across the blades. The commonest type of simple-impulse turbine is the **DeLaval**, shown diagrammatically in Fig. 1.

For best efficiency of the simple-impulse turbine the ratio ρ of wheel speed to steam speed of the jet issuing from the nozzle ought to be about 0.53. This ratio can be obtained only with heat drops not exceeding 55 B.t.u. with the usual allowable blade speeds, or 100 B.t.u. for geared units. These drops may obtain when high back pressures are employed. Usually the heat drop exceeds these amounts, leading to a decrease in speed ratio ρ, with resultant lower efficiency. Simple impulse turbines usually are built for small output, although they have been used in units up to 3000 Hp.

VELOCITY-COMPOUNDED TURBINES utilize the high-velocity jet from the nozzle more efficiently than does the simple impulse turbine. The steam, after crossing the first row of moving blades, flows through stationary curved blades or passages. These reverse the direction of the jet and redirect it against a second row of moving blades. This reversal and re-impingement may occur several times before the steam finally escapes to the outlet. When velocity-compounding consists of several rows of revolving blades mounted on the same or parallel wheels, with intermediate stationary reversing blades, the form results in the well-known Curtis stage, Fig. 2. This also is called a *velocity stage*. In another form, the reversing passages redirect the steam back against the same row of blades that the jet first crossed. This is known as the *re-entry* type of turbine, Fig. 3.

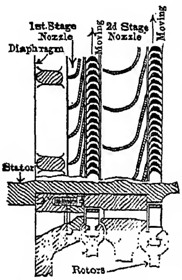


FIG. 5. Multi-stage Impulse Turbine

HELICAL FLOW TURBINES, Fig. 4, employ a forged steel wheel. Semi-circular buckets are milled in the circumferential rim at an angle of about 30° with the tangent. The steam expands to exhaust pressure in the nozzle which directs the steam into one side of the semi-circular bucket. The steam gives up part of its energy in its first reversal through 180° in the moving bucket but still retains considerable velocity. It then passes to a reversing chamber which redirects the steam into the wheel buckets. This is repeated several times in additional reversing chambers until as much energy as possible is delivered to the turbine wheel. The whole operation occurs in a single wheel which may be provided with several groups of nozzles and reversing chambers.

In all of these velocity-compounded turbines, part of the kinetic energy is absorbed each time the steam crosses a revolving blade or bucket. More work is thus obtained from a given quantity of steam flowing than in a simple-impulse turbine of the same blade speed. The friction losses, however, increase with the addition of the reversing blades or chambers. The various forms of velocity compounding are used for small steam turbines, particularly for non-condensing auxiliary services, such as driving pumps, blowers, exciters, stokers, etc. The Curtis stage forms one element of compound units.

A STAGE in steam turbines is a term used to signify that part of a machine in which a drop of pressure occurs with the generation of kinetic energy, together with such succeeding blades and passages where no further drop of pressure occurs. A stage, therefore, includes the nozzles, the moving blades and the reversing blades, or chambers, when used. Each of the preceding units consists of a "single stage."

THE MULTI-STAGE IMPULSE TURBINE consists of a series of simple-impulse turbines built on the same shaft. Each such simple turbine forms a stage. It is so designed that the steam expands through only a portion of the total pressure range in the nozzles of the first stage. On leaving the blades of the first wheel, the steam enters the second-stage nozzles, carried in the diaphragm forming the wall of the stage, and expands through a further pressure drop. This jet impinges on a second row of revolving blades. The operation is repeated in every stage until the steam is fully expanded in the final stage to the exhaust pressure.

With this construction, Fig. 5, it is possible to maintain the most efficient ratio of wheel

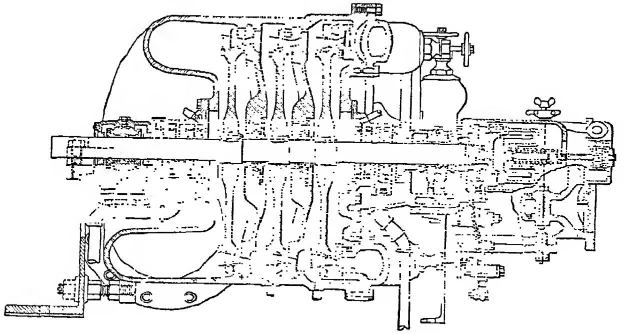


FIG. 6. Section of 3-stage Curtis Turbine

speed to steam speed by properly apportioning the total heat drop, from initial conditions to final pressure, between a suitable number of stages. Hence high efficiencies are possible with many of these turbines.

The total heat drop from initial conditions to final pressure is divided either equally or in a particular empirical manner, fixed by the manufacturer's construction, between the various stages. Steam speeds are calculated and wheel speeds selected to give the value ρ, *i.e.*, the ratio of wheel to steam speed, as high as commercially possible. Low-cost, few-stage units have low values of ρ and low efficiency. High-efficiency machines have high values of ρ and many stages. Multi-stage impulse turbines can be built for the largest output desired.

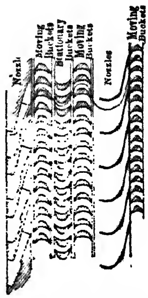


FIG. 7. Compound Impulse Turbine

Another form of turbine comprises a series of velocity-compounded or Curtis stages, known as 2-stage Curtis, 3-stage Curtis, etc. This construction is used on some of the smaller turbines. Fig. 6 shows a 3-stage Curtis turbine built by the General Electric Co.

The theoretically high efficiency of a small multi-stage impulse turbine with few stages is somewhat offset by the large whirling and friction losses of the discs and blades of the first stages, which revolve in dense media of high-pressure steam, and usually have nozzles delivering steam over only a portion of the periphery.

A velocity-compounded or Curtis stage often is substituted for several of the first simple stages in small and medium sized multi-stage impulse turbines. The resultant compound-impulse turbine, Fig. 7, is shorter, more compact, costs less, and

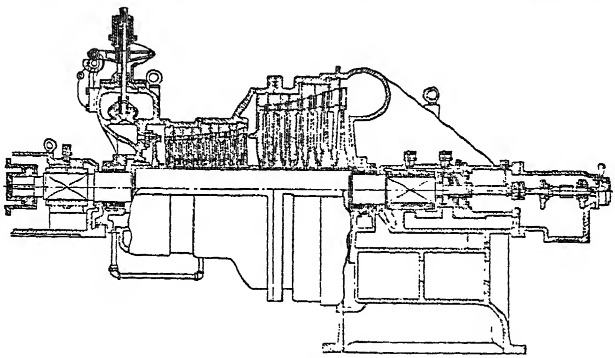


FIG. 8. DeLaval Multi-stage Turbine

is practically as efficient as the straight multi-stage impulse turbine of the same capacity. Fig. 8 shows such a compound unit built by the DeLaval Steam Turbine Co.

In impulse turbines the pressure on the two sides of the blade is substantially the same. All pressure drops occur in the nozzles between stages. Radial clearances of impulse blades, therefore, may be quite liberal, but suitable packing must be provided where the shaft passes through the diaphragm between stages.

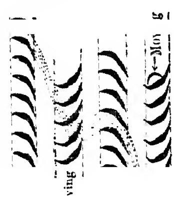


FIG. 9. Impulse-Reaction Turbine

No distinct line exists between impulse and so-called reaction turbines, as the majority of so-called impulse turbines have more or less reaction in the last stages. The term "impulse" applies to stages with no reaction or with only a small amount of reaction, say 10 to 15%, *i.e.*, 10 to 15% of the total heat drop per stage is expended, with a small pressure drop, in passing through the moving blade, thereby increasing the relative outlet velocity.

THE IMPULSE AND REACTION TURBINE, typified by the **Parsons turbine,** consists of one or more revolving drums placed inside a cylinder, with rows of blades attached alternately to the stationary cylinder and to the revolving drum. This is shown diagrammatically in Fig. 9. The passages between blades of all rows are designed to form contracting orifices; hence there is a drop in pressure and a subsequent expansion

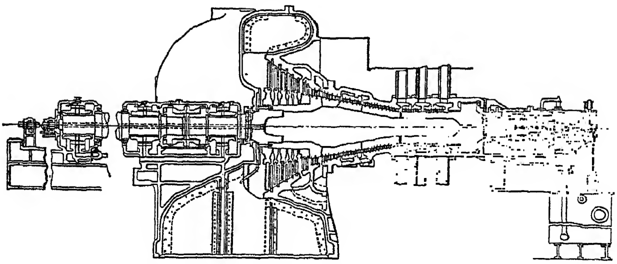


FIG. 10. Parsons Turbine (Allis-Chalmers)

of the steam through every row of blades. A row of stationary blades and its following row of revolving blades is known as a *stage* in a Parsons turbine. Fig. 10 shows a Parsons type of turbine as built by Allis-Chalmers Mfg. Co.

For best efficiency in a Parsons turbine, blade speed should be about 90% of steam speed. To obtain this efficiency, low steam speeds and many moving rows of blades, and a consequently long spindle must be used. In general, lower ratios of blade to steam speed prove more practical and commercial, giving fewer rows of blades and shorter turbines. In Parsons turbines, as in multi-stage turbines, clearances in the inter-stage seal must be small to prevent excessive steam leakage. With standard Parsons blading this necessitates small radial clearances over the ends of the stationary and moving blades. Many Parsons turbines are fitted with shrouds to provide axial, rather than radial, clearance which is accurately controlled by axial adjustment of the thrust bearing (end tightening).

On small Parsons turbines using high steam pressure, it is difficult to make sufficiently short blades for the first stages, and a Curtis stage often is substituted for several reaction stages at the high-pressure end. The resulting combination machine, Fig. 11, known as the Curtis-Parsons, is shorter and somewhat cheaper to build than the standard Par-

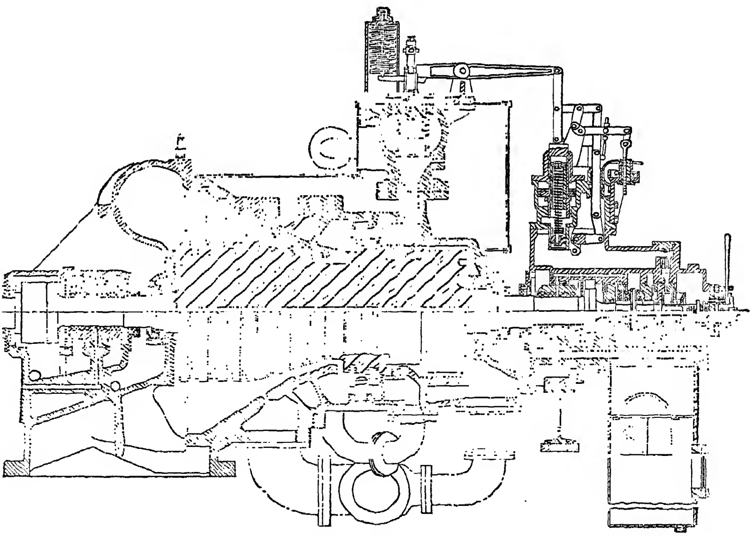


FIG. 11. Curtis-Parsons Turbine (Westinghouse)

sons. If a large pressure drop is allowed in the first nozzles, the temperatures in the casing are moderate and distortion troubles are lessened.

The Ljungstrom Double Rotation Turbine, Fig. 12, is a radial flow unit of the reaction type. It consists of intermeshed sets of blading, each rotating in the opposite direction. Two generators, tied together electrically, are required. The relative velocity of the two sets of blades is twice that obtainable with a fixed casing and a single revolving spindle. This construction leads to high capacity and high efficiency for a given diameter of blade ring. The unit is compact and usually is placed above, and supported by, its surface condenser. Its construction permits the use of high temperature steam and quick starting. The simple radial-flow design is applicable to back-pressure, non-condensing, and the smaller sizes of condensing units. Large condensing units have double flow axial blading in the exhaust end, as in Fig. 12, which is a 50,000 kw. unit built by S.T.A.L.

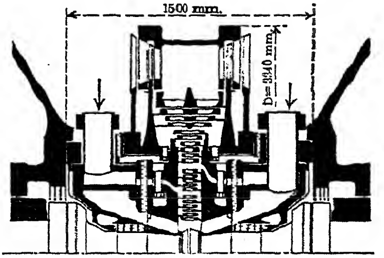


Fig. 12. Section of Blading of Ljungstrom Turbine

APPLICATIONS OF STEAM TURBINES.—The principal application of large turbines has been to drive alternating-current generators, to which they are connected by a solid or flexible coupling. Turbines of various sizes, also direct-connected, drive centrifugal pumps, small direct-current generators, fans, blowers, etc.

For a given blade speed, turbines with small diameters, operating at high r.p.m., are most economical. Such units may be connected through double helical reduction gearing to moderate speed machinery such as fans, propeller shafts, compressors, stokers, pumps, etc. Both driving and driven units then may operate under best conditions, and first cost of equipment is lower. In some cases, geared turbines have operated rope and belt drives for factory machinery, as cotton mills, rolling mills, etc. Geared turbines also have been used to drive locomotives.

Single-cylinder turbines are used (1935) in capacities up to 80,000 kw. at 1800 r.p.m. and 15,000 kw. at 3600 r.p.m. Fig. 13 shows an 80,000-kw. unit built by General Electric Co. Fig. 14 shows a 75,000-kw. unit built by Westinghouse Elec. and Mfg. Co. Unidirectional, double-flow turbines in one cylinder provide large areas for the last blade rows with low tip speeds. Larger units are multi-cylinder, of either the tandem or cross-compound types, with double-flow low-pressure cylinders to obtain the desired low-pressure blade area. Compound units also are used with reduction gearing in marine and other service. Vertical compound units have been installed to conserve engine-room space.

Impulse vs. Reaction Turbines.—Kraft (The Modern Steam Turbine) states that the impulse type is best suited for use in the high-pressure region and for small steam quanti-

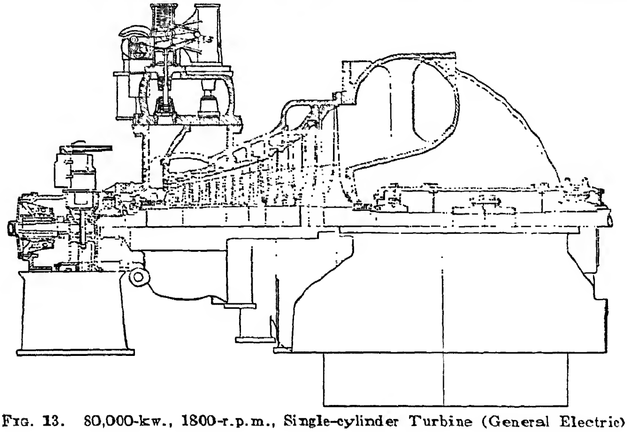


Fig. 13. 80,000-kw., 1800-r.p.m., Single-cylinder Turbine (General Electric)

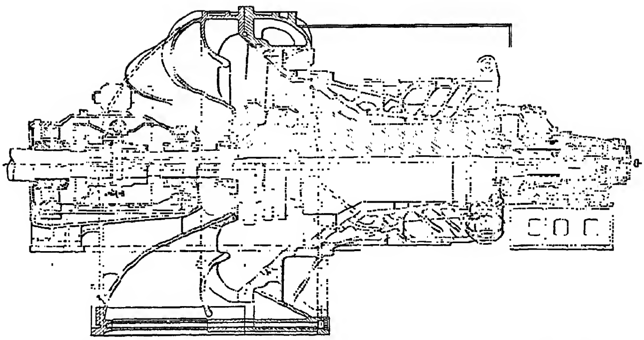


FIG. 14. 75,000-kw., 1800-r.p.m., Single-cylinder Turbine (Westinghouse)

ties. The reaction type has advantages for the lower pressure region and where large volumes of steam must be handled. Practice is tending toward the use of discs for low-pressure reaction blading at high blade speeds. In commercial practice, there is little difference in efficiency between the two types.

European builders have developed so-called *self-contained sets*. These consist of a turbine mounted directly on its condenser which may form the base-plate of the unit. The turbine often is connected to the generator through gearing. The exciter and condenser auxiliaries are driven direct from the turbine, through gearing or by connection to the main shaft. Erection costs are low and performance is good, as all auxiliary power is obtained at the steam rate of the main unit.

STEAM CONDITIONS.—Turbines can be built to operate at any steam pressure from a few inches of vacuum up to the highest steam pressures available. Central station pressures (1935) are generally in the range of 300, 400, 600, and 1200 lb. per sq. in. Industrial turbines operate at all pressures from 100 lb. to 1200 lb. per sq. in. Steam temperatures up to 1000° F. are in use, though usual practice is 700 to 850° F., with a trend towards higher temperatures. Vacuum of 29 in. can be maintained with 57° F. cooling water. With 70° F. cooling water a vacuum of 28.5 in. can be obtained.

OPERATING CHARACTERISTICS.—In the *straight condensing turbine* all the steam that enters the throttle, excepting some gland leakage in certain types, passes completely through the turbine to the condenser, in which a vacuum is maintained. Fig. 15 shows a straight condensing turbine built by the Elliott Co.

Extraction or Regenerative Turbines.—Steam is withdrawn from the turbine at intermediate stages of expansion and used to preheat the feedwater in open or closed heaters. The use of extraction turbines leads to higher station economy and they are preferred to straight condensing units. Since steam is extracted at various points, the quantity of exhaust steam decreases, and the size of condenser is smaller than otherwise. This likewise permits an increase in the rating of a given casing.

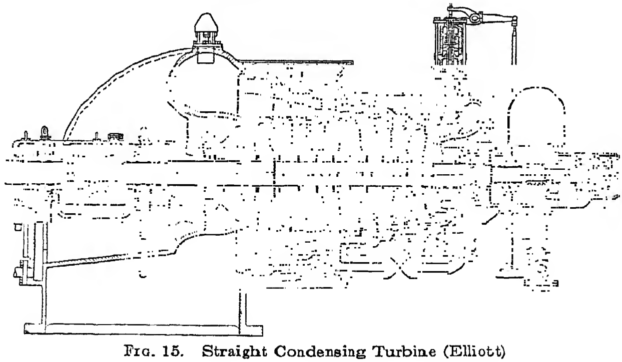


FIG. 15. Straight Condensing Turbine (Elliott)

Expansion at high efficiency from high pressures, with moderate steam temperature, to a high vacuum results in excessive moisture in the low-pressure stages. To avoid this, and to improve station economy, steam can be withdrawn from the turbine at an intermediate stage, reheated by flue gases, live steam, or other hot fluid, and then returned to the turbine. This results in a *reheating turbine*. Reheating turbines usually have extraction heaters to preheat the feedwater. This type is known as a *regenerative-reheating turbine*.

Non-condensing or High-pressure Turbines exhaust at atmospheric pressure or above. They form the high-pressure units of reheating turbines. They also exhaust to heating systems, to industrial processes, or simply to atmosphere in a few cases.

While many small non-condensing turbines for auxiliary use have comparatively low efficiencies, those used for power generation can be designed for higher engine efficiencies than the same size of condensing unit. Steam volumes are greater, leading to longer blades and lower percentage of leakage. Also the heat drop is confined almost wholly to the superheat field. See R. T. Luce, *Mech. Engg.*, April, 1931, p. 276.

In the *bleeder turbine*, steam is extracted at one or more intermediate stages, often at comparatively high pressures, for industrial use. Frequently the pressures at these bleeder stages must be maintained constant by a special regulating device forming a part of the turbine. The steam not withdrawn at the bleeder points expands through the remainder of the turbine to exhaust or to the condenser pressure. This type of turbine may operate at a given load with all the steam that enters at the throttle, flowing out of the bleeders, or with all throttle steam passing to the condenser when no steam is bled, or with any condition intermediate between these extremes. (See Burge and

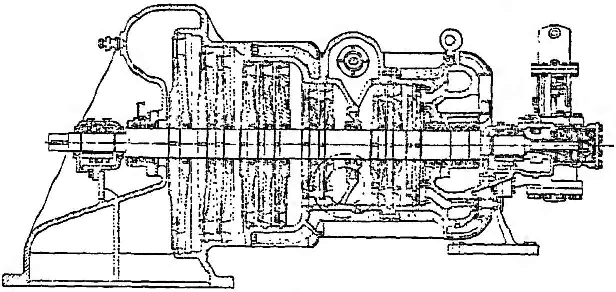


Fig. 16. Bleeder Turbine (Moore)

Chittenden, Steam Extraction Turbines, *The Engr.*, Dec. 19, 1924.) Fig. 16 shows a bleeder turbine built by Moore Turbine Corp.

LOW-PRESSURE TURBINES formerly were those that received steam at or about atmospheric pressure and expanded it to vacuum conditions. They were employed extensively to utilize the exhaust of non-condensing reciprocating engines or low-pressure steam from other sources. Certain of these types still are built, but the term "low-pressure turbine" now is frequently applied to the second cylinder of compound and of reheating turbines, where the inlet pressure to the low-pressure cylinder may be well above atmosphere. Certain of these units are made double flow to provide sufficient exhaust areas at the last blade rows.

MIXED-PRESSURE TURBINES are designed to receive steam at two or more pressures. Frequently such units serve as low-pressure turbines, receiving the exhaust of a hoisting, rolling mill or other engine which operates intermittently. These turbines have a high-pressure section to which live steam is admitted when there is a deficiency of low-pressure steam. In certain cases the unit is essentially a high-pressure turbine, receiving only such small quantities of low-pressure steam as are available, for instance, from non-condensing auxiliaries.

ACCUMULATOR TURBINES have been developed for service with steam accumulators, both of the low-pressure and of the Ruths or high-pressure types. The accumulators are large cylinders filled to 90% of capacity with water. During the charging period the pressure in the accumulator increases and the water absorbs the latent heat of the steam supplied. On discharge, this stored heat evaporates a portion of the water as the pressure falls. The pressure of steam from the accumulator falls through the whole dis-

(*Continued on page 12*)

Table 1.—Comparison of Turbine Characteristics

Classification	Type	Characteristics: Physical	Characteristics: Operating	Relative Advantages	Applications	Manufacturers
Impulse	Simple Impulse	1 nozzle or set of nozzles. Single disc with one row of blades. One passage of steam across blades. Simple casing construction.	Little heating necessary—quick starting. Clearances usually ample. Simple governing systems.	Low first cost. Comparatively small size. Heat drop limited to 100 B.t.u. for high efficiency; larger heat drops lead to decreased efficiency. Small floor space.	Certain high- or low-pressure units with small heat drops. Auxiliary drives. Turbo-blowers and compressors.	DeLaval Steam Turbine Co. Carling Turbine Blower Co.
Impulse	Simple Velocity Stage or Curtis	1 nozzle or set of nozzles. Single stage with two or more rows of revolving blades and necessary intermediate reversing blades. One passage of steam across each blade row.	Moderate wheel speeds. Large pressure drop in nozzle. Large end clearance for blades. Same pressure throughout stage. Simple governing systems. Can use comparatively large pressure drop.	Low first cost. Large power in small size. Moderate efficiency. Can utilize comparatively high heat drops. Strongly built. Small floor space.	Auxiliary drives, as stokers, pumps, exciters, small d.c. and a.c. generators, fans, blowers, etc. For emergency units requiring quick starting, as reserve boiler feed pumps, house turbines, etc. Non-condensing units for industrial plants in connection with process.	General Electric Co. DeLaval Steam Turbine Co. Moore Steam Turbine Corp. Allis-Chalmers Mfg. Co. Murray Iron Works Co. Elliott Co. Terry Steam Turbine Co. Coppus Engineering Corp. Carling Turbine Blower Co. Westinghouse Electric & Mfg. Co.
Impulse	Re-entry	1 nozzle or set of nozzles. Single row of buckets or blades. Reversing chambers to re-direct steam on buckets or blades one or more times. Simple casing construction.	Usually moderate wheel speeds. High wheel speeds used in some geared sets. Can employ comparatively large pressure drop. Simple governing devices.	Low first cost. May secure relatively good efficiency. Ruggedly built. Small floor space.	Auxiliary drives of all kinds. Geared sets. Emergency reserve units for quick starting. Paper mill drives. Small marine geared propelling units for tugs, etc.	Westinghouse Elec. & Mfg. Co. DeLaval Steam Turbine Co. Dean Hill Pump Co. Babcock-Wilcox-GoldieMcCulloch (Galt. Ont. Canada). L. J. Wing Mfg. Co.
Impulse	Helical Flow					Terry Steam Turbine Co. B. F. Sturtevant Co. DeLaval Steam Turbine Co. D. E. Whiton Machine Co.
Impulse	Multi-Stage Impulse or Rateau	Several simple impulse wheels in series, separated by diaphragms carrying nozzles or orifices. Usually many stages. Large clearances at ends of blades; small side clearance at nozzle.	Small heat drop per stage. Hence most efficient ratio of wheel speed to steam speed may be had. Fine clearances necessary at diaphragm glands. Throttling or nozzle governing, latter preferred.	High efficiencies possible. High blade speeds needed in last rows of large condensing units. Built in large as well as small sizes. Blade limitations in exhaust end of large units. Requires careful heating before starting.	Turbo-generator drive. Large auxiliary drive. Marine propulsion. Centrifugal pumping sets. Turbo-blowers and compressors. Bleeder and extraction units. Mill drives.	General Electric Co. Elliott Co. DeLaval Steam Turbine Co. Terry Steam Turbine Co. Moore Steam Turbine Corp.
Impulse	Velocity-compounded or Multi-stage Curtis	Series of simple Curtis stages separated by diaphragms carrying nozzles. Large end clearances on blades; small side clearance at nozzle. Relatively few stages.	Relatively large pressure drop per stage. Fine clearances at diaphragm glands. Can be started quickly. Few operating troubles. Usually nozzle governing.	Small floor space. Moderate efficiency. Rugged construction fit for hard service.	Non-condensing service with large auxiliaries, house turbines, exciters, etc. Condensing turbo-generator sets. Marine reversing turbines. Bleeder and extraction units.	General Electric Co. Bethlehem Shipbuilding Co. Terry Steam Turbine Co. Elliott Co. Moore Steam Turbine Corp.
		Series of alternate rows of converging fixed and moving orifices. Both diameters and blade lengths steadily increase. Steam flows axially. Fine clearances at ends of some types of blading. End-tightening used with certain blading. Usually many rows.	Small heat drop per row permits use of ratios of wheel speed to steam speed conducive to high efficiency. Usually throttle governing. Require extended warming-up period before starting.	Usually require large floor space. High efficiency possible in all sizes. Blade limitations in exhaust ends of large units. Easy to bleed for feedwater heating.	Turbo-generators. Marine propulsion. Geared drives. Extraction units. Mill drives.	Allis-Chalmers Mfg. Co. Westinghouse Elec. & Mfg. Co. American Brown Boveri Co. Bath Iron Works.
	Radial-flow Ljungstrom	Alternate series of radial rings carrying converging reaction blading arranged for radial steam flow. Low pressure stages are axial flow on large units. Alternate rings revolve in opposite directions. Hence two generators tied together electrically. Small clearances over blades.	High relative velocities lead to high efficiency with few blade rows. Radial rings require nearly constant blade length. Small leakage losses. Can be started quickly. Can use high temperatures. No critical speed through which to pass.	No foundation for unit which is supported by condenser. Quickly erected. High efficiency, particularly in high-pressure and in small size condensing units. No outer heat insulation needed.	Turbo-generators. Extraction units.	S.T.A.L. (Finspong, Sweden) Brush Engineering Co. (Loughborough, England) A.E.G. (Berlin, Germany) M.A.N. (Nürnberg, Germany) S.S.W. (Berlin, Germany) Brown Boveri & Co. (Baden, Switzerland)
	Velocity Multi-stage Impulse or Curtis-Rateau	1 Curtis stage followed by series of simple impulse stages with intermediate diaphragms carrying nozzles. Shorter than straight multi-stage impulse unit.	Only moderate pressures and temperatures in casing, due to large pressure drop in first nozzle. Favorable ratios of wheel speed to steam speed possible in both Curtis and other stages.	Compact and short. Efficiency practically the same as multi-stage impulse up to 5000 kw. and fair in larger sizes. Rugged construction in moderate sizes. Lower pressure on high-pressure gland packing box due to Curtis stage.	Turbo-generators. Auxiliary drives for larger units. Centrifugal pump drives. Turbo-blowers and compressors. Mill drive through gearing. Bleeder and extraction units.	Ingersoll-Rand Co. General Electric Co. DeLaval Steam Turbine Co. Moore Steam Turbine Corp. Terry Steam Turbine Co. Elliott Co. Murray Iron Works Co. B. F. Sturtevant Co.
	Velocity-reaction or Curtis-Parsons	1 Curtis stage followed by series of Parsons stages. Frequently of disc-and-drum construction. Larger clearances than in Parsons.	Pressure and temperatures in casing reduced by expansion in Curtis nozzle.	Shorter and more rugged than standard Parsons. Efficiency good in small sizes, and fair in large units.	Turbo-generators. Marine propulsion. High-efficiency marine geared auxiliary generator sets. Bleeder and extraction units. Mill drives.	Westinghouse Elec. & Mfg. Co. American Brown Boveri Co.

NOTE—Some manufacturers build several types of turbines and their names may have been unintentionally omitted from the above list for certain types.

The following firms are among the builders of turbines in Europe. Great Britain: *Impulse*, British Thomson-Houston Co.; Metropolitan-Vickers Co.; Frazer and Chalmers; Bellis and Morcom; John Brown and Co.; W. H. Allen and Son; *Parsons*, C. A. Parsons and Co.; Richardson, Westgarth and Co.; English Electric Co. Switzerland: *Impulse*, Escher, Wyss and Co.; Oerlikon Co.; *Parsons*, Brown Boveri and Co. Germany: *Impulse*, A.E.G.; M.A.N.; W.U.M.A.G.; Bergman E.W.; *Parsons*, S.S.W.; Guttehofnungshutte; Maffai-Schwartzkoff. Holland: Geb. Stork and Co. Czecho-Slovakia: Erste Brunner M.F.G. Italy: Franco Tosi. France: A.E. Sautter-Harlé; C.E.M.; S. G. Als-Thom.

charge period. A low-pressure turbine usually is employed with low-pressure accumulators, being fitted with a throttling governor. High-pressure accumulator turbines generally have a series of valves to admit additional steam to lower stages as the pressure falls in the accumulator. Turbines have been built to operate both as high-pressure and as accumulator turbines by means of suitable valve gear and nozzles to admit accumulator steam at the desired points.

Accumulator turbines have been used to supply steam for carrying peak or other sudden demands for power. The load curves for such peaks are generally triangular in form. This steam requirement can be found by means of a series of Willans lines (see p. 8–69) showing turbine performance at a series of pressures from full to discharge pressure. The pressure-discharge curve of the accumulator is plotted. The total steam demand over the peak is found from these two curves and from the load curve, by taking small increments of load and determining the total steam and the pressure drop for the duration of each. These steam quantities can be plotted and integrated.

When the total steam required from the accumulator is known, the amount of water in the accumulator to yield this steam is found approximately as follows: Let w' = total steam to turbine, lb.; h'_{fg} = latent heat at average pressure in accumulator, B.t.u. per lb.; w'' = storage water required in accumulator, lb.; h_{fc} = heat content of saturated water at full accumulator pressure, B.t.u. per lb.; h_{fd} = heat content of saturated water at completely discharged pressure of accumulator, B.t.u. per lb.; w_m = weight of 1 cu. ft. of saturated water at average accumulator pressure, lb.; $w' \times h'_{fg} = w'' (h_{fc} - h_{fd})$. C = storage capacity needed in accumulator, cu. ft. Hence

$$''_{fc} - h_{fd}) \times \cdot$$

Usually 3% additional water storage is provided to compensate for radiation. The shell volume is so chosen that the storage water occupies about 90% of the total volume when fully charged.

For data and illustrations of the largest installation at present (1932) of Ruths accumulators and turbines, at Charlottenburg, Germany, see *Engg.*, June 13, 1930; also see Christie, The Peak Load Problem in Steam Power Stations, *Mech. Engg.*, Dec., 1928; Taylor and Wettstein, The Ruths Steam Accumulator, *Mech. Engg.*, Aug., 1925.

TURBINE CHARACTERISTICS.—Table 1 classifies certain information regarding types, characteristics, advantages and disadvantages, services and manufacturers of the various forms of steam turbines.

TURBINE SPEEDS.—For 25 cycles, 1500 r.p.m.; for 60 cycles, 3600, 1800 and 1200 r.p.m. European turbines for 50 cycles, 3000 and 1500 r.p.m. There is a tendency to extend the capacity of the higher speed units by the use of multi-cylinders. Turbines direct-connected to pumps and blowers usually operate at the speed of the driven unit. Turbines for geared sets may run at any desired speed, and have been built for speeds of 6000 to 7200 r.p.m. in the smaller sizes. This leads to low first cost and increased efficiency.

2. STEAM TURBINE CYCLES

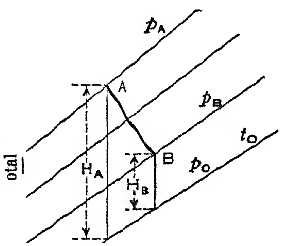
If steam could be expanded in a turbine with no friction or other losses, expansion would be isentropic. Theoretically, steam turbines operate on the Rankine Cycle, or its modifications such as the Regenerative, Reheating or Regenerative-reheating Cycles. Turbine problems involving energy transformations in the steam are based on isentropic adiabatic expansion with the necessary modifying factors. Since such adiabatic expansion occurs at constant entropy, these problems readily can be solved on a Mollier diagram. (See p. 5–18.)

In the **Rankine Cycle** (see p. 5–14), the steam is expanded isentropically from the initial steam condition before the throttle valve to the exhaust pressure. The heat drop measures the heat thus theoretically available for work (see p. 5–18). The **engine efficiency** expresses the ratio by which the actual turbine approaches the Rankine cycle in converting into work the energy available from adiabatic expansion. The numerator may be the heat equivalent of either internal, coupling or generator-output kilowatts. The denominator is the product of heat drop and total pounds of steam. It is necessary therefore to state whether the calculated ratio is "Engine Efficiency based on internal kilowatts," "Engine Efficiency based on coupling kilowatts," or "Engine Efficiency based

on generator output." 1 kw.-hr. = 3412 B.t.u. per hr. Let I kw. = internal kilowatts; h_1 = total heat per lb. of steam at initial conditions before the throttle; h_2 = total heat per lb. of steam after isentropic expansion to exhaust pressure; w = total lb. of steam per hr. Engine efficiency, based on internal kilowatts, = $(3412 \times I \text{ kw.})/w\,(h_1 - h_2)$. Similar expressions can be found for coupling kilowatts or generator output. Where turbines are sold to drive pumps, fans, or other equipment, the rating frequently is expressed as Brake Horsepower at the coupling. The numerator in the above equation then becomes $2543 \times$ B. Hp. Straight condensing, high-pressure and low-pressure turbines with no extraction or reheating, follow the Rankine cycle.

The Regenerative Cycle, the Reheating Cycle, and the combination of these in the **Regenerative-reheating Cycle** (see p. 5–15, Steam) are used extensively for steam turbines.

There are difficulties in applying the engine efficiency to turbines operating on certain of these cycles. G. Darrieus (The Rational Definition of Steam Turbine Efficiency, *Engg.*, Sept. 5, 1930) proposed a new standard which will apply to all turbines. The *Darrieus efficiency* is the ratio of the heat to work to the difference of isentropic drops of the steam entering and leaving the turbine, the isentropic heat drops being measured with relation to an arbitrary condenser pressure p_0, or to the temperature t_0 of the surroundings. Thus in Fig. 17 let A and B represent initial and final conditions, respectively, in a turbine, and let H_A and H_B be the isentropic heat drops from A and B respectively to the condenser pressure p_0. w = lb. of steam per hr.; kw. = output.



Entropy
FIG. 17

$$\text{Darrieus efficiency} = \frac{3412 \times \text{kw.}}{w\,(H_A - H_B)}.$$

This gives somewhat different values from the engine efficiency as calculated above.

The general expression of Darrieus efficiency is as follows: For any turbine to which w_A, w_B, w_C . . . lb. of steam per hour are supplied at the states A, B, C . . . , and from which w_x, w_y, w_z are extracted or discharged at the states x, y, z (where the total steam supplied = total steam withdrawn), and the heat equivalent of work done = $3412 \times$ kw., where kw. = kilowatts, internal, coupling, or generator-output, the Darrieus efficiency, internal, coupling, or generator-output respectively,

$$= \frac{3412 \times \text{kw.}}{w_A H_A + w_B H_B + w_C H_C + \ldots - w_x H_x - w_y H_y - w_z H_z}$$

where H_A, H_B, . . . H_z are the available isentropic heat drops at states A, B, . . . z referred to the surrounding temperature t_0, or to the corresponding saturation pressure p_0 of the condenser. See also A Steam Chart for Second Law Analysis, J. H. Keenan, *Mech. Engg.*, Mar., 1932, and Turbine Plant Efficiency Calculations, J. C. Smallwood, *Combustion*, July, 1934.

3. NOZZLES

CRITICAL PRESSURE.—The simplest form of nozzle consists of a circular hole with a well-rounded mouth on the upstream entrance in the side of a chamber containing high-pressure steam at absolute pressure p_1. As the pressure p_2 on the discharge side of the nozzle becomes less than p_1 the flow of steam increases until the so-called critical pressure is reached. The flow will not increase with any further decrease in the pressure on the discharge side of the nozzle. The critical pressure is $0.5457\ p_1$ for superheated steam, and $0.5774\ p_1$ for saturated steam. Its value under any conditions can be found from the equation,

$$\text{Absolute Critical Pressure, } p_c = p_1 \left(\quad \right)^{n/(n-1)}$$

where p_1 = initial pressure, lb. per sq. in., abs., and n = exponent for adiabatic expansion at constant entropy ($p\,v^n$ = constant) of the steam at the stated conditions. The value of n varies for differing steam conditions; for superheated steam, n varies with different steam tables. An average value is $n = 1.3$; for wet steam, $n = 1.035 + 0.1\,q$, where q = quality expressed as a decimal. The velocity at the critical pressure is substantially

that of sound in the gas or vapor at the pressure and density existing at the throat. See p. 3–78.

The steam jet leaves the nozzle in practically straight lines as long as p_2 equals or is greater than the critical pressure. Hence for these expansions, nothing more is needed in a turbine than a convergent passageway, with the discharge directed at the desired angle toward the revolving blades. The orifice must be convergent, for the rate of increase of velocity exceeds the rate of increase of volume, until the critical pressure is reached. If p_2 is less than the critical pressure, the pressure in the throat, or narrowest part of the orifice, remains at the critical pressure, and further expansion of the steam occurs after leaving the orifice. This causes the jet to expand in all directions. Walls are necessary to confine this further expansion with its accompanying increase in velocity and volume. To do useful work, the jet must be projected toward the blades in a fixed direction, usually at an angle of 12° to 30° to the axis of the blade row. The smaller angles should be used when high efficiency is desired. Below the critical pressure the volume increases at a greater rate than the velocity. A diverging section, therefore, is added to the throat, forming a convergent-divergent nozzle, with mouth dimensions suitable for the range of expansion. The total angle of the divergent walls varies from 6° to 12°. The smaller angles appear to give better expansion. If the nozzles are of rectangular cross-section, the sides continue to diverge for their full length. This also is done in some circular nozzles.

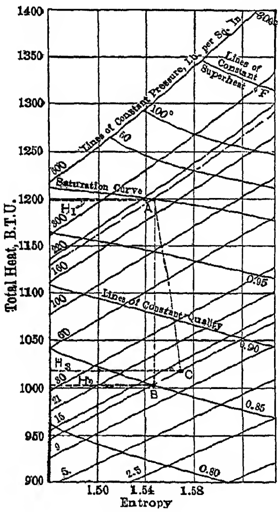


FIG. 18

THEORETICAL NOZZLE VELOCITY.—The heat drop $(h_1 - h_2)$ from any initial conditions to a final pressure can be found by the method shown on page 5–18.

As the steam expands in a nozzle, a portion of this available heat is transformed into kinetic energy and increases the velocity of the jet. The theoretical velocity V_0, ft. per sec., resulting from complete transformation of this heat drop is $V_0 = 223.8\sqrt{(h_1 - h_2)}$. This velocity is not obtained in an actual nozzle due to friction, eddy and other losses. Let η_n = efficiency of conversion into kinetic energy in a nozzle expressed as a decimal, or *nozzle efficiency*. The actual velocity V_1 of the steam jet leaving the nozzle is

$$V_1 = 223.8\sqrt{\eta_n\,(h_1 - h_2)}.$$

Another term used in considering nozzles is the velocity coefficient, k_n, which is the ratio of the actual velocity V_1 to the theoretical velocity V_0.

Nozzle losses appear as reheat. The total heat at the nozzle mouth,

$$h_3 = h_2 + (1 - \eta_n)\,(h_1$$

This is represented by point C on Fig. 18 on the pressure line p_2 of the mouth. The specific volume of the steam can be determined from conditions at point C.

In multi-stage turbines the steam may enter the nozzles of all but the first stage, with an appreciable velocity V_c, the carry-over from the preceding blade row, and with a kinetic energy $(V_c/223.8)^2$. The velocity of the steam leaving the nozzle becomes

$$V'_0 =$$

The total heat at the nozzle mouth is $h'_3 = h_2 + (1 - \eta_n)\{(\quad \cdot h_2) + (V_c/223.8)^2\}$. This exceeds h_3 above, by the amount $(1 - \eta_n) \times (V_c/223.8)^2$.

Carry Over is the portion of the absolute velocity leaving the preceding blade row which actually enters the nozzle in the proper direction. Formerly it was assumed that carry over only amounted to 50 to 75% of the absolute leaving velocity. Improvements in design of nozzle entrances, have increased this recovery to about 85 to 90% at most efficient load. Even higher recoveries are possible with further improvements in design.

Many data have been published on nozzle experiments. See Stodola, Steam Turbines; Goudie, Steam Turbines. Nozzle efficiency is greatly influenced by: *a*, form of approach to nozzle; *b*, con-

dition of steam entering nozzle; *c*, degree of roughness of nozzle; *d*, length of nozzle; *e*, thickness of partitions in groups of nozzles; *f*, conditions of outlet edges of partitions.

The Steam Nozzles Research Committee of the Inst. of Mech. Engrs. has published many valuable data on nozzle efficiency. (*Proc.*, Inst. Mech. Engrs., 1923, 1924, 1925, 1928, 1930). The following data show the performance of a set of converging impulse nozzles with partitions 0.04 in. thick, with 20° angle, and with a parallel portion beyond the throat equal in length to twice the width at the throat.

Theoretical velocity V_0 ft. per sec.	400	600	800	1000	1200	1400	1600	1800
Approximate corresponding heat drop, B.t.u. per lb.	3.2	7.2	12.8	20.1	28.8	39.2	51.2	64.8
Velocity coefficient, k_n	0.976	.955	.947	.946	.944	.944	.945	.946
Nozzle efficiency η_n	0.953	.915	.897	.895	.892	.892	.894	.895

The Nozzle Research Committee also noted that better coefficients were obtained when the nozzle partitions were well chamfered at the outlet edge. The angle of discharge is decreased by this chamfer and may be less than the nominal angle. These coefficients include any influences of supersaturation.

Kearton states that for curved nozzles used in multi-stage impulse turbines, velocity coefficients of 0.96 for velocities of 1000 ft. per sec., and of about 0.94 for velocities of 3000 ft. per sec. may be assumed. These correspond to nozzle efficiencies of 0.922 and 0.885. Superheating will increase and wetness will reduce these figures, the maximum variation being 2%.

Wirt (An Experimental Investigation of Nozzle Efficiency, *Trans.* A.S.M.E., xlvi, 1924) gives the results of tests of nozzles by means of impact tubes using air. Warren and Keenan (A Machine for Testing Steam Nozzles by the Reaction Method, *Trans.* A.S.M.E., xlviii, 1926) give results using steam, which indicate velocity coefficients of 98.3% with standard General Electric turbine nozzles, and with outlet velocities of 400 to 2000 ft. per sec., equivalent to a nozzle efficiency of 96.5% which is quite high. The results of these two papers are as follows:

Ratio of theoretical velocity to sound velocity	η_n, Reaction Test Steam (Warren and Keenan)	η_n, Impact Test, Air (Wirt)
0.86	96.8%	97.1%
0.67	96.8%	96.3%
0.50	97.4%	97.5%

Steam Shock leads to nozzle losses when the pressure at the mouth is greater than that for which the nozzle is designed. The steam appears to expand as though conditions at the mouth were those for which the nozzle was designed, until at a certain point in the nozzle the pressure becomes less than that at the mouth. Recompression then begins to the pressure at the mouth, and the volume decreases. This causes the jet to detach itself from the wall, and often results in setting up pressure pulsations. The detached jet may no longer be in the direction of the nozzle axis, and so will have an unfavorable angle of discharge. (See Stodola, Steam Turbines, p. 88 on Steam Shock.)

When steam expands rapidly from a slightly superheated condition, it does not begin to condense when the saturated condition is reached, but continues to expand as in the superheated region, thus becoming supersaturated. *Supersaturation* has been shown to exist in simple nozzles (Yellott, Supersaturated Steam, *Trans.* A.S.M.E., FSP-56-7, June, 1934). Supersaturation, with its lesser steam volumes, causes greater nozzle discharges than saturated steam. Supersaturation also tends to lessen the efficiency of nozzles due to energy loss when drops start to form and the steam mass seeks equilibrium. Supersaturation effects should be considered in nozzle design.

Commercial nozzles may not have as high efficiencies as given above, due to poor entry conditions, too wide flare beyond the throat, or too wide and unchamfered partitions at the mouth. Gains of 5% in nozzle efficiency have resulted from the redesign of nozzles in certain cases.

FLOW OF STEAM IN NOZZLES.—The design of all nozzles is based on the following formula of continuity:

$$W = A_t V_t \div 144 v_t = A_m V_m \div 144 v_m,$$

where W = weight of steam flowing, lb. per sec.; A_t = area at throat, sq. in.; V_t = velocity at throat, ft. per sec.; v_t = specific volume of the steam at throat conditions, cu. ft. per lb.; A_m, V_m and v_m are similar conditions at the mouth.

The pressure at the throat of a convergent-divergent nozzle is always the critical pressure. Hence, the flow of steam through this nozzle is constant, regardless of the value of p_2, the discharge pressure. The converging portion of such a nozzle is short, usually about 1/4 in., and the expansion here is assumed to take place with no losses if the approach is straight and has no obstructions. For this condition of flow, Goudie, in Steam Turbines, gives the following equations for the flow of steam through a convergent-divergent nozzle:

(*a*) Steam initially superheated: $W_s = 0.3155 A_t \sqrt{p_1 \div v_1}$, where W_s = lb. of superheated steam flowing per sec.; A_t = area of throat, sq. in.; p_1 = initial absolute steam

pressure, lb. per sq. in., v_1 = specific volume of steam at pressure p_1 and the stated superheat. Velocity in the throat, ft. per sec. = $V_t = 72.24\sqrt{p_1 v_1}$.

(*b*) With steam initially dry, Goudie holds that the steam remains supersaturated during its expansion from initial to throat conditions, *i.e.*, no condensation from adiabatic expansion takes place and hence the specific volume is less than if condensation does take place. The discharge W of a nozzle, lb. of steam per sec., is given by the same equation as for superheat. Some designers do not allow for supersaturation, and use a coefficient of 0.3004 instead of 0.3155 in the preceding equation.

Converging-diverging nozzles of curved form, as used in the low-pressure stages of impulse turbines, are designed with the nozzle efficiency applied to expansion up to the throat as well as to the complete expansion, owing to the loss from recompression in turning the steam around the curve. The throat velocity V'_t is then calculated from the formula $V'_t = 223.8\sqrt{\eta_n(h_1 - h_t)}$ where η_n = nozzle efficiency and h_t = total heat per lb. after isentropic expansion from initial conditions to throat pressure p_t. h_t is readily found on the Mollier diagram. If p_2 is greater than critical pressure, converging nozzles are used, usually of rectangular cross-section, and of the form shown in Fig. 5.

The area of the mouth of either convergent or convergent-divergent nozzles is given by the equation $A_m = 144\,Wv_m \div V_m$, where W = weight of steam flowing per sec., lb., v_m = specific volume at mouth conditions, found from the heat-entropy diagram; after allowing for reheat from nozzle losses, $V_m = 223.8\sqrt{\eta_n\{(h_1 - h_2) + (V_c/223.8)^2\}}$ = velocity at mouth, ft. per sec.; h_2 = total heat after isentropic expansion from initial conditions to mouth pressure p_m; V_c = carry-over velocity, ft. per sec.

When the mouth of the nozzle is too large the pressure in the nozzle falls below p_2 and the nozzle has "over-expansion." Serious eddies, shocks and other losses resulting from recompression are set up, and the loss from this cause may be serious. If the mouth is too small, the steam will not be fully expanded until after leaving the mouth of the nozzle, and is thus *under-expanded.* The loss from under-expansion is only about 30% of the loss from a similar amount of over-expansion, and hence is not of a large order of magnitude. Some designers provide for a small amount of under-expansion in all stages. This gives the best economy of the turbine at a load below maximum, and also a better range of high economy, and provides for low boiler pressure. The nozzle calculations given above may be used to design steam nozzles for uses other than in turbines.

ARC OF WHEEL COVERED BY NOZZLES.—To give the jet proper direction, a series of moderately small nozzles, spaced around the arc of the wheel, is used instead of one large nozzle. The partitions at the mouths of this group of nozzles are quite small, so that the clear opening is usually 0.88 to 0.94 of the pitch of nozzles. The length of circular arc occupied by nozzles of rectangular cross-section is given by the equation

$$ncP = A_m \div l_m \sin(\alpha - \phi/2),$$

where n = number of nozzles; P = pitch of nozzles, in.; c = thickness coefficient (0.88 to 0.94); A_m = calculated mouth area at right angles to axis of nozzle, sq. in.; l_m = radial depth of mouth, in., the minimum depth being 1.5 to 2% of mean blade ring diameter; α = angle of axis of nozzle to axis of blade row (12° to 30°); ϕ = angle of divergence of side walls of nozzle if any ($\phi = 0$ for convergent orifices). For complete admission

$$l_m = A_m / [\pi Dc \sin(\alpha - \phi/2)]$$

where D = mean diameter of the ring of nozzles, in. c is made as large as possible by chamfering off the outlet edge of the partition to a thin section.

If the nozzles of several succeeding stages do not cover the whole circumference of the wheel, the nozzle lead of succeeding sets must be calculated, as the steam is carried round the wheel a short distance in crossing the blades. The amount of lead is largely affected by the clearances between nozzles and blades on both inlet and outlet sides of the blade. The larger the clearance on the outlet side, the less important is the matter of lead. Another factor is the effect of increasing the nozzle arc of the succeeding stage to provide for the increased steam volume. This usually automatically takes care of the lead. Nozzle lead, as a rule, is a minor consideration in turbine design. See Goudie's Steam Turbines, pp. 273–275.

CROSS-SECTION OF NOZZLES.—Converging-diverging nozzles may be of circular cross-section at throat and mouth, or they may have rectangular cross-section throughout. The former are used on single-stage turbines, and sometimes in the first stage of multi-stage turbines of moderate efficiency. Frequently, these are pitched slightly towards the center of the shaft to cause the jet to enter the blades with less spilling at the shroud.

Simple converging orifices in diaphragms are of rectangular section throughout. These

orifices may be cast in place in the diaphragms or may be of built-up construction with machined surfaces and may be bolted or welded to the diaphragms in sections.

The discharge angle α of convergent orifices must frequently be increased in the last stage of impulse turbines to provide passageway for the large volume of steam present, without too high leaving loss. The pitch of nozzles is fixed arbitrarily. Usually it equals 3 to 4 times the blade pitch. Another rule is not to exceed 1 in. in width at the throat when measured at right angles to the axis of the jet.

Murray Iron Works Co. uses for converging-diverging nozzles a straight tube with rounded entrance, down the center of which projects a tapered needle valve controllable from outside the casing. This forms an annular throat between needle valve and pipe which expands at the angle of taper of the needle to a full clear mouth opening. The expansion ratio can be adjusted to suit the steam conditions by using the proper needle valve. Also the capacity of the unit can be adjusted by changing the throat area through shifting the position of the needle valve to give the necessary steam flow.

NOZZLE MATERIALS.—Most converging-diverging nozzles are made of cast brass or bronze, though some have been made of steel alloys and even of cast iron. One manufacturer uses for moderate superheat a bronze of about the following composition: Copper, 79; lead, 10; tin, 10; phosphorus, 1. Other metals, as Monel metal, Everbrite and stainless steel, have been used.

Convergent orifices usually are formed by casting sheet-metal partitions of nickel steel in the diaphragms. These contain from 3.5 to 5% nickel to increase their strength and resistance to corrosion. Chrome-vanadium steel also has been used for this purpose. When the nozzle block is of gun metal, phosphor-bronze or copper partitions are used. With built-up nozzles, Monel metal, stainless steel, and other alloys may be used.

In diaphragms with small nozzle height, the nozzle plates have been brazed or welded into diaphragms of cast or forged steel in hydrogen furnaces. They are easy to manufacture, cheaper than built-up nozzle plates, and give accurate nozzle forms.

4. BLADING

The steam leaving the nozzle in impulse turbines is directed against the revolving blades at an angle of from 12° to 16° with the plane of the blade for the early stages, increasing to 30° in some low-pressure stages. The relative entrance velocity of the jet and its entering angle can be found from the velocity diagram, as in Fig. 19. The exit angle of the blades is made the same as the entering angle in some small units, particularly those of the re-entry type, and also in the early stages of turbines.

LOSSES IN BLADING.—A portion of the energy of the steam jet is lost while crossing a blade passage, due to friction, compression and re-expansion and resultant eddies. Hence the relative velocity V_{ro} leaving the blade is less than the entering relative velocity V_{ri}. The ratio of leaving to entering relative blade velocities is called the **blade coefficient,** *i.e.*, $V_{ro}/V_{ri} = k_b$. This coefficient is influenced by the width of blade, the form of its rear flank, the total angle through which the steam is turned, the relative velocity of the steam and the smoothness of the blades. Kraft (The Modern Steam Turbine, pp. 25–26) states that higher efficiencies are obtained with increased radial length of nozzles and blades. He attributes this to the influence of losses due to disturbed flow in the region of the walls, and these losses increase with shorter blades. He shows that the stage efficiency η_s in an impulse turbine with $^3/_{16}$-in. blades is only about 42%, while with 4-in. blades $\eta_s = 85.2\%$, other characteristics of nozzles and blades remaining the same. Hence the importance of long blades.

RECOMPRESSION IN BLADE PASSAGES.—Stodola remarks that the greater part of blade losses result from compression and re-expansion in the curved passages. Belluzo (Steam Turbines, pp. 74–88) develops the principles of shock and recompression in curved passages. In the case of a curved passage, recompression results from the effect of centrifugal force upon the steam molecules which are traveling at a high relative velocity through the curved passage of the blades. If the curve of the blade passage is assumed to be an arc of a circle of radius R, ft., and of width, h, ft. (h is usually small), and if d_1 = density of steam, lb. per cu. ft., $g = 32.2$, V_r = mean velocity of steam crossing blade, ft. per sec., p_a = absolute pressure of steam at inlet and outlet of blade (stage pressure), lb. per sq. in., p_x = absolute pressure that the steam assumes as a result of recompression due to centrifugal force, lb. per sq. in., then

$$144\,(p_x - p_a) = \frac{d_1}{g} \times \frac{h}{R} \times V_r^2, \quad \text{or,} \quad p_x = p_a + \frac{d_1}{g} \times \frac{h}{R} \times \frac{V_r^2}{144}.$$

If recompression were the only consideration, wide passages with wide blades having curves of large radius would be best. Such blades have increased friction losses. Hence

blade widths have been a compromise, determined largely as a result of test and experience.

Zietemann (Dampfturbinen, p. 72) concludes that the blade coefficients are principally influenced by the total angle through which the steam is turned. He presents the following values for commercial blading where β_1 and β_2 are the entering and leaving angles of the blade.

$(\beta_1 + \beta_2)/2$.......	10	20	30	40	50	60	70	80
k_b...............	0.77	0.85	0.89	0.918	0.938	0.953	0.962	0.966

Zietemann's values depend only on the total angle of deflection. Stodola (Steam Turbines, p. 179) presents data from some Brown Boveri experiments which indicate that the maximum blade coefficient k_b, with blades having 30° inlet and outlet angles, varies with relative entering steam velocity V_{ri}. For a given blade, the coefficient curve is hyperbolic and the coefficient drops off with relative steam velocities above or below that at maximum value. The mean of the maximum values of k_b is as follows:

V_{ri}...............	400	800	1200	1600	2000	2400	2800	3200
k_b...............	91.2	91.	90.8	90.4	89.9	89.	87.8	86.2

In blading design, consideration should be given to these values as well as to Zietemann's in choosing k_b.

Aerodynamic methods are being applied to blade studies. As a result of these studies, the inconsistencies of earlier test data will be removed and correct expressions found for velocity coefficients, probably in terms of the Reynolds number and the Mach ratio.

For first approximations before the velocity diagram is completed, k_b may be assumed to vary proportionately from 0.94 at 100 ft. per sec. steam velocity to 0.885 at 1200 ft. per sec. velocity.

Fig. 19*A* is the velocity diagram for one stage of a multi-stage turbine, which stage

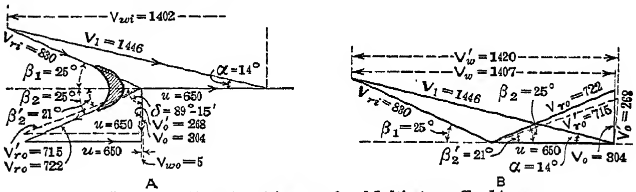


FIG. 19. Velocity Diagram for Multi-stage Turbines

operates under the following conditions: Pressure and temperature before nozzle (no carry over), 65 lb. per sq. in., abs.; 380° F.; expands to 38 lb. per sq. in., abs.; nozzle angle, $\alpha = 14°$; nozzle loss, 8%; wheel speed, $u = 650$ ft. per sec.; heat drop, $h_1 - h_2 = 45.4$ B.t.u. Velocity leaving nozzle $V_1 = 223.8\sqrt{0.92 \times 45.4} = 1446$ ft. per sec.

From the diagram, the relative entering velocity $V_{ri} = 830$ ft. per sec., and blade entrance angle $\beta_1 = 25°$. Assume $\beta_2 = 25°$ and $k_b = 0.87$. Relative leaving velocity $V_{ro} = 0.87 \times 830 = 722$ ft. per sec. Absolute leaving velocity $V_o = 304$ ft. per sec. The blade form can be drawn from this diagram.

The work done, in ft.-lb., is the product of the wheel speed u and the velocity of whirl V_w, divided by gravity g. The velocity of whirl is the sum of the co-ordinates of the absolute entering and leaving velocities in the direction of motion. This can be readily determined from Fig. 19*B*, which is the usual form of combined velocity diagram. The lower triangle of Fig. 19*A* is placed on the same base u as the upper triangle, and $V_w = (V_{wi} + V_{wo})$ can be determined graphically, as shown, and equals 1407.

Then work done in B.t.u. : g 36.5 Combined nozzle and blade efficiency = 36.5 / 45.4 = 80.4%.

ENTRANCE AND EXIT ANGLES OF BLADES.—The exit angle often is made 3° to 10° less than the entrance angle, as shown in the dotted lines, Fig. 19*A* and *B*, where $\beta'_2 = 21°$, a decrease of 4°. k_b becomes 0.862 and $V'_w = 1420$. The work done per stage now is 36.85 B.t.u., and the combined nozzle and blade efficiency becomes 36.85 / 45.4 = 81.2%.

The volume of steam can be assumed to remain constant during its passage across the blades. The length of blade on its inlet side is usually 1/16 in. longer than the width of the nozzle mouth. The outlet length of the blade should be increased over the radial length of nozzle mouth due to the decreased velocity V_{ro} at the blade outlet. If l_o =

outlet length of blade, and l_n = radial height of the nozzle, $l_o/l_n = V_1 \sin \alpha / V_{ro} \sin \beta_2$. When the angle has been closed a number of degrees, as shown above, this ratio may call for an excessive length on the outlet side of the blade. Some turbine builders vary the length of the shorter blades to meet these requirements. Others use only flat concentric shrouds for mechanical reasons, and make the inlet side somewhat longer than necessary. Frequently, some throttling exists on the outlet side and the blade functions partly as a reaction blade.

The dynamical thrust on the blades per lb. of steam $F = (V_{ri} \sin \beta_1 - V_{ro} \sin \beta_2)/g$. This is small in impulse turbines.

The theoretical required outlet length of blade is reduced approximately 1% for each 1% of reaction given to the blade. The usual limit of reaction in impulse blading appears to be about 10 to 15%. Most impulse blades on large units now are designed with a degree of reaction on the outlet side of the last rows of blading. Some impulse turbines built in Europe have end-tightened blades when a degree of reaction is used.

Some designers use a ratio t/r in the design of impulse blades, where t = width of blade passage, in., and r = radius of curvature of face of blade, in. Hodgkinson (Steam Turbines and Condensing Equipment, *Elec. Jour.*, Nov., 1924) says t/r should have decreasing values with increasing velocities. For a Curtis wheel, he suggests with 2000 ft. per sec. nozzle velocity, t/r for first row = 0.3, and for second row = 0.6.

The inlet angles of commercial blades usually are increased several degrees over that found on the velocity diagram. This allows the steam to enter without striking the rear of the blade if the initial pressure falls slightly and the nozzle velocity V_1, decreases. Less loss from shock occurs if steam strikes the face of the blade at a slight angle than if it

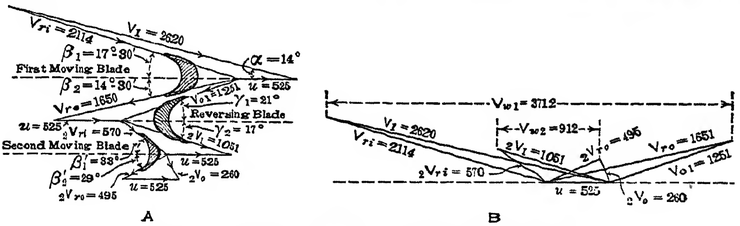


FIG. 20. Velocity Diagram for Velocity Type Turbine

strikes the rear of the blade. The former condition is *over-speeding*. The latter is *under-speeding*, and is undesirable.

BLADE WIDTH.—Several considerations influence blade width. Friction losses increase as the radius of curvature of the blade channels decreases. Hence, narrow impulse blades are undesirable. On the other hand, the surface friction of the sides of the channel may become prohibitive in very wide blades. Intermediate blade widths are therefore selected. Blade widths vary from 1/2 in. to 3/4 in. in small, single-stage impulse turbines, and from 3/4 in. to 2 in., or even larger, in multi-stage units. The wider blades are used in the larger machines. The maximum blade length should not exceed 8 to 12 times the width.

BLADE DESIGN.—When inlet and outlet angles and blade width have been chosen, the blades can be designed. The curve forming the face of the blade is drawn tangent to the lines forming inlet and exit angles. The pitch of the blades usually is chosen as 0.5 to 0.6 of the blade width, but never should exceed 1 in. Kearton gives the pitch $P = b/2 \sin 2\beta_1$, where b = blade width, in., and β_1 = entrance angle of blade. The rear flanks of the blade are made parallel to the inlet and exit angles. The curve of the back of the blade is fixed by experience and depends upon the degree of recompression to which the steam jet may be subjected. The outlet edges of the blades are made as thin as manufacturing considerations will permit. The inlet edges of high-pressure blading are made thin. On account of erosion in low-pressure blading, inlet edges are now thick and rounded.

Blading for velocity-type turbines can be computed from the velocity diagram. Figs. 20, *A* and *B* are for a Curtis turbine based on the following assumptions: Initial conditions, 140 lb. per sq. in., abs.; 450° F.; back-pressure, 20 lb. per sq. in., abs.; nozzle efficiency, 88%; blade angle, $\alpha = 14°$; wheel speed, 525 ft. per sec.; heat drop, $(h_1 - h_2) = 155.7$ B.t.u. per lb.; k_{b1}, k_{br}, and k_{b2} for first moving, reversing, and second moving rows respectively are 0.83, 0.84 and 0.87. Usually only the diagram Fig. 19*B* is drawn. The veloci-

ties of whirl from the diagram are 3712 ft. per sec. for the first row, and 912 ft. per sec. for the second row, a total of 4624 ft. per sec.

$$\text{Work done} = \frac{uV_w}{778.6\,g} = \frac{525 \times 4624}{778.6 \times 32.16} = 96.9 \text{ B.t.u.}$$

The combined nozzle and blade efficiency = 62.2%. Three rows of moving blades are used on some Curtis wheels, which will operate at low wheel speeds.

All exit angles are shown slightly less than the inlet angles, necessitating considerably increased outlet lengths for the blades.

The closure of exit blade angles in a Curtis stage, while increasing efficiency, may lead to too great a ratio, at the last row, of outlet length of blade to nozzle height, especially on long blades, since $l_3/l_n = V_1 \sin \alpha/2V_{ro} \sin \beta_2'$. In some designs, a ratio of l_3/l_n is assumed of from 2.5 to 3.5 on a two-row Curtis stage. Goudie (Steam Turbines, p. 305) shows a method of laying out the diagram and finding the blade angles and the work done to give the desired blade length ratio.

Velocity diagrams for re-entry turbines can be laid out in the same manner as for the Curtis stage. The blades often have equal inlet and outlet angles. k_b in the reversing chamber is about 3 to 5% greater than given by Zietemann. The discharge angle of the reversing chamber can be found graphically. It must be such that the relative velocity of steam to blades is at the angle fixed by the first part of the diagram for the blade outlet. See Goudie (Steam Turbines, pp. 288–9) for method of calculating helical flow blading.

SUCTION EFFECT.—When the steam jets pass through the clearance between nozzle mouth and blade entrance, the space corresponding to the thickness of the nozzle partition and in the direction of the nozzle angle, is not filled by the jets. The jets on either side of this space aspirate steam from clearance, and this suction effect is further increased by the pumping effect at blade entrance. The resulting losses may be reduced by chamfering the outlet edges of these partitions to a fine line to produce a continuous band of steam from the nozzles. This suction of steam from the inlet side of the disc may cause a pressure difference on the two sides of the disc unless enough holes are provided in the disc to balance pressures. Minimum clearance between disc and nozzle will reduce suction effects, but clearance must have minimum values for various sizes and speeds of discs, on account of disc fluttering and vibration, and of thermal expansions. End tightening of blades has been used to lessen suction effect in impulse blades. Suction effects also result from an inlet edge of the blade slightly wider than the nozzle mouth.

With partial admission, two losses occur. The steam jet must accelerate the dead steam in the blade passage when this comes in front of the nozzle opening. Eddies form during this action. The steam in the blade channel just passing from in front of the jet draws in clearance steam. If this clearance steam does not enter freely, some of the energy in the working steam delivered to this last channel is dissipated in the choking that follows.

SPEED RATIO.—An important factor in turbine performance is the speed ratio ρ, the ratio of the mean blade speed to the steam speed leaving the nozzle. Stodola states that turbine efficiency depends on this ratio alone, and not on the individual values of the velocities. For maximum theoretical efficiency of impulse wheels with no blade losses and no carry over, $\rho = \cos \alpha/2$ and for Curtis wheels, $\rho = \cos \alpha/2n$, where α = angle that the nozzle axis makes with the plane of the blades, and n = number of moving rows in the Curtis stage. Stodola indicates that highest efficiencies in a stage of a multistage impulse unit occurs when $\alpha = 12$ to $15°$.

Practice and consideration of blade losses and carry over show that values higher than the theoretical values of ρ give the best performance. Stodola (Steam Turbines, pp. 250–254) indicates that ρ may equal 0.60 for maximum efficiency under certain conditions. When highest economy is not desired, or where other losses, as in single-stage units, offset the higher blade efficiency at high wheel speed, much lower values of ρ are used. Variations of ρ above or below its most efficient value result in decreased efficiency. These variations may result from changes in steam pressure, temperature, vacuum, or load, acting either independently or together. Hence it is possible to analyze the effect of these changes if the effect on ρ is computed. Sets of curves have been printed on various papers showing the relations between ρ and stage efficiency. These indicate trends, but the absolute values given should not be used unless all assumptions on which the curves are based are known.

When carry over is large, larger values of ρ than $\cos \alpha/2$ may give higher stage efficiencies when this efficiency is based upon the isentropic heat drop only in the nozzle.

PARSONS BLADING consists essentially of a series of converging nozzles. The absolute exit velocity, or carry over, of the previous row is added to the velocity

produced by the pressure drop across the blade. Fig. 21 is a typical velocity diagram for a Parsons stage of one stationary and one moving row.

Provision is made for 50% reaction in usual designs, *i.e.*, half the heat drop is expended in the stationary guide blade and half in the moving blade. With this construction, $V_1 = V_{ro}$; $V_{ri} = V_o$; the discharge angle α is the same for both rows, varying from 18° to 23°. A usual value is 18 1/2°. Speed ratio $= \rho = u/V_1$. The diagram work per stage $= 2\,(V_1/223.8)^2 \times (2\rho \cos \alpha - \rho^2)$ B.t.u. The efficiency of this stage is proportional to $(2\rho \cos \alpha - \rho^2)$. It depends principally on ρ, as the outlet angle usually is a fixed quantity. This efficiency is a maximum at about 91% with $\alpha = 18.5°$, when $\rho = 0.94$, the efficiency decreasing if ρ decreases. In practice, ρ varies from 0.6 to 0.85 in land turbines, and seldom is less than 0.75 in large units. The higher values of ρ involve expensive, but economical turbines. Lower values of ρ give smaller, and less efficient machines. The work done $= (u V_w/(778\, g)$ B.t.u. per lb., where V_w is found as in Fig. 21*B*.

Losses in Parsons Blading.—One of the principal losses in Parsons blading is leakage over the blade tips, due to the difference in pressure across the blade row. The leakage passes directly across the clearance, while the working steam in the blades is turned through the angle α. The percentage loss from leakage through the clearance is

$$L = 100C \div (lM \sin \alpha + C),$$

where L = loss, percent; C = clearance, in.; l = length of the blade, in.; α = exit angle of blade; M = thickness coefficient = unity for most Parsons blading.

The steam passing through the blades $w_1 = w/C_1$ where w = total steam per hr. in lb. flowing through the turbine, and the clearance factor $C_1 = (1 + C\,/\,l \sin \alpha)$. The bucket efficiency $\eta_b = \eta_d/C_1$ where η_d = the diagram efficiency as calculated from a diagram similar to Fig. 21.

Leakage losses are greatest with short blades and proportionately large clearances,

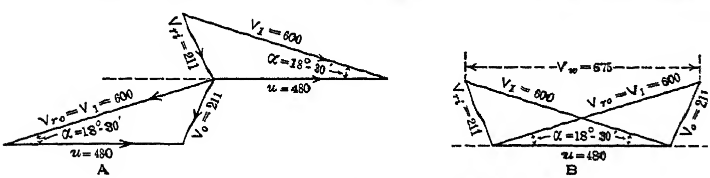


FIG. 21. Velocity Diagram for a Parsons Stage

and become comparatively small for the long blades of the last rows. A Curtis stage sometimes replaces the high-pressure section of Parsons turbines where the leakage losses are high, thus forming a Curtis-Parsons turbine. In Curtis-Parsons turbines the clearance on the shortest blades is never less than 0.025 in. With no clearance the efficiency of these blades is substantially that of nozzles, ranging from 92 to 95%. Clearance losses reduce this in standard turbines to 85–90%.

The face and rear flanks of Parsons blading are made up of a series of curves. One effective form is made from the intersection of two ellipses. On account of low entrance velocities, the entrance edge of Parsons blades can be quite blunt. To reduce erosion in low pressure blading, a blunt bull-nosed entrance edge has been successfully used.

Widths of Parsons Blades vary, with the different manufacturers, from 3/8 in. for blades 4 in. long, to 1 1/2 in. for blades 30 in. long. The pitch of blade rows is fixed somewhat by the type of shroud used to secure end packing, as it must be possible to shift the spindle toward the exhaust to clear these shrouds in lifting the rotor.

AREA THROUGH LAST ROW OF BLADES.—A factor of considerable importance in turbine design is the large volume of steam to be passed through the last row of blades, particularly when high vacuum will be used. If the area of the passage through the last row is too small, the steam will have a high exit velocity, involving a considerable energy loss. This area, therefore, should be as large as possible, while still retaining sufficiently small blade angles to insure good diagram efficiency with safe blade lengths.

Long low-pressure blades must be warped, *i.e.*, they must have varying entrance angles and cross-sections from base to tip. Kearton (Steam Turbines, p. 461) shows the details of such a blade for the last-stage wheel of a turbine operating at 1500 r.p.m. The ratio of blade length to mean diameter is 26.5%. The particulars of the blade are:

	Base	Mean	Tip
Mean diameter, in.	86	117	148
Peripheral velocity, ft. per sec.	563	766	970

The velocity diagrams at base, mean diameter, and tip are shown in Fig. 22. The inlet angles to the blades as required by the diagram varied from 38° at the base to 55° at the tip, with the outlet angle of 36° at tip and center and 44° at base. The blade can be formed by drop forging in a warped shape to give these angles, or by milling from solid bars with special convex- and concave-shaped cutters.

Kearton (Steam Turbines, p. 262) gives the centrifugal stress, lb. per sq. in., in steel blades of uniform cross-section as $f_c = 4.09\, l \left(\frac{d}{100}\right) \times \left(\frac{N}{100}\right)^2 = 0.215\, u^2 m$ for steel blades, where d = mean diameter, in.; N = r.p.m.; l = length of blades, in.; u = wheel speed, ft. per sec.; m = ratio of blade length to mean diameter. He also shows that $f_c = 1.88\, A\, (N/100)^2$, where A = area of annulus of last blade row, sq. ft. From this it is evident that the stress in the blades of the last row is directly proportional to the annular area of the blade ring for a given speed. These factors should be multiplied by 1.074 for brass or bronze blades.

CONSTRUCTION OF BLADING.—Blades may be rolled, drawn, or drop forged, or they may be machined all over from rectangular bars. They may be fixed in place by caulking pieces, or the base of each blade may form its own spacer. Blades of some impulse turbines straddle the disc, being held in place by rivets. DeLaval mills a bulb-shaped end on a straight shank at the blade base and drives these into similarly shaped grooves on the edge of the discs. The inverted T-base with one or more sets of shoulders is a common form for impulse turbines. These blades are entered into the groove on the disc at a slot which is closed by special devices when the last blade is entered. Modifications of the simple inverted T-base also are used. General Electric uses a straddle-base for its long blades which fits over a T-shaped rim on the disc. Sometimes the straddle ends of the blades are welded to the disc.

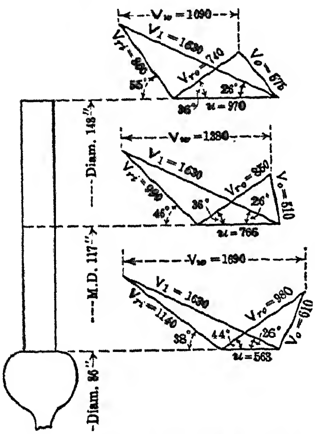


FIG. 22. Velocity Diagrams through Blading

Serrated grooves to receive blades and distance pieces are common with Parsons blading. Another form has a dovetailed groove into which the blades and distance pieces are inserted. These were formerly tightened by caulking. Westinghouse upsets a foot on the lower end of certain blades, which foot projects under the distance piece and is in turn locked in the dovetail groove by special driving wedges on one side. Allis-Chalmers sets the shorter blades cut from a rolled strip in a jig to insure uniform blade spacing and correct blade angle, and an alloy foundation ring is cast around their bases. A shroud ring then is silver-soldered to the outer end of the blades. The foundation rings are machined with one or more projections to fit under shoulders on one side of the groove and then balanced. A soft metal caulking strip on the other side of the groove holds the ring in place. No sharp corners or edges are permissible on any part of the blade, being a source of weakness from vibration fatigue. Westinghouse uses side entry blades on large low-pressure stages, with each row on a separate disc. These blades have serrated roots entering grooves milled nearly axially in the rim of the wheel, and as each blade is driven into place it firmly locks the preceding blade.

Parsons have developed a hollow low-pressure blade rolled from billets of stainless steel in which the hole tapers with the taper of the blade, and with the root integral with the blade. The walls gradually thicken towards the base. This construction substantially reduces the weight of the blades. With equal centrifugal stress these blades can operate at 15% greater blade speeds than solid blades, thus permitting a larger annulus at the last blade row and a greater output.

Blades of Parsons turbines are strengthened against vibration by lacing wires, silver-soldered to the blades at experimentally-determined points along their length. Copper-nickel, 5% nickel-steel, and pure nickel have been used for these wires. Shrouds on the ends of blades also lessen vibration.

Gaging of Reaction Blades is the ratio of the net area for steam flow on the mean diameter measured at the blade outlet at 90° to the direction of the jet, to the area of the

annular space occupied by the blade ring. Thus, a 25% gaging means a net area for flow of 25% of the blade ring annulus.

In some turbines, blades are adjusted by twisting, with special tools, after final assembly, to rectify irregularities due to workmanship. With Kingsbury bearings, it is no longer necessary to re-gage blades to correct end thrust.

Shrouds.—Some blades, particularly small ones, are formed with projections at their tips to form their own shroud rings. Shroud strips may be held by riveting over the tenons machined at the ends of the blades.

Shrouds may consist of flat strips in impulse turbines with a clearance of 0.03 in. between sections. Where radial clearance over the blade tips is of importance, as on Parsons blading, channel-shaped shrouds sometimes are used, with narrow channels projecting toward the casing to form small clearance.

Sometimes flat shrouds, which project on one side toward the base of an adjacent row, are used for end-tightening. The clearance between shroud and blade base can be adjusted to a much lesser amount than the permissible end clearance of similar blades. The reduced leakage with such shrouds results in better turbine economy.

For further information on blading materials, see: Materials and Design of Turbo-Alternator Plant, Lasche and Kieser, translated by Mellanby and Cooper; Brown Boveri *Review*, Jan., 1934, p. 30.

Strength of Blade Roots.—All blade roots must be sufficiently strong to withstand the stresses due to centrifugal force. Methods of calculating such stresses can be found in Stresses in Turbine Blading, by G. Stoney, *Engg.*, April 26, 1918, or in Goudie's Steam Turbines. The centrifugal force on a blade is

$$F = 0.00002842\,N^2\,Wr,$$

where N = rev. per min.; W = weight, lb., of blade and shroud ring (if any) beyond the section of the base carrying the load; r = radius, in. from the center of the shaft to the center of gravity of the blade and shroud figured beyond the section under stress. At normal speeds, blades should not be stressed greater than 0.5 their elastic limit. This allows the turbine a certain overspeed without risk of blade troubles.

Blade Failure may be due to the presence of some foreign substance in the casing, to defective material, to poor workmanship or to vibration. The first cause can be avoided by care in assembling the machine. All reliable manufacturers give particular attention to avoiding the second and third causes and are now devoting much study to vibration. See Vibration, Vol. 3 of this series.

Corrosion.—Certain blade materials corrode badly if the feedwater is not deaerated to free it of oxygen and carbon dioxide. Corrosion often is rapid in idle turbines into which a small amount of steam leaks through the throttle valve. If kept dry, corrosion should not occur. Kerosene injected at the throttle valve of the turbine before shutting down, gives the blades an oily coating and prevents corrosion. Aluminum-bronze blades have corroded badly in turbines receiving wet steam carrying magnesium and calcium chlorides.

Erosion of the inlet edges of low-pressure blades in turbines with high tip speeds has required the renewal of these blades in from 3 to 7 years. This is due to impact of water drops formed as a result of expansion. F. W. Gardner (The Erosion of Steam Turbine Blades, *The Engineer*, Feb. 5, 12, & 19, 1932) shows that the drops of water must be extremely fine; that they tend to concentrate on the outer ends of the blades; that the force of impact of the drops on blades moving 1000 ft. per sec. is about 90,000 lb. per sq. in.; that it is impossible to remove by any separating device all the water that condenses in these blades; and that hard blades and hard sheaths on blades are necessary to withstand this erosion. See Brown Boveri *Review*, Jan., 1934, p. 29, for curves showing braking losses caused by moisture in blading, given in percentage of the useful output at different pressures and temperatures. Curves are shown of improvements resulting from drainage grooves.

Christie and Colburn (*Turbines*, N.E.L.A., 1932) found that erosion was most serious in turbine blades with tip speeds over 1000 ft. per sec.; that it is more pronounced in reaction than in impulse blading; and that erosion is most severe in about 1 to $1^1/_2$ in. from the blade tip and around binding wires. They also show how steam conditions at the entrance to the last blade row may be estimated.

The use of hardened strips fastened or brazed to the inlet edges of the blades, of coatings of hard alloys as stellite, tungsten tool steel, tantalum, etc., fused onto the blade flank, of heavy chromium plating, of drainage systems to remove interstage moisture, of hard alloy steel blades, and of combinations of these, are all being tried to reduce erosion. Water drainage grooves are provided around the casing of some units to draw off water formed by condensation on blades and nozzles. These water drainage devices are said

to remove 25 to 30% of the moisture present. The low-pressure bleeder point also serves to draw off such water.

Blade Design must avoid any sudden change of form or corners where stress may be concentrated. Changes of section are made very gradually. The heat treatment and method of manufacture, such as drawing, forging, rolling, or pressing, must be such that the blade is not under any internal stress. The shaping of blade roots should be done by milling from the bar or forging. Pressing or stamping sets up internal stress in the blade root and is undesirable. In many cases, the distance-piece is made an integral part of the blade root, thereby increasing its strength.

Blade Material.—Brass has been used for blading of small turbines operating at low pressures and temperatures. Steel with 4.5 to 5.5% nickel has been used for highly stressed blading. When heat treated, such steel has an ultimate strength of 70,000 to 90,000 lb. per sq. in., and an elastic limit of 45,000 to 50,000 lb. per sq. in. Cupro-nickel (78–82% Cu, 22–18% Ni) with ultimate strength of 70,000 lb. per sq. in., elastic limit 50,000 lb. per sq. in., is widely used for Parsons blading where steam temperatures and blade stresses are comparatively low. Manganese-copper (4–5.5% Mn, 94.5–96% Cu), ultimate strength, 60,000 lb. per sq. in., elastic limit, 31,000 lb. per sq. in., has been utilized for low-pressure blading and temperatures under 450° F. Monel metal, ultimate strength, 85,000 lb. per sq. in., elastic limit, 50,000 lb. per sq. in., has been used with moderately high steam temperatures and stresses, as it does not readily corrode. Stainless steels (11.5 to 13% Cr, 0.10 to 0.15% C), ultimate strength, 90,000 to 100,000 lb. per sq. in., elastic limit, 60,000 to 70,000 lb. per sq. in., have been applied to blading subject to high temperature and in highly stressed low-pressure blades subject to corrosion. Some heat-treated blade material has an ultimate strength of over 100,000 lb. per sq. in. and an elastic limit of 70,000 lb. per sq. in. Other materials being tried for blading are chrome-nickel-, tool-, Hecla A.T.V., and chrome-vanadium steels and nickel-bronze. The choice of blade material depends on the steam temperature, length and stress of blade, and character of boiler feedwater. McVetty, Working Stresses for High Temperature Service, *Mech. Engg.*, Mar., 1934, presents data on allowable stresses where creep must be considered.

For further information on blading materials, see Materials and Design of Turbo-Alternator Plant, Lasche and Kieser, translated by Mellanby and Cooper.

Blade Clearance.—As the same pressure exists on both sides of impulse blades, the clearance over their ends may be large. The axial clearance between nozzle mouth and blade on some small impulse turbines varies from $^1/_{32}$ in. to $^1/_{16}$ in. Large machines have axial clearances up to $^1/_2$ in. Suction effects make it desirable to keep this clearance as small as mechanically possible.

The radial clearances over the ends of Parsons blades must be kept as small as possible, due to the pressure difference across each row. Consideration also must be given to the rigidity of the casing. Various formulas are used by builders of Parsons turbines for determining this clearance. Some formulas are based upon distance between bearings, others upon blade length and mean diameter of row, with consideration of the taper on the ends of the blades. An older formula for blades without taper at the top is

$$C = 0.015 + 0.003\,D + 0.005\,l_b,$$

where C = clearance, in., D = mean diameter of blade ring, ft., l_b = length of blades, in.

Stodola (The Efficiency of Reaction Blading, *Engg.*, Oct. 2, 1925) quotes one builder of Parsons turbines as allowing on large single-cylinder turbines a clearance of 0.001 × mean diameter, and on short two-cylinder units 0.0008 × mean diameter.

The clearance of end-tightened blading can be set at substantially that of the dummy pistons. This usually will be less than half the end clearance given by the above formula. The gain in efficiency by the use of end-tightened blades results from this reduction.

The axial pitch of blade rows on Parsons blades is given by the formula

$$P_a = 2B + 0.25 + (l_b/16),$$

where P_a = axial pitch of the blade rows, in., B = width of blades, in., l_b = length of blades, in. Greater pitch may be required by end-tightened blading.

Since leakage is one of the principal losses in Parsons turbines, great attention is given to maintaining clearances at as low a figure as is consistent with reliable operation. With end-tightened blading or blading in a conical cylinder, it is important that the clearance be made small enough to maintain high efficiency under load. This may be done by moving the thrust between limit stops, thus providing large clearance during the starting period and moving the thrust to secure minimum clearance and high efficiency when under load.

Blade Length.—Maximum blade lengths have been increased (1935) on the last low-pressure wheels to 35% of the mean diameter of the blade ring, while tip speeds have

increased to about 1250 ft. per sec. Further developments in blade materials may permit still longer blades. See Turbines (N.E.L.A., 1932) for data upon maximum blade lengths and tip speeds. The minimum blade length is from 1.5 to 2% of the mean diameter.

THE EFFICIENCY AT THE WHEEL PERIPHERY is found by combining the nozzle efficiency η_n with the diagram efficiency η_d. Stodola (Steam Turbines, p. 222) makes the following statements on the efficiency at the wheel periphery: 1. Efficiency depends only on ρ, the ratio of wheel speed to steam speed, and not on the individual values of the velocities. 2. Efficiency varies with the peripheral velocity according to a parabolic law with the maximum values over a narrow range. 3. With a small nozzle angle α the maximum value of efficiency at the wheel periphery is attained when the peripheral velocity is nearly half of the steam velocity. 4. The best efficiency for constant blade coefficient k_b is higher the smaller the nozzle angle α. 5. The effect of energy loss in the nozzle makes itself felt nearly four times as much as the effect of energy loss in the blades.

NOZZLE AND BLADE EFFICIENCIES IN TURBINE DESIGN.—Data upon nozzle and blade losses are not complete, and the factors given above can be used only with judgment. Experience has indicated that the use of such factors in turbine design may be misleading. Many designers start with internal efficiencies based upon tests of similar units from which stage efficiencies can be calculated, when the reheat factor is known. Then blade and nozzle efficiencies can be deduced with considerable reliability. Such deductive design based upon test performance is preferable to synthetic design using assumed efficiency factors. Detailed methods of proportioning nozzles and blading will be found in Stodola, Steam Turbines; Goudie, Steam Turbines; Zietemann, Dampfturbinen; Dampfturbinen; and Forner, Thermodynamische Berechnung der Dampfturbinen.

5. STEAM SUPPLY TO TURBINE

The steam supplied to turbines from boilers should be free from moisture, dust, acid, and corroding chemicals and as nearly pure as possible. Any corrosive element in the steam will destroy rapidly turbine blading, starting at the dew-point in the blading. Blade surfaces are continually swept clear by the high velocity steam, thereby accelerating this corrosive attack.

Although feedwater in modern plants consists of condensate and evaporated make-up water, impurities may be carried into boilers from condenser leakage. Chemicals must be added to maintain the desired sulphate-alkalinity ratio so that the average concentration in the boiler drums may range from 1500 to 6000 parts per million. Due to the evaporation of any moisture in the superheater, some impurities are carried over as dust or vapor and deposited on the governor valves, nozzles and blades of the turbine, closing off the passages and decreasing the turbine capacity. The decreased capacity may range from 15 to 50% in a few weeks in some cases. When this scale consists of soluble salts, it may be removed by washing with water during the overhaul period, by partly filling the turbine with water and slowly revolving the spindle or by dissolving the deposits with wet steam. The last operation is accomplished by slowly desuperheating the steam at the rate of 100° F. drop per hr. with the turbine in operation until wet steam enters the throttle. In some cases load is removed and speed reduced for this operation. This wet steam is allowed to pass through for one to two hours, thoroughly washing off all soluble scale, when superheat is slowly restored. Washing with kerosene also has been done on certain units. (See Turbines, N.E.L.A., 1931.)

On certain high-pressure turbines this blade deposit consists of a scale made up largely of silica and iron oxide, which has to be scraped off. This trouble has been overcome by reducing the boiler concentration and placing a separator on the line from the evaporator.

6. ROTORS

Rotors of small high-speed impulse turbines usually consist of a disc or wheel, carrying the blades or buckets, pressed on a shaft and held against a shoulder by a lock nut. Some turbines built upon the Curtis principle and with low wheel speeds, use two or more discs, each with a single row of blades, instead of one disc with several rows, which is a less expensive construction. Such turbines generally are designed with shafts sufficiently large so that they operate well below their critical speed.

Multi-stage impulse turbines have a series of discs mounted on the shaft, with intermediate diaphragms between to carry the stage nozzle, and usually, the labyrinth. The discs have holes for equalizing the steam pressure on the two sides. The edges of all such holes should be well rounded. The surfaces of the discs should be turned smooth and preferably polished.

Discs of Curtis Wheels carrying two rows of blading have a rapidly tapering section and a heavy hub. In single-stage machines, the blade speed at the mean diameter usually varies from 200 to 500 ft. per sec. Large machines have had Curtis wheels running at mean blade speeds of 600 to 650 ft. per sec. Many builders of compound turbines keep this mean blade speed low to reduce the windage loss of the idle blades, as well as to reduce the disc friction.

Discs for Multi-stage Impulse Turbines are made either of uniform thickness, of a conical section, or of a hyperbolic tapering section. With only a relatively light rim, the stresses are less than in the Curtis wheels. They operate at mean blade speeds varying from 400 to 900 ft. per sec., and with tip speeds up to 1200 ft. per sec.

DISC MATERIAL.—Discs on small turbines with low wheel speeds may be cut from rolled boiler plate of ultimate strength of 65,000–70,000 lb. per sq. in., and elastic limit of 30,000 to 40,000 lb. per sq. in. For larger discs, quenched and tempered carbon steel forgings of elastic limit of 50,000 lb. per sq. in., or quenched and tempered nickel-steel forgings of elastic limit of 65,000 lb. per sq. in. are used.

Gibb (Post War Land Turbine Developments, *Proc.* Inst. Mech. Engrs., 1931) gives full specifications for rotor forgings of C. A. Parsons and Co., England. These specify exhaust-end turbine discs of 100,000 to 110,000 lb. per sq. in. ultimate strength.

Carbon steel forgings are used with ultimate strength, 75,000 lb. per sq. in.; elastic limit, 28,000 lb. per sq. in.; elongation, 20 to 24%; reduction in area, 35 to 40%.

Where stresses are high, chrome-vanadium-steel discs, forged and heat treated, have been used that have a yield point not less than 70,000 lb. per sq. in., an ultimate strength of 100,000 lb. per sq. in., an elongation of at least 19%, and a reduction of area of not less than 45%. Extreme care is taken in the selection and testing of test pieces and in disc inspection. Guy and Jones (Metropolitan-Vickers Rateau Marine Turbine, *Engg.*, Feb. 15, 1923) give for forged discs: Ultimate strength, 107,500 lb. per sq. in.; yield point, 67,000 lb. per sq. in.; elongation, 17%; reduction of area, 40%; P and S each less than 0.05%. At 25% overspeed, the maximum working stress should not exceed the elastic limit. At normal speed this allows a working stress of 16,000 to 20,000 lb. per sq. in. Stresses at normal speed are figured with a factor of safety of about 2.

Certain manufacturers over-stress the discs by operating them at overspeeds. The discs then are allowed to age, resulting in an increase in the elastic limit of the material.

Care must be taken in forging the discs to thoroughly work all of the metal. No subsequent heat treatment can make up for lack of proper working. Discs are rejected for sponginess, dirty steel, blown steel, and slag inclusions that cannot be scraped out.

DISC DESIGN.—Discs usually are designed by assuming a thickness of disc under the rim, such that it will not buckle or bend during machining and erection. This thickness may be 3/8 in. on small wheels, increasing to 1 in. on some Curtis wheels and large-diameter impulse discs. The thickness at other points in a disc of hyperbolic profile is found from the equation $t = Cr^a$, where t = thickness, in.; r = radius at desired point, in.; C = a constant found for the conditions under the rim. The exponent a varies from (-0.4) to (-0.8) in multi-stage impulse wheels, and is taken as (-1) for Curtis wheels. Wide variations exist in the disc designs of various builders. Some discs have been made too thin, and have given trouble in service due to nodal vibration.

The first step in disc design is to assume the thickness under the rim. A value of exponent a then is chosen, depending on the type of disc desired. Since r is known, the value for the constant C can be found when the thickness is chosen. The thickness at any other radius readily can be calculated from the above equation. Present practice (1935) on large turbines is to make the discs rather heavy, as these are less liable to vibration.

The theoretical hyperbolic curve near the hub is modified to one or more arcs of a circle with much greater curvature than the hyperbola. This is done to provide a heavy hub in which a keyway may be cut without weakening the disc section. The proportions of the rim are determined by the size, number of rows of blades and the methods of fastening the blades to the rim. The rim is connected to the narrowest part of the disc by a section of easy curved profile, frequently consisting of arcs of circles of relatively short radius. The flanks of the discs between the curved sections at hub and rim are sometimes made straight taper, or wedge-shaped for easier machining. This form is slightly stronger than the hyperbolic profile, and is more easily made. The departures from true hyperbolic form make the solution of disc stresses complicated and tedious.

Rotors for small turbines for high pressures and temperatures and for very high speeds frequently are made complete from a solid forging, with the discs turned to form by removing the metal between them. This gives a strong, rigid construction.

DETERMINATION OF DISC STRESSES.—After preliminary designs are finished, calculations are made to determine whether the discs have sufficient strength. Three

stresses, radial, tangential, and axial, may act at any given point. The axial stress is of negligible value if there is no sudden change in axial thickness, as at a hub. Radial and tangential stresses can be calculated by neglecting the axial stress. Stodola in Steam and Gas Turbines developed formulas for the determining of these stresses, but which are complicated in the form presented.

S. H. Weaver (Disc-wheel Stress Determinations, *Trans.* A.S.M.E., xxxix, 173) describes a method of calculation which is simpler and more readily applied. For machining purposes radial sections of discs usually consist of straight lines and arcs of circles. The equations of these lines present mathematical difficulties in calculating stresses. Hence the section is assumed to consist of one or more hyperbolas of the equation, $t = cr^a$. In this equation the exponent a is the shape-constant of the profile and has a negative value when the thickness decreases with a larger radius; a zero value for a constant or uniform thickness; and a positive value when the thickness increases with a larger radius. For a given disc profile, Fig. 23, the value of a may be found from

$$a = (\log t_2/t_1) \div (\log r_2/r_1) \quad \text{or} \quad (-\log t_1/t_2) \div (\log r_2/r_1).$$

One or the other of these formulas is chosen which will give one or more values for the ratio of the thicknesses.

Stodola's equations for tangential and radial stress are as follows:

Notation.—m_1, m_2 and p are algebraic quantities as given in the equations of Group II, below; a = shape constant of profile of the particular portion of the disc section; V = Poisson's ratio of deformation = 0.3 for steel; E_1 = Young's modulus of elasticity; R = radial stress at radius r, lb. per sq. in.; T = tangential stress at radius r, lb. per sq. in.; r = any radius in disc section, in.; b_1 and b_{11} = boundary condition constants; ω = angular velocity of rotation in radians; u = mass of disc material per unit of volume = 0.2815 lb. per cu. in. for average steel; y = radial elongation, in.

$$R = \frac{E_1}{1 - V^2}\{(3 + V)\,pr^2 + (m_1 + V)b_1 r^{m_1 - 1} + (m_2 + V)b_{11} r^{m_2 - 1}\},$$

$$T = \quad E_1 \qquad\qquad {}^{-1} +$$

It will be necessary to know two stresses in order to determine the values of the condition constants, b_1 and b_{11} and to transform these equations. Known radial stresses R_1 at radius r_1 and R_2 at radius r_2 are assumed. The tangential stresses at r_1 and r_2 are

$$T_1 = Ar_2^2 - BR_1 + CR_2, \qquad \text{(I)}$$

where A, B, C, D, E and F have the values given in Equations (II).

The stresses due to the external centrifugal load and the weight of the disc itself vary as the square of the speed. If all stress values are calculated for 1000 r.p.m., the stresses at any other speed can be found by multiplying by the square of the speed ratio.

The following formulas (Group II) based on 1000 r.p.m., assist in solving for the various stresses:

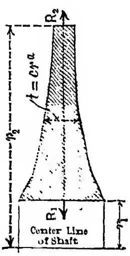


Fig. 23

$$\frac{\log}{\log(1/K)}, \qquad \frac{\log}{\log(1/K)}.$$

$$-\;0.3a + 1;$$

$$m_2 = -(a/2) + \qquad -\;0.3a + 1;$$

$$B$$

$$E \qquad \text{(II)}$$

$$B + a;$$

$$\frac{3.3(C -}{1 + 0.4125a}$$

$$\frac{3.3(F - K^2E) - 1.9}{1 + 0.4125a}.$$

The factors A, B, C, D, E and F are functions only of the shape constant a and the ratio of radii K.

These factors become relatively simple for the portions of the disc where the wheel section is of constant thickness, as at the hub and sometimes at the rim. At 1000 r.p.m. the functions reduce to the following (Group III) equations:

$$0. \qquad K = \frac{r_1}{}. \qquad C = \frac{}{1 - K^2}.$$
$$B: \quad F = C - 1. \qquad C - 2.$$
$$A: \quad 6.6 + 1.4K^2. \qquad 6.6K^2 + 1.4.$$

Since the determination of the values of the functions takes time in making a stress calculation, Weaver has prepared the following approximate equations (Group IV) for the more rapid determination of these functions by the use of common logarithms. The error in these equations is about 0.7% as a maximum.

$$B = \frac{\log (1/K)}{5.43}(a^2 - 1.2a) - (a/2) +$$
$$F = B + a;$$
$$E = \qquad \frac{}{10}(a^2 + 10a) - (a/2) + \frac{2}{1 - K^2} - 2\Big),$$

between the limits 0.8 and 0.1 for K, and (-5) and 0 for a;

between the limits 0.97 and 0.8 for K and 40 and 0 for a; . . . . . (IV)

between the limits 0.8 and 0.4 for K and (-5) and 0 for a;

between the limits 0.97 and 0.8 for K and 40 and 0 for a;

$$A = 3.1(1 - K)^{2.09} a + (6.6 + 1.4K^2);$$
$$D = 1.25(1 - K)^{1.75} a + (6.6K^2 + 1.4).$$

These functions can be plotted in simple alignment charts with negligible errors (see *Gen. Elec. Rev.*, Oct., 1917; Moyer's Steam Turbines).

The following example shows the method of applying these formulas. Fig. 24 is a half section of a disc designed to run at 3600 r.p.m. and carry two Curtis blade rows. The disc is divided into five sections, 1, 2, 3, 4 and 5. The curved profiles are assumed to be portions of hyperbolas whose a value can be found from Equation II for each section. All sections are assumed to have the same thickness where each joins the adjacent section. Rings 1 and 5 are of uniform thickness, hence $a = 0$. Ring 4 has a positive value of a, since thickness increases rapidly with the radius; a is negative in sections 2 and 3, since the thickness decreases as the radius increases.

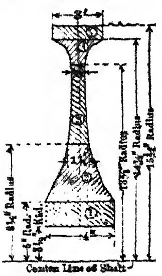


FIG. 24

Two of the radial stresses must be known. The radial stress at the bore may be taken as zero, as the shrink fit is supposed to be almost neutralized at normal speed by the centrifugal expansion of the bore, and at some over-speed the radial stress is zero. The outer radial stress of the blade load equals the centrifugal force of blades, shrouds, etc. The centrifugal force is

$$\text{C.F.} = 0.00001421\ N^2\ wd_1,$$

where N = rev. per min.; d_1 = diameter, in., to center of gravity of blades; w = total weight of blades, shrouds, etc., lb. The centrifugal force per inch of circumference at diameter d_2 at the edge of the disc = C.F. $\div\ \pi d_2$. This is assumed in the problem at 172 lb. per in., at radius $r = 15\,^3/_4$ in.

The unknown radial stresses at the lines dividing the imaginary rings are taken as e between rings 1 and 2, as f between rings 2 and 3, as g between rings 3 and 4, and as h between rings 4 and 5. The data may now be collected in Table 2. Constants A to F may be calculated from Equations (II) and (III); approximate results may be calculated from Equations (III) and (IV) or may be read from the alignment charts referred to above. The values given in Table 2 are calculated from Equations (III) and (IV).

There is only one thickness at any one radius owing to the construction of the disc. Hence, at the dividing line between any two imaginary rings, there can be only one radial stress and one tangential stress. Take the line between rings 1 and 2, at radius 5 in. The outer tangential stress of ring 1 must equal the inner tangential stress of ring 2. Hence,

$$T_2 \text{ (ring 1)} = T_1 \text{ (ring 2)}.$$

or $$(Dr_2^2 - ER_1 \cdot \qquad)_{\text{(ring 1)}} \qquad {}_1 + CR_2)_{\text{(ring 2)}}.$$

Similar equations can be written for each imaginary line at the given radius separating rings, substituting the values of radial stresses, e, f, g and h assumed above. These four equations next

can be solved for the unknown radial stresses e, f, g and h. The equations for the equal tangential stress at the various radii are as follows:

At radius 5 in.:

$$4.64 \times (5.0)^2 - 1.92 \times 0 + 2.92 \times e = 6.02 \times (8.5)^2 - 3.47 \times e + 1.57 \times f.$$

At radius 8.5 in.:

$$3.11 \times (8.5)^2 - 1.74 \times e + 1.28 \times f = 6.59 \times (13.375)^2 - 3.27 \times f + 2.3 \times g.$$

At radius 13.375 in.:

$$3.74 \times (13.375)^2 - 1.87 \times f + 1.74 \times g = 8.11 \times (14.875)^2 - 3.73 \times g + 20.86 \times h.$$

At radius 14.875 in.:

$$7.06 \times (14.875)^2 - 3.43 \times g + 18.51 \times h = 7.85 \times (15.75)^2 - 17.35 \times h + 18.35 \times 172.$$

These equations can be solved easily by a substitution method as follows:

From equation for 5 in. radius,

$$e = 0.246f + 49.9.$$

Substituting this value of e in the equation for 8.5 in. radius and solving,

$$f = 0.558g + 252.5.$$

Substituting this value of f in the equation for 13.375 in. radius and solving,

$$g = 4.71h + 360.8.$$

When this value of g is substituted in the last equation it is found that

$$h = 243.$$

Substituting in the three preceding equations, the following values are found:

$$g = 1503, \quad f = 1091, \quad e = 318.$$

The tangential stresses at the various radii can be found by substituting in either side of the foregoing equations the values of e, f, g and h as follows:

At radius 5 in.: $T_5 = 4.64 \times (5.0)^2 + 2.92 \times 318 = 1045.$

At radius 8.5 in.: $T_{8.5} = 3.11 \times (8.5)^2 - 1.74 \times 318 + 1.28 \times 1091 = 1068.$

At radius 13.375 in.: $T_{13.375} = 3.74 \times (13.375)^2 - 1.87 \times 1091 + 1.74 \times 1503 = 1244.$

At radius 14.875 in.: $T_{14.875} = 7.85 \times (14.875)^2 - 17.35 \times 243 + 18.35 \times 172 = 894.$

(Note.—Some decimals have been dropped to simplify the solutions.)

The tangential stress at the bore is found as follows:

$$T_{1_b} = Ar_2^2 - BR_1 + CR_2 = 7.29 \times (5)^2 - 2.92 \times 0 + 3.92 \times 318 = 1430.$$

At the rim:

$$T_{2_b} = Dr_2^2 - ER_1 + FR_2 = 7.28 \times (15.75)^2 - 16.35 \times 243 + 17.35 \times 172 = 817.$$

The stresses at 1000 r.p.m. are as follows:

At radius	3.5	5	8.5	13.375	14.875	15.75
$R =$	0	318	1091	1503	243	172
$T =$	1430	1045	1068	1244	894	817

The stresses at intermediate points in any ring can be computed from the known values of radii and thickness at these points by Equations (I).

At 3600 r.p.m. the stresses above should be multiplied by the square of the ratio of the speeds, *i.e.*, by $(3600/1000)^2 = 12.96$.

The resultant stresses are as follows:

At radius	3.5	5	8.5	13.375	14.875	15.75
$R =$	0	4,121	14,139	19,479	3,149	2,229
$T =$	18,533	13,543	13,840	16,122	11,586	10,588

Discs should be able to withstand an overspeed of 20% without exceeding the elastic limit. The stresses at this speed would be $(1.2)^2 = 1.44 \times$ stresses at normal load.

The maximum stress in the above table is 19,479 lb. per sq. in. at radius 13.375 at 3600 r.p.m. and at 4320 r.p.m. (20% overspeed) it would be 28,050 lb. per sq. in., which is within the usual elastic limit of steel used in discs. If this stress is considered excessive, the design may be modified by

Table 2.—Data on Stresses at Lines Dividing Imaginary Rings in Fig. 24

Ring No.	1	2	3	4	5
r_1	3.50	5.00	8.5	13.375	14.875
r_2	5.00	8.5	13.375	14.875	15.75
t_1	4.00	4.00	1.25	0.625	3.00
t_2	4.00	1.25	0.625	3.00	3.00
$K = r_1/r_2$	0.70	0.588	0.636	0.899	0.944
a	0	−2.19	−1.53	14.78	0
R_1	0	e	f	g	h
R_2	e	f	g	h	172
A	7.29	6.02	6.59	8.11	7.85
B	2.92	3.47	3.27	3.73	17.35
C	3.92	1.57	2.30	20.86	18.35
D	4.64	3.11	3.74	7.06	7.28
E	1.92	1.74	1.87	3.43	16.35
F	2.92	1.28	1.74	18.51	17.35

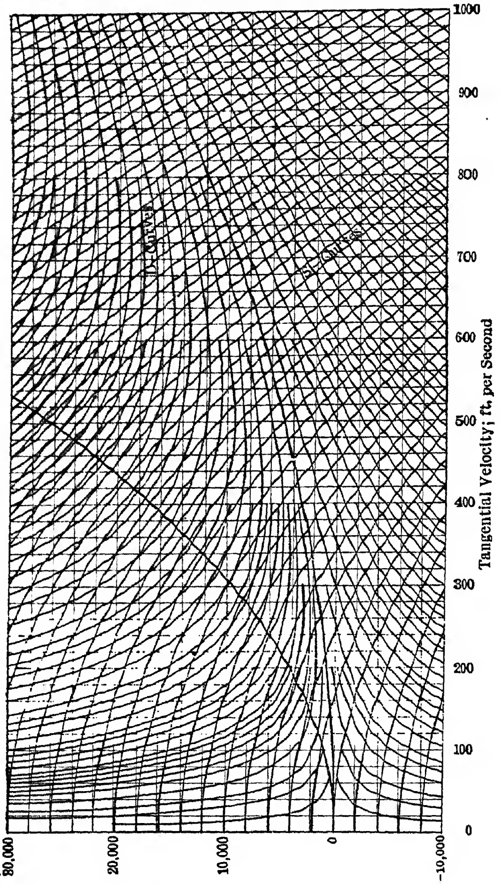
thickening the metal at this point and allowing a smaller taper on the disc. The radial elongation in inches at any radius r in inches is

$$y = (T - 0.3R)\ r + E_1,$$

where E_1 = modulus of elasticity for steel = 29,000,000 for usual disc material. The radial elongation is as follows at 3600 r.p.m.:

Radius..................	3.25	5	8.5	13.375	14.875	15.75
Elongation, y............	0.00212	0.00212	0.00281	0.00474	0.00546	0.00539

Weaver states that the approximate method generally gives stresses about 1% too high. The great advantage of this method is the reduction of the time required for computation.



Stress in lb. per Square Inch

FIG. 25. Disc Stress Diagram

H. Haerle (Strength of Rotating Discs, *Engg.*, Aug. 9, 1918) presents a simple method for the determination of disc stresses from a single diagram, shown in Fig. 25, which can be applied to any disc profile, and yield results sufficiently accurate for all practical purposes. The general formulas for disc stresses given by Stodola, and stated above in

discussing Weaver's methods, form the basis of Haerle's method. These formulas reduce to much simpler expressions when applied to a disc of constant thickness, in which case $a = 0$ in the expression $t = Cr^a$, where t = thickness of disc, in., at radius r, in., C = a constant, a an exponent governing the curvature and found by means of Equations (II) on page 8-27.

Let T = tangential stress, lb. per sq. in.; R = radial stress, lb. per sq. in.

Haerle's methods are based on the assumption that

$$S = T + R = \text{sum of principal stresses,}$$

and

$$D = T - R = \text{difference of principal stresses.}$$

The following equations derived from those of Stodola, apply to a disc of constant thick.

$$D = (1 - V)\frac{u}{g}\{$$

$$K_1 = \frac{}{(1 - V^2)u} \quad \text{and} \quad K_2 = \frac{}{(1 - V^2)u},$$

where V = Poisson's ratio (= 0.3 for steel); u = weight of disc material, lb. per cu. in.; $g = 32.2 \times 12$ (in.); ω = angular velocity, radians per sec.; U = tangential velocity of the disc, in. per sec.; E = Young's modulus (= 29,000,000 for steel); b_1 and b_2 are constants depending on stress conditions at bore and rim as in Stodola's formulas.

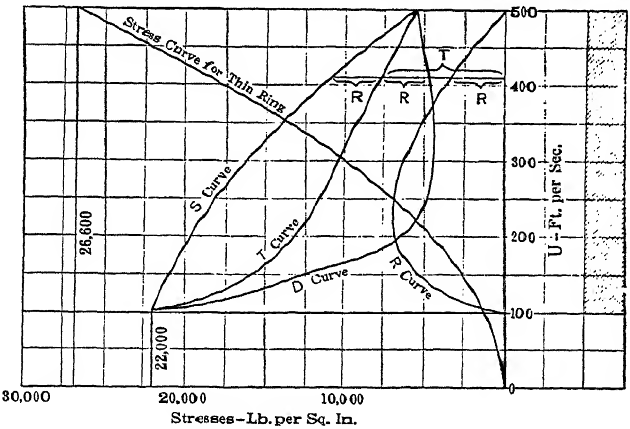


FIG. 26

The only variables at a given radius in the equations for S and D are K_1 and K_2. Series of curves as represented by the above equations are plotted in Fig. 25, each curve being based on a different value of K_1 and K_2, respectively. The problem is now simplified to that of selecting the proper curve or set of curves according to the specific details of the disc under consideration and finding S and D from the curves.

If S and D can be determined from the diagram, then

$$T = (S + D)/2 \quad \text{and} \quad R = (S - D)/2.$$

Fig. 25 is plotted with stresses as abscissae and tangential velocities as ordinates. The heavy line on the diagram curving toward the left is the stress curve for a thin ring.

The following examples show the application of this diagram: In the case of a disc of uniform thickness with a concentric bore and no load on the rim, the radial stress R at both bore and rim must be zero. Then

$$S = T + R = T \quad \text{and} \quad D = T - R = T \quad \text{or} \quad S = D = T$$

at both rim and at bore. That is, the S and D curves must intersect at both bore and rim, and these must be the same curves on the diagram at both places, for, since the boundary conditions are fixed, the values of K_1 or K_2 are the same throughout the disc. For example, let the rim speed be 500 ft. per sec., and the bore speed 100 ft. per sec. The same S and D curves must intersect at

the 500 ft. and 100 ft. ordinates, as shown in Fig. 26, at 5500 lb. and at 22,000 lb. respectively, which must be the tangential stress at these points, since the radial stress at both points is zero. Tangential stresses at other speeds may be plotted by bisecting the distance between the S and D curves at each speed. Radial stresses can be found by halving the difference between the S and D curves and plotting from the zero ordinate.

The ease and rapidity with which problems relating to discs of uniform thickness can be solved by this S and D chart suggests the application of this method to discs of other than parallel profile. If the hyperbolic or other profile is replaced by a stepped disc consisting of a number of concentric rings, each of constant thickness as in Fig. 27, this method can be applied.

The assumption is made that, provided the steps be comparatively small, the stresses in adjacent concentric layers on either side of the step are inversely proportional to the axial dimensions or thickness t' and t as in Fig. 27. Hence

$$\frac{t}{t'} = \frac{R'}{R} = \frac{T'}{T}.$$

Let ΔR denote the increment (positive or negative) of the radial stress at the step. Then

$$\Delta R = R - R' = R(1 - t/t').$$

The following expression is derived for ΔT:

$$\Delta T = V\Delta R = VR(1 - t/t'),$$

where V = Poisson's ratio = 0.3 for steel.

d

$$\Delta D = D - D' = -0.7\Delta R$$

r the change in the S and D curves at the step. It is now quite easy to determine the S and D

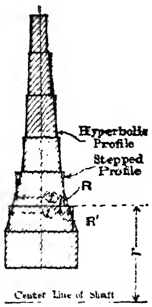


FIG. 27

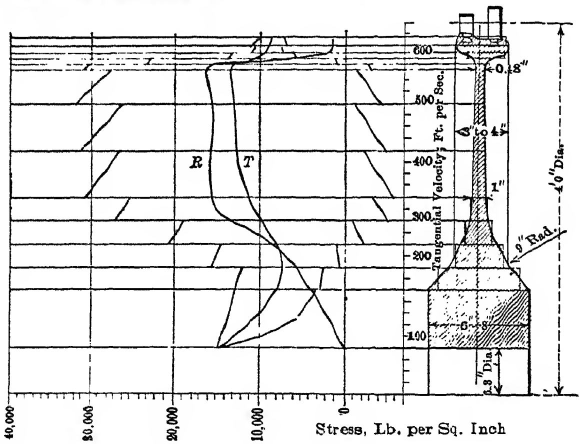


FIG. 28

values at all points on the irregular disc profile, as shown in the example, Fig. 28, when applied to a disc with tapered sides, for a Curtis stage with two rows of blades. The mean diameter is 48 in., and the normal speed, 3000 r.p.m. The stepped disc, in substitution for the actual profile, is shown by the fine lines. Assumed peripheral velocity of the disc proper, 615 ft. per sec. The net weight of blades, shrouds, etc., is assumed as 65 lb. at normal speed, exerting a centrifugal pull of 2860 lb. per inch of circumference. The net width of the disc at the periphery, after deducting for dovetail grooves for blades is t' = 2.2 in. Hence, radial stress due to blade load R (at periphery) = 2860/2.2 = 1300 lb. per sq. in. The tangential stress at the periphery may be assumed; let T = 9000 lb. per sq. in. Hence at the periphery,

$$S = 9000 + 1300 = 10{,}300 \text{ lb. per sq. in.}$$

$$D = 9000 - 1300 = 7{,}700 \text{ lb. per sq. in.}$$

These values constitute the point of origin of the S and D curves across the outermost step. It is assumed that R = 0 at the bore. Hence $S = D = T$ at the bore, *i.e.*, the curves intersect at the ordinate of the bore. If forced fit must be allowed for, R at the bore may be chosen with either a positive or negative value to suit the particular conditions.

Beginning at the tangential velocity of the periphery, follow the S and D curves across the radial extension of the first step, *i.e.*, from tangential velocity, 615 ft. per sec., to tangential velocity, 600 ft per sec. At step 600 read, $R = (S - D)/2$, find ratio t/t' from the substituted profile, and calculate ΔR, ΔS and ΔD. Plot the new S' and D' for the second step and continue the S and D

curves across it. The procedure is the same for all other steps. If tangential stress T at the periphery has been chosen correctly, the S and D curves for the innermost step must intersect at the bore of the disc. If they do not, another value of T at the periphery may be chosen and the process repeated until intersection occurs at the bore.

The axial thickness of the several concentric rings which form the substituted disc section correspond to the thickness of the true profile at points midway between the steps, and the stresses as determined by the S and D curves at these points coincide with the actual stresses in the original profile. Hence, if the horizontal distance between the S and D curves in the center of the stepped rings is bisected, points on the true tangential stress curve are obtained. A smooth curve through this series of points represents graphically the distribution and magnitude of the tangential stress T throughout the disc. The radial stress R at the middle of each step can be determined by measuring the distance of the tangential stress curve T from the S or D curves, and plotting this distance from the zero axis at the various radii. Another smooth curve through these points indicates the magnitude and distribution of the radial stress R. Both these curves are clearly shown in Fig. 28. Values of R and T have been scaled off from these curves at radii corresponding to the various peripheral speeds U in the first column of Table 3, and are tabulated in the last two columns for general information. They could be scaled at any other point if desired. The maximum stress in this disc is a radial stress of 16,600 lb. per sq. in. and it exists just below the rim.

M. G. Driessen (A Simplified Method of Determining Stresses in Rotating Discs, *Trans.* A.S.M.E., APM-50-10, 1928) discusses Haerle's method as above outlined and presents a larger chart for S and D curves than Fig. 25. The purpose of this paper is: 1. To shorten the steps necessary to pass from one element of the disc to the next. 2. To provide an alternative to this cut-and-try method. 3. To indicate a manner in which the results once obtained can be used for all other conditions under which this disc is used, *i.e.*, for different loads at the circumference and for different speeds. The suggestions simplify Haerle's method.

As in Haerle's solution, the change of stress at a change of section is assumed inversely proportional to the thickness. Driessen proposes to find the new S and D curves for the new section as follows: ΔR for the new section $= R - R' = R(1 - t/t')$. Haerle shows that $\Delta S = 1.3\,\Delta R$, and $\Delta T = -0.7\,\Delta R$. For the first section $S - D = 2R$ and for the second section $S' - D' = 2R'$. The change is $(S' - D') - (S - D) = 2\,(R' - R) = 2\Delta R$. Hence $\Delta S = (1.3/2)\Delta R$ and $-\Delta D = (0.7/2)\Delta R$. If $(S - D)$ and $(S' - D')$ are measured in inches on the chart, the distances ΔS and $-\Delta D$ in inches readily can be found, and the new points for starting the S and D curves for the changed sections easily are located on the chart and the quantities in lb. per sq. in. can be read from the diagram.

In Haerle's method the radial and tangential stress at one diameter, usually the hub were assumed, and the rim stresses determined. If the radial rim stress did not fit actual conditions, the disc was recomputed for other assumed tangential stresses at the hub until the desired radial rim stress results from the calculations. Driessen shows that if the tangential stress at the hub, T_h, is plotted against the radial stress at the rim, R_r, it follows a straight line. Hence when two points in this relationship are found, a straight line can be drawn through these points and any other T_h can be determined from this line for the actual R_r that prevails. This greatly shortens the computations.

Driessen points out that with large shrink fits, the stresses no longer are proportional to the squares of the speeds. He then outlines a method for computing and recording the limiting stresses of any disc, so that if an earlier design of disc is considered for a new turbine, reference to these limiting stress values will determine its suitability.

Other methods for calculating disc stresses will be found in: Stodola, Steam and Gas Turbines; Martin, Steam Turbines; Goudie; Steam Turbines; Kearton, Steam Turbines. For calculation

Table 3.—Stresses and Dimensions of Turbine Disc, Fig. 28

U	t	t'	$1-\frac{t}{t'}$	R*	ΔR	ΔS	ΔD	T Actual	R Actual
615	......	2.2	....	1,300	0	0	0	9,000	1,300
600	2.2	3.26	0.325	2,120	690	900	− 484	9,400	1,300
590	3.26	1.80	−0.81	1,960	− 1580	− 2060	1150	9,800	2,500
580	1.80	0.68	−1.65	4,100	− 6760	− 8800	4750	10,900	6,800
570	0.68	0.47	−0.44	11,450	− 5030	− 6580	3550	13,600	16,300
560	0.47	0.55	0.145	17,050	2480	3220	− 1740	14,100	16,600
500	0.55	0.73	0.246	18,250	4500	5850	− 3150	13,500	16,200
420	0.73	0.92	0.206	18,170	3740	4860	− 2620	13,100	16,200
340	0.92	1.18	0.220	18,440	4060	5300	− 2850	11,700	16,000
300	1.18	1.90	0.380	16,280	6200	8060	− 4350	10,000	13,600
260	1.90	3.20	0.406	11,500	4670	6100	− 3270	8,300	8,800
220	3.20	5.11	0.375	7,800	2920	3800	− 2050	7,700	5,900
185	5.11	6.30	0.189	5,150	970	126	− 680	8,300	4,200
82	6.30	6.30	0.0	0	0	0	0	14,800	0

* Values at inner diameter of steps.

of discs with conical profile see Rotating Discs of Conical Profile (*Engg.*, Jan. 5, 26, 1923); B. Hodgkinson, Rotating Disc of Conical Profile (*Engg.*, Aug. 31, 1923); G. A. Arrowsmith, The Design of Rotating Discs (*Engg.*, Oct. 5, 1923).

SHRINK FITS FOR DISCS.—Experience has shown that discs tend to work loose on shafts if shrinkage allowance in the bore is insufficient. Tests covering the action of stress on disc bores are given in Lasche and Kieser's Materials and Design of Turbo-alternator Plant. Discs frequently are secured to the shaft by a conical bush. When a keyway is used on a disc, the key should serve only as a safety member, and exert neither radial nor lateral pressure. Sometimes two keyways are cut at opposite ends of a diameter to equalize stresses and to maintain balance.

Discs often are bored with an allowance for pressing on the shaft of 0.0015 in. per inch of bore diameter, requiring 1 $^1/_4$ to 2 tons pressure per inch of shaft diameter to force each disc in place. Such press fit compensates for the increase in bore of the hub of the disc when under stress, and prevents it from creeping along the shaft.

DISC AND BLADE VIBRATION.—(See Campbell, Protection of Steam Turbine Disc Wheels, *Trans.* A.S.M.E., xlvi, 1924.) Failures of some of the discs and blades of early turbines led to intensive studies of vibration. These developed the fact that stationary discs and blades vibrated harmonically in an even number of segments, between which were radial lines of quiet, called nodes. The nodes appeared at definite frequencies, as 4, 6, 8, 10, 12 or even higher numbers of nodes. Between these given frequencies the wheel was comparatively quiet. The higher the number of nodes, the higher the frequency of vibration and the less easily is vibration excited. Both disc wheel and the blades vibrate as a continuous disc and must be considered as a unit in this type of vibration. The frequency of a given type of vibration is determined by two factors: *a*, the stiffness, and *b*, the mass of the vibrating body. The stiffer the body, the faster it vibrates, and the more massive it is, the slower will it vibrate. Centrifugal force exerts a stiffening effect on a disc and increases the frequency at which the nodes appear.

The combined frequency of a particle, f_r due to the combined effects of stiffness and centrifugal force is $f_r = \sqrt{f_s^2 + B N_s^2}$, where f_s = natural frequency of a particle of mass m, with an elastic support of such stiffness that a force R_s is required to produce unit deformation; $f_s = (1/2\pi)\sqrt{R_s/m}$; N_s = rev. per sec.; B = a speed coefficient which varies with the design of the wheel and the type of vibration. B has a low value when vibrating sectors extend well into the wheel, and higher values as the number of nodes increases. B varies usually from 2 to 3.

The critical speed $N_s = f_s/\sqrt{(n/2)^2 - B}$, where n = number of nodes. Minor resonant speeds occur, the equation for the first of which is $N_{S1} = f_S/\sqrt{(1 + n/2)^2 - B}$. See Campbell's paper for other values.

When discs are revolved, their circumferences develop, at certain speeds, a form of wave motion which travels around the wheel circumference in the opposite direction to rotation. This wave motion has an even number of nodes which move around the wheel with the wave. Every part of the wheel rim thus vibrates over a period of time during each revolution. If the number of nodes is the same, the frequency of vibration of every particle along the edge of a disc wheel is the same either for standing vibration or for traveling waves. The speed of the traveling wave per second equals the number of complete vibrations of the corresponding standing wave per second multiplied by the length of a complete wave. For a traveling wave all particles vibrate through the same amplitude but their time phases vary successively along the wheel edge.

When these backward traveling waves have the same speed backward that the wheel has forward, a standing wave results. This condition has been most conducive to disc and blade failure. The speeds at which the standing wave forms, are called the **wheel critical speeds.** Wave trains in discs may be started by the application of a small extra force at a given point due to uneven nozzle dimensions, thick partitions, or other lack of symmetry. These waves persist after formation if the speed is suitable for the wave.

The effects of vibration are: 1. The wheel may burst; 2. The wheel may rub; 3. Blades may fail from fatigue. Mathematics have been developed to predict disc stresses, and discs are made heavier than formerly, but one can only be sure of a wheel after a direct test. For this purpose, machines have been developed in which to test discs for vibration under working temperatures and speeds. Safe limits between the operating speed and the critical speed are, for 4 nodes, 15% above or below the operating speed, and for 6 nodes, 10% above or below.

When disc wheel critical speeds fall within these limits, the wheel is *tuned*, that is, metal is removed either from the disc itself or from the blades until the critical speed has been shifted beyond the specified limits. Discs are smooth finished throughout so that

no tool marks may form starting points for fatigue cracks. All sharp edges are rounded off for the same reason.

Tangential vibration may be set up in the blades themselves at certain frequencies, depending upon blade design. Long slender blades must be so designed and tuned that the critical speed falls outside the limits above noted. Blades can be tuned by affecting their stiffness by: 1. Soldering or welding the ends to a shroud; 2. Adding an intermediate stiffening member between tip and base such as a lashing wire; 3. Decreasing the mass of the buckets. The position of the lashing wire has a very great effect on the resistance of blades to vibration.

The frequency of vibration of a reed is $f = tC/l^2$, where f = frequency, cycles per sec.; l = length, in.; t = thickness, in.; C = constant of proportionality. For turbine blades a factor S represents the scale of equivalent thickness. Comparing two blades $S_2/S_1 = f_2 l_2^2/f_1 l_1^2$. Thus a test blade is 14 in. long, with a frequency of 62. The scale of thickness for a similar blade 24 in. long with a frequency of 37 is $S_2/S_1 = 37 \times 24^2/52 \times 14^2 = 1.755$. That is, the required thickness of the new blade is 1.755 that of the one tested.

Blades that may be subject to resonant vibration which leads to fatigue failure, are heavier and somewhat wider than earlier forms and have both inlet and outlet edges rounded. Where possible, long blades are made sufficiently stiff without the use of lashing wires.

Further details and the mathematics of the subject will be found in Campbell's paper above mentioned; Tangential Vibration of Steam Turbine Buckets, W. Campbell and W. C. Heckman, *Trans.* A.S.M.E., xlvii, p. 654, 1925; The Experimental Investigation of Vibrations in Turbine Wheels and Blades, B. Pochobradsky, L. B. W. Jolley and J. S. Thompson, *Engg.*, Oct. 30, 1931; Vibration of Steam Turbine Discs, J. von Freudenrich, *Engg.*, Jan. 2, 1925; Axial Vibration of Rotating Steam Turbine Disc Wheels, R. H. Collingham, *The Engineer*, April 3, 10, 1931; Contribution to the Calculation of Blade Vibration in Steam Turbines, W. Peter, Brown Boveri *Review*, May–July, 1934; Influence of Lashing and Centrifugal Force on Turbine Blade Stresses, R. P. Kroon, *Trans.* A.S.M.E., APM–56–2, Mar. 1934; Design and Calculation of Steam Turbine Disc Wheels, I. Malkin, *Trans.* A.S.M.E., APM–56–8, Aug. 1934.

Compound Vibration.—While the critical speed can be calculated by the preceding formula, and the expected vibration will occur at that speed, it often is noted in practice that the turbine goes through several critical frequencies before reaching the calculated critical speed, due to the elastic scale of the foundations and the masses of the turbine parts. The foundation is subject to the periodic forces induced by the turbine speed, to the influence of the superimposed mass, and to the elasticity of the supports. If deflection in the supports is large, liability to vibration is increased. On the other hand, the steel columns supporting the unit may be too stiff. The completed unit, therefore, frequently is studied by a vibrometer, and adjustment made both in machine balance and in foundations to secure the desired quiet operation at normal speed.

DISC LOSSES.—Formulas for disc losses give conflicting values. Research has not definitely (1935) fixed these data. Several current formulas follow. Goudie (Steam Turbines, p. 529) gives the following formulas for disc and blade ventilation losses. Total disc friction in horsepower, $\text{Hp.}_d = 0.0607\, D_1^2\, (u_1/100)^3/v$. Total blade ventilation loss in horsepower for unenclosed idle blading, $\text{Hp.}_b = 0.458\, D_m\, (u_m/100)^3 \times (Bsl^{1.5}/v)$, where D_1 = diam. of disc, ft.; u_1 = peripheral speed of disc, ft. per sec.; v = specific volume of steam, cu. ft. per lb.; D_m = mean diameter of blade ring, ft.; u_m = peripheral speed of blades, ft. per sec.; l = blade length, in.; s = fraction of mean circumference not receiving steam from nozzles; B = a correction factor for the number of blade rows; $B = 1$ for single row, = 1.23 for 2-row, and 1.8 for 3-row wheels.

The detrimental effect of partial admission is diminished by placing a U-shaped ring in the casing which practically encloses the section of idle blades.

For enclosed blading, the loss, Hp._b is only 0.25 to 0.5 of the value as found above.

Stodola (Steam Turbines, p. 201) presents a later formula by Forner which gives lower losses than the above. This formula applies to an unenclosed wheel in an open casing with no steam admission, and for blade lengths from 0.4 to 4 in.

$$\text{Hp.}_r = 10^{-10} \beta D_m^4\, n^3\, l\, /v$$

where Hp._r = total friction loss in horsepower, other notation as in previous formula, except that $\beta = 0.83$ for discs with a single row, = 0.97 for 2-row, and = 1.32 for 3-row blades. With enclosed wheels between diaphragms of a multi-stage turbine, the loss is only $^1/_2$ to $^1/_4$ of the above amounts, being smaller the larger the disc diameter. Zietemann (Dampfturbinen, pp. 77–78) presents data from Brown Boveri and Co. that indicates still lower losses than either of the above formulas.

Stodola (The Efficiency of Reaction Blading, *Engg.*, Oct. 2, 1925) gives the following empirical formula for disc loss alone as $N_{kw} = 10^{-6} \times 0.256 \times d^{4.2}\, (n/1000)^{2.8}/v$, where N_{kw} = loss in kilowatts; d = diameter, in.; n = r.p.m.; v = specific volume of steam,

cu. ft. per lb. He states that this has been established by Brown Boveri and Co. and gives more accurate results than his earlier formulas.

F. Ribary, Brown Boveri *Review*, July, 1934, gives the following formula for the friction losses of an unenclosed disc:

where N_{kw} = friction loss, kw.; R_e = Reynolds number = Du/v; D = diameter, ft.; u = peripheral velocity, ft. per sec.; v = kinematic viscosity of the surrounding steam; γ = specific weight of surrounding steam, lb. per cu. ft.; g = gravity (32.2 at sea level).

Later designs embody elements for enclosure of idle blades and for reduction of clearances at side of discs to reduce pumping action in the surrounding steam. Even bolt heads are shielded.

Hodgkinson (Steam Turbines and Condensing Equipment, *Elec. Jour.*, Nov., 1924) calls attention to the displacement loss, which is the energy required in the case of idle blades in impulse turbines, to sweep out the inert steam from the blade passages when they come into the active arc of steam admission. This displacement loss for each nozzle group is $w_d = 8.5\, hbV_w/v$, where w_d = displacement loss, lb. of steam per hour; h = mean blade height, in.; b = blade width, in.; V_w = mean blade velocity, ft. per sec.; v = specific volume, cu. ft. per lb., corresponding to steam condition in impulse wheel chamber. Individual or group control of nozzles does not produce the gain in efficiency sometimes expected, on account of this displacement loss. The greater the arc occupied by the nozzles for a given steam flow, the smaller the blades will be, resulting in lower displacement and windage losses.

SHAFT DIAMETERS for impulse turbines are a compromise fixed by two considerations. To keep leakage low at the diaphragm labyrinths the diameter should be small. This results in a shaft lacking stiffness. Thermodynamic considerations thus require a small shaft, operating considerations a large shaft. Shaft sizes tend to larger diameters, favoring safe operation at some sacrifice of efficiency. With spherically-seated bearings, the maximum deflection of shafts carrying discs varies from 0.005 to 0.030 in. Shafts with solid bearings are stiffer, and have deflections only 50% of the above.

Kearton (Steam Turbines, p. 249) suggests the following formula for the first approximation of shaft diameter, $d = l\sqrt{ND/K}$, where d = mean diam. of middle portion of spindle, in.; l = span between bearing centers, in.; N = r.p.m.; D = average mean diam. of discs, in.; K = a constant; for land turbines K = 6,000,000 or above.

Rotors of Parsons turbines sometimes consist of solid shafts throughout, of solid shafts in the high-pressure section with rings or discs on the low-pressure end, of a hollow cylinder fastened rigidly to the spindle ends, or of other modified constructions. In Parsons turbines, with small radial clearances of blades a stiff shaft is required. With spherically-seated, self-adjusting bearings, the maximum deflection varies from 0.001 to 0.005 in. Hence, larger and stiffer shafts are used than with impulse turbines. When high mean blade speeds are necessary in the low-pressure section of a Parsons turbine, in order to handle economically the large volume of exhaust steam, discs of hyperbolic or conical cross-section, either integral with the spindle end or placed as rings on the shaft, are used. These discs carry from 1 to 7 rows of blading on their outer rims.

A bore hole concentric with the finished rotor, with a maximum eccentricity of about 0.02 in., is drilled through each large shaft for complete periscopic inspection of its interior structure. In heat treating, this hole assists attaining uniform heating and in relieving

In one design, solid discs are welded together at their peripheries, annealed to eliminate welding stresses, no through shaft being provided. The advantages claimed are: The stiff rotor runs far below its critical speed; the distribution of material is excellent; the spindle heats quickly and uniformly; the wheels cannot work loose; no keys or keyway are needed; the stresses are small; and the weight is low.

STRESSES IN DRUM ROTORS.—J. M. Newton in High Speed Steam Turbine Rotor Design and Construction, Junior Inst. of Engrs., 1910, gives methods of calculating stresses in drum rotors.

Goudie (Steam Turbines, p. 379) suggests the following method of calculating the stresses in drum rotors: Let f_t = total tangential stress at a given section in the drum, lb. per sq. in.; f_{t_1} = stress in the drum when there are no blades, lb. per sq. in.; f_{t_b} = additional stress due to blade load, lb. per sq. in.

Then $$f_t = f_{t_1} + f_{t_b} = \frac{\omega^2}{g}$$

where ω = angular velocity, radians per sec.; r = mean radius of the drum, in.; g = 32.2;

w = weight, lb. per cu. in., of the drum steel; w_b = weight of blading per ring, lb. per in. of circumference at radius r; P_a = axial pitch of blade rows, in.; d = equivalent thickness of rim, in., allowing for the blade groove; and

$$\left(\frac{f_t}{\omega^2 r^2} - \frac{w}{g}\right.$$

Let d_1 = actual thickness of drum, in.; d_2 = depth of groove, in.; b = width of groove,

Then $$P_a d = P_a d_1 - bd_2 \quad \text{and} \quad d_1 = d + bd_2/P_a.$$

The weight of the blading for Parsons turbines can be found by the following equation suggested by J. M. Newton:

where b = axial width of Parsons blade row, in.; l = length of blades from base of groove to tip, in.; α = a coefficient varying from 0.15 to 0.3; β = a coefficient varying from 0.06 to 0.11. This approximate formula, when the constants are known for the particular form of blading, saves the calculation of the weights of the blades themselves and also of the packing pieces.

Considerable judgment must be used in applying these formulas for drums, as several arbitrarily variable factors are involved. There also is considerable leeway in the choice of permissible stress f_t, which varies from low values up to about 18,000 lb. per sq. in. for mild steel, and 25,000 lb. per sq. in. for nickel-steel forgings as a maximum. Furthermore, the drum must be made of such thickness that no distortion is possible when caulking the blades. The minimum thickness at the high-pressure end of hollow drum spindles should not be less than 1 1/2 in. Rotors with blade ring construction have the additional stiffening effect of the radial pull in the web just as in a disc.

STEEL FOR DRUM ROTORS.—Certain parts of rotors may be subject to low wheel speeds and low temperatures. Low-carbon steel forgings, having the following properties, may be used for such purposes: Elastic limit, 22,000 to 25,000 lb. per sq. in.; ultimate strength, 65,000–70,000 lb. per sq. in.; elongation, 15 to 18%; reduction of area, 20 to 25%. This steel must be of the best quality with 0.50 to 0.60% Mn, 0.25% C, 0.25% Si, and not over 0.025% of S or P. The limiting stress at 20% overspeed should not exceed 20,000 lb. per sq. in., which gives 13,900 lb. per sq. in. stress at normal speed.

On account of the sensitivity of alloy steel to the effects of bad judgment in heat treatment, some builders prefer to use carbon steels for all rotor parts where the stresses will permit. One such steel contains 0.48% C, and has 85,000 lb. per sq. in. ultimate strength, and 45,000 lb. per sq. in. elastic limit.

Various alloy steels are used for rotor parts subject to high temperatures and to high stresses. These parts are often of 3 to 5% nickel-steel, and have the following properties: Elastic limit, 40,000 to 60,000 lb. per sq. in.; ultimate strength, 80,000 to 95,000 lb. per sq. in.; elongation, 20%; reduction of area, 35 to 40%. Such forgings are given an elaborate heat treatment to ensure relief from internal stress and to provide the desired properties. Nickel-molybdenum steel with 2.5% Ni and 0.2% Mo, 90,000 lb. per sq. in. ultimate strength, and 50,000 lb. per sq. in. elastic limit, and chrome-molybdenum steel with 0.6% Cr, 0.25% Mo, 100,000 lb. per sq. in. ultimate strength, and 60,000 lb. per sq. in. elastic limit, also are used for highly stressed drum rotor parts.

BLADE SPEEDS ON PARSONS ROTORS.—In order to pass the large volumes of steam in low-pressure stages, tip speeds of 1257 ft. per sec. and blade lengths of 40 in. have been used, resulting in a mean blade speed of about 920 ft. per sec. Early Parsons rotors consisted of a series of barrel-shaped cylinders of increasing diameters. Losses are reduced by making these rotors of conical shape with no abrupt changes in diameter. This conserves carry-over and reduces eddy losses. Small diameters are necessary at the high-pressure end, with blade speeds of 200 to 300 ft. per sec. in order to have as long blades as possible with the small volumes of high-pressure steam, and thus reduce blade tip leakage.

In the earlier drum arrangement a definite proportion of the total work was assigned to each drum. In modern (1935) designs with conical casings, each row of blades develops the work for which it is designed and which depends upon the ratio ρ of wheel to steam speed. In certain turbines, the small-diameter portion of the turbine with its many rows of blades has been replaced by a Curtis wheel, which gives a shorter and more compact turbine with lower steam pressures in the casing.

DYNAMIC BALANCE.—All rotors must be in dynamic balance to avoid excessive vibration at high speed. Lack of balance may be due to non-homogeneous disc or drum material, to slight eccentricity of the rotating masses, or to errors in workmanship. Static

balance first is obtained by mounting the shaft on carefully leveled knife edges and applying counterweights until it remains at rest in any position. Disc wheels are balanced similarly by mounting them on true arbors. For theory and description of unit, see A Statical Balancing Machine, E. H. Lamb, *Engg.*, May 27, 1932.

Static balance of the assembled shaft and its disc or drum is no assurance that it will be in good dynamic balance, as two heavy masses in the completed rotor may be so placed as to form a static couple. This will cause severe vibration at high speed, due to the dynamically unbalanced centrifugal forces resulting from these masses. They may be balanced by providing additional masses to set up an equal and opposite couple.

All manufacturing plants now have dynamic balancing machines in their plants where rotors, partly or completely assembled, can be quickly and accurately balanced. When it is necessary to balance a rotor in the field, it is run up to the operating speed, the high spot is marked and counterweights added by a cut-and-try method until balance is obtained. This operation is tedious and difficult, and final balance depends upon the skill and experience of the one who adjusts the balance weights. See section on Vibration.

CRITICAL SPEED.—As every horizontal rotor deflects somewhat under its own weight, it is never possible to have the center of mass and the true center line of the shaft coincide. As the rotor speeds up this eccentricity of mass results in an increasing centrifugal force tending to bend the shaft. Any slight unbalancing in the mass aggravates this condition. At a certain speed this unbalanced centrifugal force neutralizes the elasticity of the shaft which resists deflection. The shaft deflection progressively increases, and if unrestrained, failure would result. In an actual turbine the shaft will rub on the glands and blading before this happens, causing considerable damage. The speed causing indefinitely large deflection of the rotor for a small initial eccentricity is the *critical speed* of the shaft. If the speed is increased above the critical speed, the shaft begins to straighten and tends to revolve about its true center of mass. Operation may be very smooth under these conditions, although other critical speeds may be encountered at still higher speeds.

The calculation of critical speed is difficult, except in the simplest forms of shafts and wheels. Methods of finding critical speeds will be found in Stodola's Die Dampfturbinen; Morley's Strength of Materials; Goudie's Steam Turbines; Kearton's Steam Turbines; and in Calculation of Vibration and Whirling Speeds, by Professor A. Morley (*Engg.*, June 30, 1909). Some engineers use the following formula for a rough approximation:

$$\text{Critical speed, r.p.m.} = 188/\sqrt{y},$$

where y = maximum shaft deflection, in. The deflection y depends on bearing and coupling conditions being less with solid bearings and couplings than with flexible ones.

Some small turbines run at speeds above their critical speed. Hence, in starting they pass through this speed. To avoid serious deflections they are brought to speed quickly, thus passing through the critical speed so fast that no extreme vibration can occur. Running speeds in all machines are chosen about 30% above or below critical speed. If the critical speed figures out only slightly above running speed, the shaft is made larger and stiffer and the critical speed thereby raised.

Rotor shafts often are bent if a rub occurs from any cause. This is due to local overheating on the side that rubs. The resulting expansion merely makes the rub worse, and finally the overheating causes a permanent set in the shaft. Danger from rubs of such character is being removed by the use of thin metal labyrinth packings in diaphragm glands and the use of thin-tipped blades in Parsons blading.

7. TURBINE DETAILS

Thrust Bearings

There is little end thrust in impulse turbines. In small units, thrust bearings frequently consist of collars placed on the shaft on each side of one of the bearings.

Large turbines often are fitted with a marine type of thrust bearing comprising a series of collars turned on a sleeve keyed to the main shaft. These fit between stationary rings in a sleeve fastened to the pedestal. Each half of the thrust sleeve on Parsons turbines can be adjusted in opposite direction, so that the spindle can be set relatively to the stationary blades in both directions. Allowable pressures on these collar bearings should not exceed 50 to 60 lb. per sq. in. of bearing face. In certain impulse turbines, where the thrust is small, the thrust bearing rests in a spherical seat to provide flexibility in alignment.

Large impulse turbines are built with a moderately large single-thrust collar firmly fastened on the shaft. This revolves between two babbitt-faced bearing rings fitted with

liners in the bored casing and made in halves. The rings have large radial oil grooves with well-rounded edges to insure ample lubricating oil, which is furnished at the bore of the rings. Slight end play of 0.005 to 0.010 in. is permitted for the shaft. Emergency or squealer rings sometimes are fitted to warn the engineer when the thrust surfaces have worn a certain amount, usually 50% of the clearance of the first-stage wheel. Ball bearings are used as thrust bearings on some small turbines.

Thrust collars and bearings are provided on some generator rotors which are connected to the turbine by a flexible coupling. These thrust bearings always should be provided if the generator is to operate uncoupled as a synchronous condenser.

KINGSBURY THRUST BEARINGS.—Reynolds' theory of lubrication has been applied to thrust bearings by Albert Kingsbury in America, and A. G. M. Michell in England, and their names designate forms of bearings widely used by turbine builders.

See Vol. 3 for detailed data upon Kingsbury thrust bearings. The Kingsbury bearing for steam turbines usually is designed to carry a load of 250 to 425 lb. per sq. in. to allow for dirty or worn oil and to provide a wide margin of safety. Under ideal conditions it could carry 3000 lb. per sq. in. In the fixed type of thrust bearing, liners are placed between the thrust bearing cage and the pedestal to fix the spindle position in the turbine. The bearing sometimes is mounted in a cage which can be adjusted axially to properly locate the moving blades relative to the casing.

The mean speed on Kingsbury thrust blocks may be as high as 215 ft. per sec. However, to keep the diameter small, lower speeds generally are used. Ample quantities of oil at low velocities must be supplied.

Couplings

Solid couplings are used on some turbines. These stiffen the shafts of both turbine and generator but require careful alignment. Spacer rings are provided in solid couplings to permit the spindle to be lifted out for inspection and to allow the generator to operate as a synchronous condenser. The claw type, which has a certain amount of flexibility, consists of two halves, each in two parts. The inner sleeve, keyed to the shaft, has jaws on the outer flange. The outer sleeve has a plain flange to bolt to the other outer sleeve on the other shaft. A set of claws, cut on the other end of the outer sleeve, fits into the jaws of the fixed sleeve. The two halves of the coupling are held together by fitted bolts. Lubrication of the bearing surfaces of the jaws is usually insured by an oil catcher and holes through the jaws to the bearing faces. Hardened steel plates are used on the jaw wearing surfaces.

The Westinghouse coupling has the two inner and outer sleeves, but instead of claws, the machine end of the outer sleeve has a flange carrying hardened steel pins projecting axially into bronze-lined holes in the flange of the inner sleeve. This forms a pin drive coupling instead of the claw form.

Both claw and pin couplings are subject to wear that results from slight mis-alignment or unbalance in the coupling and which cannot be entirely overcome by lubrication.

A second type of pin-type coupling used on several turbines consists of two standard flange couplings, each fastened to its respective shaft. Steel pins, firmly fastened to one half-coupling, project into holes in the other half-coupling where it carries a rubber bushing, brass-lined, bearing against the inner face of the bored-out hole.

The Fast flexible coupling comprises two hubs, each keyed to its respective shaft. Each hub has external spur teeth cut on it, at the maximum distance possible from the shaft end of the hub. A sleeve surrounding these hubs is flanged and split vertically at its center for disconnecting the two shafts. The two halves are bolted together through the flanges. Each half of the sleeve has internal spur teeth cut on its bore at its outside end, which engage the external teeth of the hub. The sleeve is carried at each end by an oil-tight supporting ring. The error in alignment of the two shafts can be about 10 times the clearance between the external and internal teeth, which are in an oil bath when in operation. The Fast coupling has proved very satisfactory where expansion from heat, as on turbo-drives for auxiliary equipment, makes it difficult to maintain correct alignment. It is now used with certain modifications on many small and large turbines.

Various plans for securing true alignment include leveling pads on the bed-plates, squaring and leveling coupling faces, stretching piano wire over the centers and checking up, and the use of surface gages. Allowance must be made in non-condensing units for expansion above the bed-plate on heating up. On some turbines the outer end of the machine is raised slightly to make the coupling faces come square after allowing for the natural deflection of the heavy rotating field.

Turbines may be thrown out of alignment by pipe strains. Piping must have bends so arranged that no strains are transmitted to the turbine casing. All base plates have a certain amount of deformation when stress is applied. Hence reliance cannot be placed on these to take all pipe strains without affecting the turbine. Spindle jacking gear,

operated by oil, is incorporated in the coupling design of large units for turning the spindle during erection or overhaul.

Dummy Pistons

TYPES OF DUMMY PISTONS.—Balance, or dummy, pistons are used on Parsons turbines to equalize the end thrust towards the exhaust due to difference of pressure between the inlet and outlet of each row of moving blades and also to the unbalanced pressure on the annular surfaces when the drum is stepped-up in size. In some turbines, there is a balance piston for each diameter of blade drum on the spindle; three is the usual number. Pipes fitted outside of the casing connect with the corresponding blade section, and provide the equalizing pressure. Using three balance pistons increases the overall spindle length, and requires an irregular cylinder construction to care for the large low-pressure balance piston on large machines. Such cylinders may distort under sudden temperature changes, due to variation in load. In some turbines, the low-pressure balance piston is in the exhaust. The webs carrying the blade rings of the last drum have steam equalizing holes. Dummies on this piston have radial clearance only. On some designs, a single balanced piston of small diameter is used with high-pressure steam on one side and low-pressure steam or vacuum on the other. Any unbalanced thrust is taken by the Kingsbury bearing.

Balance pistons with axial clearance consist of a steel ring, pressed or keyed to the spindle, with a number of rectangular grooves or teeth cut in its outer face. The stationary dummy section is either integral with the casing, or consists of a cast-iron ring fastened to the casing. Dummy strips of brass, bronze, manganese-copper, nickel, cupro-nickel or Monel metal are caulked in grooves in this ring, and project into grooves on the balance piston, forming a labyrinth packing. Dummy strips are cut and fitted in sections approximately 1 in. long. Small clearances between the side of the groove and a projecting sharp edge of the dummy are adjusted by the thrust blocks. More recent forms (1935) of dummy packing have several axial knife-edge projections instead of one, thereby increasing the number of throttlings in the dummy piston. Experimental data are used as a basis for calculating losses from such dummies. Dummies must provide space between throttling points to act as expansion chambers, must dissipate heat readily should contact occur, and one material must wear away rapidly with little heat generation. With radial clearances, dummies frequently consist of plain radial strips alternately deep and shallow, sealing against alternately low and high lands on the spindle dummy piston. Radial dummies depend on throttling through the small radial clearance at the tips of the projecting teeth for reducing the leakage. Ljungstrom turbines have a dummy using a thin nickel-steel ribbon, which provides fine radial clearance. Sealing strips of nickel ribbon, chromium stainless steel and other alloys are used.

AREA OF DUMMY PISTONS.—Goudie (Steam Turbines) says that the dynamic thrust on the blades in an axial direction is usually less than 1% and never exceeds 2% of the total thrust in Parsons turbines, and may be neglected. For the annular area A_d of the balance piston for each cylinder of Parsons blading, he gives on p. 424, the formula

$$A_d = \frac{P_2 A_1}{P_1 - P_c} + \frac{\Sigma(P_i - P_o)a}{2(P_1 - P_c)}$$

where P_1 = pressure on front of dummy; P_2 = difference of pressure on any annular drum area A_1; A_1 = annular area of any step-up in drum at entrance; P_c = condenser or back pressure; $(P_i - P_o)$ = drop in pressure in a group of blades of constant mean diameter; a = annular area between drum and casing at a group of blades; pressures are lb. per sq. in., absolute; areas are in sq. in.

LEAKAGE OF STEAM THROUGH DUMMY PISTONS.—H. M. Martin (*Engg.*, Jan. 10, 1908, and Jan. 3, 1919) discusses the leakage of steam through dummy pistons and submits a formula which is claimed to check within 1% of the actual loss. From this the following equation is derived:

$$W = 0.4722A\sqrt{\quad} \times \frac{1}{N + \log_e r}$$

where W = steam leakage, lb. per second, through whole dummy; A = area available for flow of steam at any dummy constriction, sq. in.; P_1 = initial pressure, lb. per sq. in.; v_s = specific volume of steam at pressure P_1, cu. ft. per lb.; N = number of rings or constrictions in the dummy; r = ratio of initial to absolute final pressure over the dummy $= P_1/P_2$, where P_2 is the pressure on the rear side of the piston, lb. per sq. in., absolute.

Goudie (Steam Turbines, p. 512) reduces the results of several formulas for balance pistons and dummies to a series of curves and an alignment chart which makes the solution of the problem comparatively simple.

To determine the number of rings required at each piston, Morrow, in Steam Turbines, suggests the empirical equation,

$$N = (40P_1 - 2600W/A) \div \{540\,(W/A) - P_1\},$$

where N = number of dummies in piston; P_1 = steam pressure before piston, lb. per sq. in., absolute; W = permissible leakage of steam, lb. per sec.; A = annular area of clearance through dummy, sq. in.

Naylor (Steam Turbines, p. 99) gives for leakage through Ljungstrom disc packing

$$w = 0.4722 \cdot \qquad \frac{p_1}{v} \times$$

where A_1 = first ring leakage area, sq. in.; A_n = last ring leakage area, sq. in.; $\beta = A_1/A_n$. (It is assumed that the diameters of the labyrinth rings increase in arithmetical progression); w = steam leakage, lb. per sec.; p_1 = pressure before packing, lb. per sq. in., abs.; v_1 = specific volume at condition p_1, cu. ft. per lb.; N = number of dummy strips or throttlings; p_o = exhaust pressure, lb. per sq. in. abs., and $r = p_1/p_o$.

CLEARANCE OF DUMMIES.—Radial dummies must be used where the balance piston is distant from the thrust, and considerable expansion can occur. Side-contact dummies are ground to fit by revolving the spindle slowly and drawing up on the thrust until contact occurs. They are afterwards set, when thoroughly heated, by drawing up on the thrust, the spindle revolving very slowly, until first contact is heard by listening on the casing. The thrust then is moved over to obtain the desired running clearance, which varies from 0.004 in. on small Parsons turbines, up to 0.012 to 0.015 on large turbines. Balance pistons near the middle of the spindle require somewhat larger clearance. When finally set, all thrust blocks are locked after allowing sufficient end play for lubrication. This adjustment should be checked periodically to detect wear in the thrust collars due to clogged oil supply or dirty oil. Radial dummies are given a clearance of 0.010 to 0.025 in.

Labyrinth Glands

Glands must be provided in all turbines where the shaft leaves the casing. Impulse turbines also require glands where the diaphragms between stages encircle the shaft. The following glands are used: Carbon rings, dummy pistons, labyrinth teeth, water glands, water glands combined with some form of labyrinth.

Carbon Ring Glands comprise several carbon rings, each in its own compartment of a cast-iron or steel case. The several segments of the ring are pressed together by a garter spring or an arched flat spring. The rings usually are divided into three or four segments. Clearances on the shaft diameter are from line-to-line, to approximately 0.006 in., depending on the size of shaft.

Dummy Piston Glands are used in certain Parsons turbines.

LABYRINTH GLANDS for diaphragms consist of cut teeth projecting from the shaft toward a smooth stationary casing, from the casing toward the shaft, or from both with the teeth alternating similar to radial clearance dummy pistons. The ends of these teeth are knife-edged, and usually no clearance is allowed in design. The turbine is turned over slowly when first assembled, and a clearance of about 0.005 in. is worn on the sharp ends of the teeth. The gland loss varies approximately in the inverse ratio to the square root of the number of constrictions. The object of these constrictions is to throttle the steam into a larger space where it forms eddies and thus restricts flow. Gland strips with "pine-tree," comb, and other forms of fingers are used. Where impulse blading is placed upon drums, labyrinth packing has been put upon the blade ends. Metal ribbon teeth also have been used in such glands.

The high-pressure labyrinth gland, where the spindle leaves the casing, can be made in two sections. The longer inner part seals against the internal pressure, the outer part against atmosphere. Steam is withdrawn from or supplied to the gland at this middle point. Steam above atmospheric pressure must be supplied at the intermediate point in the exhaust-end gland to seal it against air leakage, which would tend to destroy the vacuum. The high-pressure gland leakage either is used in this low-pressure gland, or led to an intermediate stage, continuing its expansion in the turbine. Provision usually is made to admit live steam to both low-pressure and high-pressure glands to seal them on starting. The clearance between the teeth of these glands and the shaft or casing is usually from 0.005 to 0.008 in. Materials used for labyrinth glands are babbitt, aluminum, brass, and bronze, and for high temperatures, stainless or other alloy steels.

Later designs of labyrinth glands are planned so that any rubbing causes the parts automatically to separate. Glands made of packing rings of anti-friction metal are used on some small turbines which hardly warrant the expense of a more elaborate type.

WATER GLANDS where the spindle leaves the casing, consist of a small impeller or paddle wheel, fastened to a long sleeve or hub on the shaft, which revolves in a gland casing. This gland is supplied with water under a pressure of 10 to 15 lb. per sq. in. The water is unable to leak along the shaft, as the action of the impeller holds it in a solid ring against the outer casing. On the other hand, air cannot leak into the turbine because of this solid ring of water under pressure much greater than atmosphere. Several forms of combined water and labyrinth glands can be steam sealed when the turbines are run at speeds too low to maintain the water-gland seal. The governor may regulate the supply of water or steam to the glands. Proper leak-off passages are provided for steam and water.

Water glands generally are used outside of the labyrinth glands on high-pressure ends of impulse turbines, to prevent steam leakage into the turbine room, which leads to sweating troubles in winter. On account of the high temperature at this gland, condensate must be circulated through it, discharging into the feed system.

Any water free from acids or scale-forming impurities may be used in the glands. Condensate from the hotwell pump generally is used for this purpose.

Power Required by Water Gland.—Guy and Jones (Metropolitan-Vickers Rateau Marine Turbine, *Engg.*, Feb. 9, 1923) state that experiments indicate that the power required by a water gland with the paddle completely immersed in water is $P = 6\ u^3D^2/10^6$, where P = horsepower required; u = peripheral velocity of paddle wheel, ft. per sec.; D = diameter of paddle wheel, ft. Under actual operating conditions the paddle is not fully immersed on both sides and tests indicate that the power required is about 50% of that given by the formula.

The water required by water-glands varies from 0.5% to 2% of the condensate, but little of this is lost, since at the low-pressure end the vapor from the gland enters the exhaust and is recovered in the condenser. At high-pressure glands, the water must circulate, and the heat it absorbs can be fully recovered. Water glands usually seal above $^1/_3$ to $^1/_2$ of normal speed.

Steam Required by Casing Glands can be estimated by considering the small clearances at the ends of the teeth as equivalent nozzle area, and applying the following nozzle formulas. Since the pressure drop in both high- and low-pressure shaft glands is less than critical pressure, Goudie (Steam Turbines, p. 497) gives

$$W = 0.3155A\sqrt{P_1/v_s}\ \text{lb. per sec. for superheated steam;}$$

$$W = 0.0173AP_1^{0.96875}\ \text{lb. per sec., for dry or slightly wet steam,}$$

where A = average annular clearance area, sq. in., over the teeth of the gland; P_1 = initial pressure, lb. per sq. in., absolute (at turbine side on high-pressure end, and from steam supply line at low-pressure end); v_s = specific volume of superheated steam in cu. ft. per lb. at pressure P_1.

Leakage of diaphragm glands depends on the number of stages. If few are used, the pressure difference across the gland will exceed the critical ratio and the above equations can be used. Leakage decreases toward the exhaust end. The pressure drop across the diaphragm is frequently less than the critical ratio. Goudie (Steam Turbines, pp. 523–4) suggests that this be calculated by first finding the allowable heat drop $(h_1 - h_2)$ from adiabatic expansion between these two pressures, assuming a nozzle efficiency of 85%, and calculating the resultant velocity, in ft. per sec., from the formula,

$$V = 223.7\sqrt{0.85(h_1 - h_2)}.$$

The leakage, in lb. per sec., is $w = AV/v$, where A = average clearance area through the gland, sq. ft.; v = specific volume of steam after expansion, cu. ft. A chart to solve problems in labyrinth packing is presented in *Engg.*, vol. 128, p. 65, 1929.

Bearings

Turbine bearings may be divided into two classes: Self-oiling, and forced lubrication. Self-oiling bearings are used only on small turbines and generally consist of babbitt-lined cast-iron shells, with oil supplied by oil rings revolved by the shaft. Bearings for large units and for reduction gears always have forced lubrication from the main oiling system.

Large turbines may have spherically-seated, self-aligning bearings. In Parsons turbines, the spherical seats take the form of four pads under which are placed steel adjusting shims of varying thickness. Clearances at the ends of the blades can be equalized by changing the shims and thus shifting the position of the shaft relative to the casing.

As impulse turbines do not require close clearances over the ends of the blades, their bearings have plain spherical seats. When light shafts are used in some forms of impulse turbines, it is desirable to decrease the deflection and increase the critical speed, by using solid parallel bearings. Such bearings also are used for reduction gearing where accurate alignment is essential.

Large bearings consist of cast-iron shells split in half horizontally and lined with white metal. They are relieved for 30° at the sides above and below the joint, except for $^3/_4$ in. at each end, by re-boring slightly oversize with a separating plate between the halves. The lower halves are scraped to fit the journal for an arc of 120° at the bottom. Bearings usually are bored 0.002 to 0.003 in. large per inch diameter of journal. With forced lubrication, oil usually is supplied both at the top and sides of the bearings. Oil throwers either are turned on the shaft, or attached to it at the outer end of the bearing, to prevent escape of oil. Oil guards are provided on the bearing cover for the same purpose. Provision is made for the escape of entrained air from the bearing pedestals.

The design of the spindle usually fixes the size of the bearing. The rubbing velocity of the journal should not exceed 150 ft. per sec. The bearing pressure is found by dividing the total load on the bearing by the product of its length and diameter. A safe limit of this pressure is 135 lb. per sq. in. The maximum limits reached have been 195 ft. per sec. velocity and 175 lb. per sq. in. pressure. The ratio of bearing length to diameter varies from 1 to 2.5. Bearings are made shorter than formerly, as this reduces the total length of the turbine. The design should be such as to reduce spattering and splashing of oil, which leads to oxidation troubles and acid formation.

Bearings on some small turbines are so located that they are heated by conduction through the casings and through the shaft. Such bearings may heat to 250° F. and a heavy oil is required. Usual bearing temperatures range from 125° F. to 150° F. with occasional cases where the oil leaves at 175°. See Hodgkinson, Journal Bearing Practice, *Proc.* Inst. Mech., Engrs., 1929.

Bearings depend for their proper functioning upon the supply of a thick wedge-shaped oil film on the side of the bearing where the shaft turns downward. This film spreads out fan-like and maintains a separation of the two metal surfaces. There is no true coefficient of friction, but the shearing action of the oil offers a resistance to motion which is the so-called coefficient of friction. Kraft (The Modern Steam Turbine) states that this factor is 0.008 for a bearing of good workmanship. The heat, B.t.u. per min., generated in a bearing, $H = (\pi dN\mu W)/(12 \times 778)$, where d = bearing diam., in.; N = r.p.m.; μ = so-called mean coefficient of friction between journal and bearing; W = total load on the bearing, lb.

After analyzing the data of Lasche, Kearton (Steam Turbines, p. 191) presents the following formula for the heat generated in the bearing, in B.t.u. per min.

$$H = (1.72\, ld^2N)/\{100\ (t - 32)\},$$

where l = length of bearing, in.; d = diameter of bearing, in.; N = r.p.m.; t = temperature of bearing, deg. F. This formula indicates that the friction loss decreases as operating temperature of the bearing increases, which is in accord with practical experience.

The Oil Required Per Bearing can be estimated by the following formula, also from Kearton (p. 192): $G = 0.00206ld^2N/\{s\delta(t - t_s)(t - 32)\}$, where G = U. S. gal. per min.; l = length of bearing, in.; d = diam., in.; s = specific heat of the lubricating oil (about 0.45); δ = specific gravity of the lubricating oil at the working temperature (about 0.86); t = maximum oil temperature deg. F.; t_s = oil supply temperature to the bearing. $(t - t_s)$ usually may be assumed to be 20° F., although a greater temperature rise is allowed on some turbines.

Hot Turbine Bearings may be caused by insufficient oil, due to plugged-up oil pipes or oil grooves, or by failure of the supply. Sometimes bearing clearances are insufficient to admit the proper amount of oil, particularly at the sides of the bearings. Heating also may be due to too heavy an oil, to emulsified oil, or to old used-up oil. The true cause of the heating should be found and corrected at once, as no chances should be taken with bearings.

8. REDUCTION GEARING

See Section on Machine Design for design of reduction gearing. Also Section on Marine Engineering for marine applications.

The Efficiency of Reduction Gears is difficult to determine by mechanical methods. A common method of calculating gearing efficiency is to measure the friction heat carried away by the lubricating oil and to allow for radiation from the gear casing. Carefully made laboratory tests by both input and output measurements, and by heat measurement on single-reduction gears, show practically the same losses. Large single-reduction gears have shown efficiencies of 98 to 99%. The efficiency of small single-reduction gear sets ranges from 96 to 98%. Double-reduction gearing has given efficiencies varying from 88 to 97% on test.

9. TURBINE LUBRICATION

Oiling Systems

Small turbines provided with ring-oiled bearings require merely the maintenance of a suitable supply of pure mineral oil, changed at frequent intervals, in the reservoir below the bearings. The outer surfaces of the pedestals dissipate the heat generated by friction.

Large turbines have a complete self-contained oiling system, which consists of fine wire screens (35 to 60 mesh) to remove coarse particles, an oil pump, an oil cooler, and suitable piping systems.

The Oil Pump is driven by gearing from the main turbine shaft, or if reduction gearing is used, sometimes from the slower speed shaft. The pump, usually of the rotary gear type, supplies oil on occasion at 100 lb. per sq. in. pressure when used in an oil relay. The volumetric efficiency of gear-type pumps ranges from 70 to 80% with hot oil. Oil pressures to the bearings range from 2 lb. to 5 lb. per sq. in., depending on the type of turbine.

An Auxiliary Turbine-driven Centrifugal Oil Pump is placed on medium size and large turbines to circulate oil through the bearings before starting the unit.

On large units, a second pump may furnish oil under the bearings at 600 to 1000 lb. per sq. in. pressure, to lift the bearings and form an oil film on starting the unit after a shut-down. On some units, this is a multi-plunger reciprocating pump with a plunger and piping for each bearing.

Oil Coolers are built in many forms with brass or copper cooling coils. These coolers should be readily accessible for cleaning if raw water is used. Frequently, condensate forms the cooling medium, in which case less cleaning is necessary. The cooler surface is proportioned for the desired oil temperatures and the available cooling water. Usually the oil is circulated through the tubes. According to Stodola, the heat transfer coefficient in oil coolers is low, varying from 10 to 20 B.t.u. per sq. ft. per hr. per deg. F. temperature difference. Coolers lower the oil temperature about 20 to 30° F. Oil should pass to the bearings at temperatures between 105° F. and 140° F. Temperatures leaving the bearings range from 130° to 160° F. It is frequently specified that sufficient oil cooler capacity shall be installed to keep the maximum temperature of the oil below 150° F. when using cooling water at the maximum temperature specified by the purchaser for summer conditions.

Oil Pump Capacity is fixed by the total amount of oil required by the bearings and thrust and the amount needed by governing and overspeed gear, together with a liberal additional amount to provide for pump slip, air vents, etc. Each manufacturer has developed his own standards for oil pump capacity.

The oil pump capacities furnished by one builder for 3600 r.p.m. units varies from 33 gal. per min. on 1000-kw. units to 75 gal. per min. on 6000-kw. turbines. On 1800-r.p.m. units the capacities range from 115 gal. per min. on 10,000-kw. units to 400 gal. per min. on 60,000-kw. turbines.

Oil Reservoirs vary in size with the different turbines and with the several manufacturers. A frequent requirement for units of 5000 kw. and over is that the capacity of the oil reservoir at the turbine shall be such that it will take not less than 10 min. to circulate a quantity of oil equal to the tank capacity. With smaller turbines the period for complete circulation shall not be less than 5 min. Special precautions must be taken to minimize the danger of fires in oil reservoirs. Emergency drain lines to reservoirs, CO_2 and other non-inflammable flooding equipment, and location of the reservoir in a fireproof room below the turbine, now are used.

OIL PIPING.—Seamless copper, with brazed flanges and cast iron or brass fittings, has been used for oil piping. As this is expensive, brass pipe, with screwed fittings, is used. Steel piping and steel tubing also have been used with welded nipples for branches and joints. This material should be well pickled before assembly to remove mill scale and rust. The threads are made tight against oil pressure by shellac or similar material. Only gate valves should be used on oil lines.

Following are recommendations (Station Piping, Edison Elect. Inst., Aug. 1933) for the

enclose sufficiently so that it is not used as a ladder or for hanging things upon. Design with good supports and as nearly free from stress as possible.

Main oil lines should be of standard weight pipe, or seamless tubing of equal wall thickness, with extra heavy steel flanges and steel fittings; use welds as extensively as possible. Erect barriers around all oil joints when these must be used. Weld all threaded joints. Provide lock washers on all flange bolts. Male and female flange joints are preferred. Use metal-to-metal unions.

No connections should be made with less than 1/2-in. pipe size, with shut-off valves as close as

possible to the main pipe. Screen reservoir vents, and locate them at a distance from the steam lines. Reservoir covers should be self-closing.

Avoid gage glasses and use a mechanical indicator for oil level. Nipples for pressure gages must have restricted openings.

Keep oil system, particularly oil storage or reservoir, as remote as possible from high temperature steam connections.

Lubricating Oils

LUBRICATING OIL FOR TURBINES must be a properly refined, highly filtered, pure mineral oil, free from alkali or acid. While organic acids form from oxidation of unsaturated compounds or those containing sulphur, oxygen, or nitrogen, in the presence of water during use, that oil should be chosen which has the lowest organic acidity, not only when fresh but also after continued use. Organic acidity expressed as the amount of potassium hydroxide required to neutralize 1 gram of oil, should not exceed 0.07 mg.

Oil should be able to separate rapidly from water and not form emulsions; if these are formed they should be quickly separated on heating. Oil must be free from components of low boiling point, in order to maintain constant viscosity. It should have little tendency to break down and form sludge, when agitated at the actual operating temperature and mixed with air and water. The ideal lubricating oil should have maximum adhesion and minimum cohesion. Some operators of large turbines use an oil of as low as 100 sec. Saybolt at 100° F., claiming thereby to materially reduce friction losses. Color is of slight, if any, value in judging a lubricating oil, as the manufacturer can adjust color at will.

Foaming Oil may be caused by water in the system or may be due to air entering through leaks in the oil pump suction piping or strainer. Air also may be forced into bearing pedestals by the ventilating system of the generator. Foaming can be corrected by removing the water and preventing the entrance of the air.

Oils Which Tend to Decompose and age rapidly in service will deposit a hydrocarbon sludge in bearings, piping and oil cooler. Besides plugging passages in bearings, thrusts and couplings, it forms an insulation on the cooling coils. Higher oil temperatures result, causing still more rapid aging. Such oils should be removed and filtered, the system thoroughly cleaned and new, higher-grade oil purchased. Cotton waste must not be used to clean the oil system, as lint is left behind to cause trouble later by stopping up the oil passages. Good practice requires the oil to be drawn from turbines in operation every six weeks or two months and replaced with fresh filtered oil.

Reduction Gears have higher tooth pressures per sq. in. than the usual bearing pressures, and require a heavier oil than bearings. In some cases, these gears have their own oiling systems, using a heavy oil. When both gears and turbine bearings are on the same system, either a medium or a heavy oil may be used. Hot oil may be supplied to the turbine bearings and only the gear supply circulated through the oil cooler.

FILTRATION AND PURIFICATION OF OILS are necessary with moderate and large-size turbines. This usually is done in one of the following ways: 1. Continuous by-pass system. 2. Batch system. 3. Continuous by-pass batch system.

The continuous by-pass system continuously takes out a certain amount of oil for treatment. The batch system takes out a large amount at periods. The continuous by-pass batch is a combination of the other two methods. Separation of water and sludge may be secured by: *a*, Passing through a centrifugal separator or filter such as the DeLaval, Sharples, and others, which separate the substances on the basis of their specific gravities. This removes water and dirt, but not alkalies and acids nor soluble sludge. *b*, Bag filters, which remove insoluble sludge, but will not remove acids nor soluble sludge. *c*, Separating tanks, in which the oil is cooled and where water and sludge can settle out and be withdrawn. Much soluble sludge separates out when cooled. Usually oil reservoirs in the turbines are provided with piping which allows any water that gets into the oil to overflow automatically when it settles in the reservoir. See Lubrication (N.E.L.A., 1922–23–24).

Commercial turbine oils are available from a large number of refiners. Manufacturers in some cases offer to the purchaser a list of turbine oils that have proved satisfactory in the past, and from these a suitable oil generally can be selected. Non-inflammable fluids for lubricating systems are being tried (1935) to lessen danger from oil fires.

CHARACTERISTICS OF OIL.—Oil must be free from tarry, slimy or saponifiable matter, and also from soaps or other materials added to give body to the oil. It should contain no dirt, grit or other suspended matter. The specific gravity should be between 0.86 and 0.88 at 60° F.; flash point, not below 334° F.; fire test, 375° F. Acids can be detected by blue litmus, which turns red in the presence of acids. Animal and vegetable oils, used as adulterants, can be detected by a milk-white emulsion which forms when the oil is shaken with a strong solution of borax. After standing in a cool place the

clear mineral oil will be at the top, the borax solution at the bottom, and the emulsion in between.

Sludge in oil is highly undesirable. It plugs piping, coats oil cooler tubes and causes hot bearings by reducing the lubricating value of the oil. Sludge forms as a result of agitation of impure oil in the presence of water and air. To test the sludging properties of oils, a sludge accelerator has been developed which produces sludge in 1% of the time in actual service. For a description of this equipment and information on its use, see Lubrication (N.E.L.A., 1924). This report also contains detailed instructions for making flash, fire, viscosity, pour, total acidity, alkalinity, corrosion, carbonization, and demulsibility tests on lubricating oils.

VISCOSITY.—The viscosity is fixed by the oiling system of the turbine and the temperature conditions. Oils of the following Saybolt viscosities, all at 100° F., have given satisfaction:

Ring-oiled bearings with or without water jackets	150–200 sec.
Ring-oiled bearings, subjected to extreme radiated heat	300–500 sec.
Circulating systems, ordinary conditions	140–200 sec.
Circulating systems, with reduction gears	250–350 sec.

For specifications for steam turbine oils see Lubrication (N.E.L.A., 1922).

10. GOVERNORS

Turbines may be governed by throttling or by nozzle control. In *throttling governing*, the steam is throttled to some pressure lower than steam line pressure, at which lower pressure the heat available for work is sufficient to maintain the load. In *nozzle governing*, a series of valves are provided which admit steam under control of the governor to each valve in series as the load increases. The only valve under throttling action is the last one being opened; the others operate at full steam line pressure.

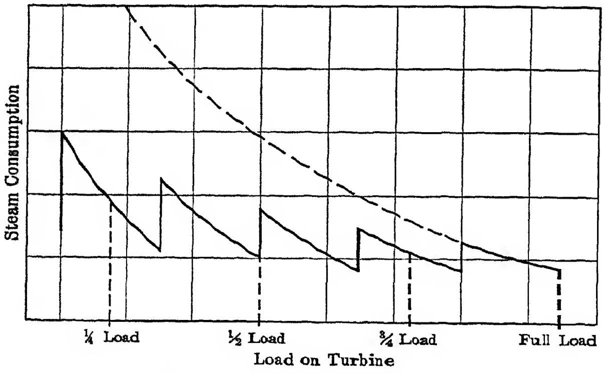


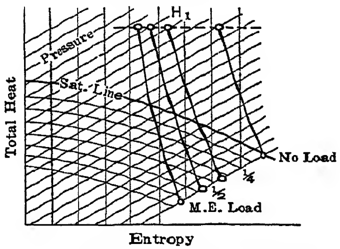
Fig. 29. Effect of Nozzle Governing on Steam Consumption

Theoretically, nozzle governing should be the more efficient, and in practice this is evident on small single-stage turbines. The water rate curve on such a nozzle-governed turbine is shown in Fig. 29. The solid line shows steam consumption with nozzles cut out consecutively, while the dotted line shows probable steam consumption with throttle governing. On large units, these theoretical gains are offset somewhat by idle blade losses, and in multi-stage units by the fact that the pressure drop in the first stage becomes excessive at light loads, leading to serious under-expansion and high nozzle and blade losses. Some builders have overcome this by providing two sets of nozzles, one for light load only, and the second for heavy loads, at which time the light load nozzles are cut out by suitable change-over devices in the governing system.

Throttle Governing is used on turbines having full peripheral admission, as on Parsons and some multi-stage impulse turbines. The water-rate curve with throttle governing can be made to simulate that of nozzle governing by selecting a *Most Efficient Load* at a relatively low percent of rating and by providing for the additional capacity by secondary,

tertiary, and even quaternary valves admitting steam to lower stages of the turbine.

Christie and Colburn (Turbines, Prime-movers Comm. Report, N.E.L.A., 1932) point out that the type of governing has a serious effect upon end-point conditions, and upon quality in the last stages under light load conditions. Figs. 30 and 31 show condition curves with the two types of governing.



30. Effect of Throttle Governing on Condition Curve

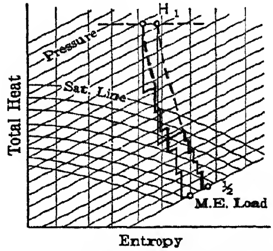


Fig. 31. Effect of Nozzle Governing on Condition Curve (1/2-load curve is displaced to right and should be superimposed on M.E. curve)

According to Podnossoff, on reduction of load with throttle governing, more operate in regions of increased superheat or quality, whereas with nozzle control, at the same reduced steam flow, more stages are transferred to lower temperatures and quality in the wet steam regions. This is evident from a study of Figs. 30 and 31. Such considerations are of commercial significance in the operation of high- and low-pressure turbines in series with a reheater between, and also in connection with blade erosion in single-cylinder turbines which operate for long periods at light loads.

Governors for Small Turbines.—The speed of most small turbines is controlled by a powerful centrifugal governor mounted directly on the end of the rotor shaft. Movements of the governor spindle are transmitted through a single lever to the regulating throttle valve which is usually a poppet valve. The governor weights are carried on levers by tool-steel knife edges in tool steel bearings requiring no lubrication. In some types, ball bearings are used instead of knife edges. One type has weights which roll on flat tempered steel springs with no pins, bushings, bearings or knife edges. Sometimes the lever connecting the governor spindle to the regulating valve also is provided with knife edges. These governors must be examined frequently to see that no parts are loose or worn. Some designs use a ball-and-socket, or similar connection, to the regulating valve lever. This connection must be well lubricated, and lost motion taken up when wear occurs. Fig. 32 is such a governor for a small Sturtevant turbine. Other companies use similar direct-connected governors. Speed regulation of these governors is about 2% above and below normal.

In several designs of small geared turbines the governor either is mounted on the slow-speed gear shaft or, in other designs, is geared from it. These governors, revolving at comparatively low speeds, are heavier and more powerful than when directly mounted on the high-speed shaft.

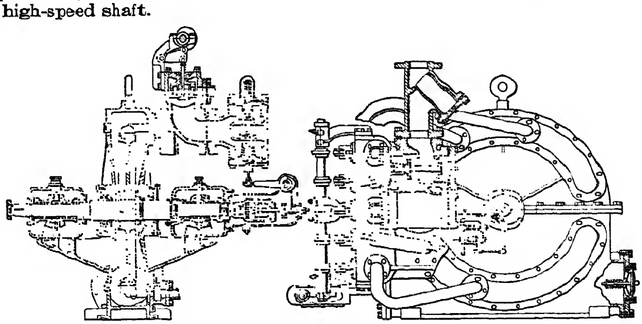


Fig. 32. Section of Small Sturtevant Turbine

Care must be taken that all levers always are free, and that stuffing-boxes on the regulating valves are not screwed too tight. Wear in collars of some types of governor must be taken up periodically if good regulation is to be maintained.

Governors for Large Condensing Turbines.—Condensing turbines may use governors of the centrifugal, fly-ball, or inertia types, geared from the turbine spindle. The oil pump often is placed at the lower end of the governor spindle and driven through the same gears. Regulation is obtained by some form of oil relay. Oil at from 25 to 100 lb. pressure is led from the oil supply system to a small relay valve, controlled by the governor. This compensated relay valve admits oil above or below a piston directly connected to the regulating valve, or controlling the admission valves through a proper mechanism. It often is provided with a device to open the secondary overload valve when the primary valve has lifted a certain distance. The relay on some machines oscillates, so that the valves are always in action.

On some turbines, the motion of the oil piston is transmitted by levers or by rack and pinion to a cam shaft which opens in succession a series of spring-loaded valves. Each valve admits steam to one or to a group of nozzles. This method of governing gives a comparatively flat water-rate curve over a considerable range of load. When reheat is used, intercepting valves and dump valves are usually oil operated, and may be actuated by the governor or the overspeed trip.

Pressure Regulating Governors are placed on many turbines driving pumps, etc. These are usually differential control valve gears, which admit sufficient steam to the turbine to maintain a given pressure or pressure differential at the pump discharge or at a specified point in the piping system. Pressure regulating systems must be arranged to work with the regular speed and overspeed control governors.

An Oil-operated Governor has been developed by Westinghouse which depends for its functioning on the variation in oil pressure delivered by an impeller pump mounted on the main shaft. The oil pressure varies as the square of the turbine speed. The discharge oil from the impeller is led to the governor housing, where it acts upon the operating piston of the main governor valve. Suitable relay valves are provided to insure only such movement as will tend to maintain the speed substantially constant. Remote control is provided in these relays for synchronizing and for varying the speed in operation, to give the desired frequency at a given load. Arrangements also permit tripping-out the turbine and of applying an overspeed test if desired.

To lessen the danger of fire due to breakage or leakage of oil control piping, this is now welded as much as possible. As a further precaution, a non-inflammable fluid such as chlorinated diphenyl is being introduced for high pressure turbine control systems.

Governors on Turbo-alternators have their regulation fitted to the service for which the unit is designed. In general, 2% variation between no load and full load is allowed for electric lighting service. Greater variation may be desirable where the load changes are large. Too close regulation is not necessary or desirable, as this may cause the load to surge between machines that are in parallel. Surging often can be reduced by increasing the speed variation of the governors. Turbines operating in parallel with hydro units usually do not require close regulation.

Since the introduction of electric clocks, constant frequency must be maintained on the power system of public utilities, regardless of load. This is obtained by hand or automatic regulation of frequency on the governor auxiliary spring, which also is used to synchronize the unit with other turbines.

Regulating Valves.—Balanced valves are used in nearly every case for regulating valves of the throttling type. These were formerly of the double-seated poppet type. The difficulty of keeping the two seats tight when using superheat, led some builders to adopt the balanced single-seated valve. Valves and seats for low temperatures are generally of close-grained cast iron, well annealed and seasoned before final turning, to remove the growth in the iron. Such valves must be kept absolutely steam-tight when closed. Monel metal and Everbrite seats are used with superheat. Alloy steel bodies and valves must be used with high temperatures.

Secondary Valves are generally of the balanced type, and can be adjusted to open at any desired amount of load. In large units, a multiplicity of control valves is provided to give better efficiency over a wide range of load. Single-seated valves, stream-lined to reduce eddy losses, are used on nozzle-controlled turbines.

Steam Strainers in front of the throttle valve prevent grit, pipe scale or other foreign substances from reaching the turbine. These are made of perforated brass, with holes $^1/_{16}$ to $^1/_8$ in. diameter, and generally in basket form, although flat strainer cages are used on some units. The steam chest of many large units is bolted rigidly to the turbine foundation and connected to the cylinder by several flexible steel pipes. This removes any pipe stress from the turbine casing.

For methods of testing governors see A.S.M.E. Code for Speed Responsive Governors.

OVERSPEED GOVERNOR.—All turbines have some form of automatic overspeed governor. A small centrifugal governor sometimes is provided in the end of the spindle shaft. This trips a latch at about 10% above normal speed, allowing a heavy weight to drop and unlatch a spring-loaded throttle valve which instantly closes. The usual form of overspeed governor consists of a bolt-headed pin in the spindle end at right angles to its axis. The centrifugal force on the unbalanced bolt-head is opposed, up to 10% overspeed, by a spring. At that speed, the bolt flies out, striking a trigger, which releases the spring-loaded throttle valve and permits it to close. On some large machines, the trigger operates a steam relay valve which causes a second small piston to unlatch the main throttle valve and allows it to close. A modification of this form of governor is an eccentric ring on the governor spindle, the centrifugal force of the eccentric part being balanced by a spring. At a given overspeed this spring is compressed, the eccentric ring strikes a trigger and the throttle valve is closed either by a spring or by an oil relay.

LOAD RELEASE, particularly when the turbine is provided with several stages of bleeding, may cause serious overspeeding. *Turbines* (N.E.L.A., 1927) presents data on a 30,000-kw. turbine where the speed rose 13% before slowing on release of full load.

Westinghouse has developed an anticipator device which closes the main throttle upon complete loss of load of the generator and before the rotor has speeded up sufficiently to operate the overspeed governor.

Mixed pressure and extraction turbines may have quite complicated governing gear to regulate the steam from two sources, to maintain constant pressure on the bleeder supply outlet, or to meet the electrical demands on the generator end. Also, where steam accumulators are used alone or as auxiliaries to the normal steam supply, the governing gear must control several variable conditions. For discussions of such governor gear see: Kraft, Steam Turbines; Zietemann, Dampfturbinen; Stein, Regelung und Ausgleich in Dampfanlagen.

BLEEDER PRESSURE CONTROL may be achieved by a pressure regulating valve outside of the casing, which returns steam to the low-pressure section of the turbine when the bleeder pressure reaches a desired amount. In other designs, it consists of a pressure-controlled piston outside of the casing which, through a link, controls an oscillating port ring or valve in the casing. This port ring or valve can oscillate to give any degree of opening to the ports leading to the nozzles of the low-pressure portion of the turbine or to shut them off entirely.

THROTTLE VALVES often are opened against a heavy spring. The handle for opening the valve is fastened to the valve stem by a latch which may be released, usually by an oil relay, when the overspeed governor acts, thus forming a combined throttle and trip valve. It serves as a throttling valve only when coming up to speed. The unit always should be shut down by tripping this throttle valve.

Other overspeed governors on small turbines release a flap valve, which closes of its own weight. Still others control a butterfly valve in the steam supply.

When any of these overspeed governors operate, the speed of the turbine must decrease considerably below normal before the valve can be hooked up and the governor gear reset.

11. STEAM TURBINE CASINGS

CASING MATERIALS.—Casings of simple impulse turbines for low steam pressures and temperatures are made of cast iron, often split horizontally, with all steam and exhaust connections usually in the lower half. Several designs split the casing vertically, where one head may be removed and the wheel and shaft removed. The steam chest sometimes is carried around one end in the form of a cored passage, with hand-controlled valves to the nozzles. When high-temperature steam is used, casings must be made of steel. Casings for pressures of 1200 lb. per sq. in. and higher have been made of forgings.

Casings for large turbines usually are split horizontally, so that the upper half can be removed to permit examination and repairs. High-pressure sections of turbines carrying temperatures over 425° F. should be of cast steel.

CASING CONSTRUCTION.—While strength is an important consideration in casing design, rigidity under varying temperature conditions is more desirable. Hence, they usually are cast without ribs on the outside and with only sufficient flanges for bolts. They approach the smooth barrel form, and all large abrupt changes in diameter should be avoided. If flanges or ribs are provided, these must not be too deep, or distortion will occur due to slow adjustment to changes in temperature. Cored passages are to be avoided.

Stationary dummy rings often are cast separately and fastened to the inside of the casings.

Gibb (Post-war Land Turbine Development, *Proc.* Inst. Mech. Engrs., 1931) gives

specifications for cast steel casings as: Ultimate strength, 60,000 to 72,000 lb. per sq. in.; yield point, 30,000 to 35,000 lb. per sq. in.; elongation, 25%; reduction in area, 45%.

Curves of temperature measurements on the flanges and bolts of a 35,000-kw. Metropolitan-Vickers turbine, 350 lb. per sq. in., 750° F. are given in *Engg.*, Mar. 11, 1932 These indicate that flange and bolt temperatures lag several hours behind steam temperatures on starting up, but follow closely any subsequent changes. The bolts are substantially at flange temperature; the maximum difference noted was 75° F.

Narrow flange faces are used on high-pressure turbines to increase the unit compressive stress on the joint up to 10,000 lb. per sq. in. A groove sometimes is cut inside the bolt holes and connected to the exhaust to reduce the pressure difference across the outer portion of the joint. In some cylinders, subject to high pressures and temperatures, vertical saw-cuts are made from the outside of the flange to the bolt holes so that expansion and contraction of the casing wall will be influenced as little as possible by the temperature of the outer part of the horizontal flanges.

The high-pressure end of casings often is supported on sliding feet, located near the horizontal center line, to allow free expansion. Transverse alignment is obtained by vertical keys. In some designs, a flexible support, in the form of a steel I-beam or channel under the high-pressure end, holds the unit in sidewise and vertical alignment but permits longitudinal expansion through flexure of its web. With such construction, the cylinder is bolted solidly to the pedestal which rests on the I-section.

Some single-disc, overhung turbines have two safety blades, placed 120° apart, which rupture at a predetermined overspeed, thereby unbalancing the turbine. The shaft end then rubs on a heavy steel restraining ring in the casing, which braking action prevents further overspeed.

Casings must be designed to withstand bursting pressures 50% above normal working load. For low-pressure cylinders, this may be taken as 30 to 50 lb. per sq. in. The deflection between supports should be a minimum. The pull of the vacuum on the exhaust outlet, when an expansion joint is used, must be considered in figuring this deflection. The lifting force of a spring-supported condenser also must be considered.

Cylinders should be blanked off and subjected to about 25 lb. steam pressure for about 24 hours before final boring, to season the metal and relieve casting stresses. Cylinder supports should be as near to the center line of the machine as possible, to avoid changes in shaft alignment from heat changes. Usually the generator end of the casing is fastened rigidly to the bedplate with provision for expansion at the opposite end.

The thickness of the metal casing is usually calculated by the thin cylinder formula, $t = pd/2f$, where t = thickness, in.; p = internal pressure, lb. per sq. in.; d = internal diam., in.; f = allowable stress in the material, lb. per sq. in. See Stodola (Steam Turbines) for further details.

Drainage grooves to remove moisture are provided in many of the last stages of large turbines. These appear capable of removing 25% of the moisture present.

BOLTS as large as 4 in. diam. are in use. When used at high temperatures and pressures they are tightened to a definite stretch. Each bolt is heated to a certain temperature by an electric heater placed in a central hole in the bolt. The nut is set up snug and the bolt allowed to cool. The resultant stretch then is checked. Similar heating is used when bolts are to be removed. Bolts or studs for high temperatures sometimes are made extra long with a collar under the nut to spread the stress on heating up over such length as to avoid over-stressing.

Chrome-nickel-steel of 140,000 lb. per sq. in. ultimate strength, and yield point of 90,000 lb. per sq. in., has been used for bolts. Creep at the working temperature must be taken into consideration when deciding upon the loading of the bolts. Thompson and Van Duzer (*Trans.* A.S.M.E., FSP–56–9, July, 1934), state that a heat-treated chisel steel gave the best performance with 1000° F. steam. Its composition was: Carbon, 0.45 to 0.50%; chromium, 1.25%; tungsten, 2.0%; vanadium, 0.25%.

DIAPHRAGMS, when cast in halves, must be made sufficiently strong and rigid to avoid undue deformations due to temperature and pressure differences. They usually are dished towards the high-pressure side to increase strength. Diaphragms in large machines are of cast steel, fitting on centralizing supports in turned circumferential slots in the casing. These diaphragms sometimes carry a shoulder projecting almost to the adjoining diaphragms and just outside the blade ring. If the disc fails, the flying pieces must rupture this heavy steel shoulder before piercing the casing.

Diaphragms generally are built with the orifices around the outside made of sheet steel partitions, varying in thickness from $^1/_{16}$ to $^3/_{16}$ in., cast into the diaphragm. In another form of construction, properly-formed brass or bronze guide vanes are spaced and held by pins on the outer edge of the cast-iron diaphragm, a steel ring being shrunk on their outer edge to hold all the guide vanes in place. Still another form consists of

bronze crescent-shaped blades, placed in nozzle segments and bolted to the periphery of the diaphragm. On small diaphragms the partitions are carefully located and hydrogen-welded in place. Another design has a steel central section with accurately formed nozzles mounted straddling the outer edge of this central part, like blades, and riveted or welded in place. The outer ends form a solid rim for the diaphragm. This produces robust, rigid nozzles of high efficiency.

Design constants for diaphragms are largely based upon experience, since split diaphragms are not easily analyzed mathematically.

EXHAUST OUTLETS of large turbines are stiffened and reinforced against the crushing effect of the vacuum. This is done by cast-in or bolted partitions, or by means of heavy staybolts with extra heavy pipe spacers. These partitions or struts must not be placed close to the last blade row since they may cause undue blade vibration.

Kraft (The Modern Steam Turbine) states that the design of the exhaust should fulfil the following conditions: *a*. The transformation into pressure of the kinetic energy of the steam leaving the blades must begin as soon as possible after the last wheel; it must take place as quickly as possible yet it must be gradual; *b*. The curvature must be gentle; the area of the steam passage in a direction perpendicular to the flow must increase continuously, causing the steam velocity to decrease gradually; *c*. The jets of steam escaping from the last wheel must not interfere with each other and cause eddies; *d*. As few guiding surfaces as possible should be provided in the exhaust casing, to keep down the friction losses against the walls. Diffusing guides are employed in all large exhausts. See Robinson, Leaving Velocity and Exhaust Loss in Steam Turbines, *Trans.*, A.S.M.E., FSP-56-10, July, 1934, for data on exhaust hood losses.

ATMOSPHERIC EXHAUST.—Provision usually is made for an atmospheric exhaust on all condensing turbines, to discharge exhaust steam to atmosphere through a relief valve. These valves are difficult to keep tight. The valve and piping are expensive and take up room in a plant. In modern operation, where machines are started and shut down under vacuum, it is considered improper to operate non-condensing, particularly when high superheat is used. Hence the relief valves are seldom, if ever, used. Turbines have been installed with no relief valve, but with a lead blow-out or explosion diaphragm set at 2 lb. per sq. in. gage and placed over the atmospheric port. These turbines have a special vacuum control device which trips the throttle valve if the vacuum drops below a certain value.

12. ERECTION AND OPERATION

Turbine Foundations

The foundation should have sufficient weight and mass to hold the turbine rigid against vibration. The sub-foundations are determined by the character of underlying material, but must be so designed that the concentrated weights of the turbo-generator are spread over an ample area to prevent springing or settling. For bearing power of soils see Vol. 3.

A type of foundation extensively used is reinforced concrete, with heavy side walls and columns under the base of the unit, tied together by deep arching bridges. Necessary openings for condenser connections, air ducts, drain and steam piping, etc., should be provided together with additional reinforcing around these holes. The maximum unit pressure of the turbine and generator on the concrete should not exceed 250 lb. per sq. in. *Reinforced concrete foundations* are heavy and impose added loads upon the sub-foundation. They generally are used where the footings rest directly on rock. The concrete superstructure occupies space that might be used for auxiliary equipment. To overcome this objection *steel foundations enclosed in concrete* have been used to reduce the mass. Concrete should consist of a 1–2–4 mixture, well placed and seasoned.

STRUCTURAL STEEL FOUNDATIONS are widely used, as these permit the placing of much auxiliary equipment under the turbine itself. This type imposes least weight upon the sub-foundations, and hence is favored where the footings are supported upon piling, caissons, or a reinforced concrete mat. The steel piers are connected by a deep I-beam superstructure to carry the unit. The supports must be strongly buttressed and braced to prevent deflections and vibrations. The beams must be designed for a maximum deflection under load of 0.020 to 0.030 in. Columns should not deflect axially more than 0.030 in. The allowable deflections usually are stated upon the turbine builder's foundation plans.

THE DESIGN AND CONSTRUCTION OF TURBINE FOUNDATIONS usually is carried out by the purchaser. While the turbine builder will assume no responsibility for such plans, it is desirable to submit them to the builder for his approval in order to avoid mistakes, omissions of openings for drains, ducts, etc., and for suggestions as to resonance, rigidity, etc.

Foundations generally are designed with rigidity, mass, etc., solely in view, without

regard to resonance. Resonance must be considered, particularly with structural steel foundations, for periodic vibrations may occur corresponding to the r.p.m. of the turbine and to certain multiples of this speed. This vibration may be horizontal or vertical and may affect different parts of the foundation in opposite ways. The foundation may even be too stiff for the particular turbine load and speed, and may vibrate severely as a result. Resonance in foundations may lead to objectional vibration in turbines themselves.

In case of vibration trouble, a study of the structure by means of a vibrometer may indicate means for removing the difficulty. These instruments may be used to analyze vibrations in the turbines. See T. C. Rathbone, Vibration of Turbine-generator Foundations (*Power*, April 3, 10, 1928) and Dohme, Modern Steam Turbine Foundations (*Power*, Aug. 2, 1927).

Provision must be made in foundations for the necessary air ducts for air coolers, when used, for pipe openings, and for cable and drain outlets. Provision also must be made for the down-pull of the vacuum if an expansion joint is placed between turbine and condenser. This down-pull equals the product of the area of the exhaust opening in sq. in., multiplied by the vacuum, as shown by a mercury column reduced to absolute pressure in lb. per sq. in. In some turbine designs this pull is carried entirely by the exhaust end supports. In others it is divided between front and rear supports. When the condenser rests on springs and is bolted directly to the turbine exhaust, there is no down-pull due to vacuum, but there may be an upward lift from the springs when the condenser is empty. Foundations should be designed to support machine load plus 25% for impact, condenser load, floor load, and dead loads. Cross bracing must be provided to care for a force equal to 25% of the machine load.

Some turbines are leveled by steel shims on the steel work, and bolted in place. In other cases, the steel is so designed that a concrete top is provided and 1 to 1 grout is poured around the bed plates in the usual way after leveling with wedges.

Floors between adjacent turbines in a power-house are frequently built across between turbine foundations. Vibrations thus are communicated to all parts of the power-plant. To prevent this transmission of vibration, the floor beams may be carried on separate columns leaving a space 1 in. wide between floor and foundation. This afterwards is filled with a board of soft wood or a rubber strip is fitted loosely in the opening. These do not transmit vibration.

In many large stations only an operating platform about 6 ft. wide is built around the turbine bed plate, leaving the remainder of the engine-room open to the basement floor. This provides better light around the condenser auxiliaries, and the condenser equipment is more accessible to the crane. Dismantling and repairs can be carried out on the basement floor. If these were done on a floor at the turbine elevation, heavy steel or concrete flooring and floor supports would be required.

Erection of Steam Turbines

Before proceeding with the erection of a steam turbine one should be sure that the foundation is sufficiently strong and rigid to carry the unit. This is particularly necessary if the turbine is to be placed on the engine-room floor, as is sometimes done with small units. In erection, if turbine and generator are on one shaft, it is only necessary to level the unit carefully. If the set consists of a turbine with a separately-driven unit, as a generator or pump, or if gears are used, care must be taken to insure not only correct levels but also accurate alignment, particularly at the couplings. Leveling pads are furnished on many bed-plates. Levels should be checked with the turbine heated to operating temperature, as the alignment of some designs is affected by expansion due to the heating of the exhaust end. Leveling is done by steel wedges at frequent intervals under the edge of the bed-plate, allowing a space of 1 in. under the bed-plate for grouting.

There should be sufficient wedges to insure that no deformation occurs between them. After the grout has set for a day, they can be withdrawn. Some engineers slightly raise the outboard bearings of both turbine and generator to obtain more accurate alignment at the coupling by allowing for the deflection due to weight of the spindle and revolving field. This, however, is unnecessary with most couplings. Leveling always should be done with all weights in place. Detailed information on erection will be found in the instruction books of manufacturers. Kearton's Steam Turbine Operation also contains many data and suggestions for alignment methods, erection, etc.

When leveling is completed, the bed-plate is grouted in place, using one part of high-grade Portland cement and one part of sharp sand, mixed in a thin grout and well rammed to prevent air bubbles under the base. This should cover the foot of the bed-plate at least 3 in. Provision usually is made to grout in bed-plates supported on steel. Occasionally lead, 1/2 to 1 in. thick, is used in place of grout on structural steel foundations.

Sometimes the inside of the bed-plate is completely filled with cement to add more mass to the foundation and lessen vibration.

Some bed-plates are provided with foundation bolts. These are desirable, but not necessary. They are drawn up tight after the grout has set.

Great care must be exercised when erecting the unit to prevent any dirt lodging in the bearings or oil chambers, or foreign matter in the blading. All oil chambers should be carefully cleaned, but waste must not be used on account of the lint left behind.

Oil piping generally is blown out with steam and allowed to dry thoroughly before assembly, to remove dirt that may have gotten in during shipment. Steam lines should be blown out with full steam pressure before permanently connecting up, to remove any pipe scale or dirt from the inside.

In assembling turbo-generators, great care must be taken to avoid injuring the stator coils or bruising the laminations when entering the rotating field into the stator frame. This may be prevented by placing wooden strips or bearing plates on the laminations.

Steam Piping

PIPING to turbines must provide for expansion and contraction, both of the turbine and of the pipe line, so that the least possible stress will be imposed on the turbine. Pipe vibrations also must be avoided.

The steam chest containing the governor regulating valves is either, *a*, bolted directly to the turbine casing, or, *b*, fastened firmly to the foundation with flexible pipe connections from it to the turbine casing. The throttle valve, usually of the automatic spring-closing type operated by the overspeed governor, bolts directly to the steam chest and may be supported by springs, or in case *b*, be supported by the foundation. In the latter case stresses from the piping cannot be transferred to the casing.

In both cases, it is desirable, in order to keep the flange joints tight, that there be minimum stress on the throttle valve from piping. Allowance should be made in case *a*, for the maximum turbine expansion from cold to working temperature, and the same for the piping. This expansion may be halved and piping made up cold by springing into position for half the total expansion. Torsional stresses also should be avoided. See Sect. 5 for various data on pipe, flanges, etc.

Practice is tending towards welded pipe joints for high pressure piping. In some cases this includes welds between piping, throttle valve and steam chest. See Field Welds in Pressure Pipe Lines of Steam Systems, H. N. Boetcher, *Trans.* A.S.M.E., FSP–56–1, Jan., 1934.

Drains must be provided for all low points in piping, steam chests, bends, etc. The manufacturer's drawings should show the location of these drains.

Exhaust Piping must be absolutely air tight. When an expansion joint is used, consideration must be given to the collapsing effect of the vacuum, and provision made by suitable brackets, anchor bolts or other devices to prevent distortion.

Plant Design

Turbo-generator plants are arranged in either of two ways: *a*. The turbo-generators are set with their axes at right angles to the boiler-room wall. Condensers are placed directly under the turbines. The units are spaced to give ample room between them for operating and for dismantling. This distance often is fixed by the space required by auxiliaries, which usually are in the basement. Space must be provided at the ends of the units to permit the field to be withdrawn from the generator. With large units, this results in a wide turbine room and long crane span. *b*. Large turbines often are placed with their axes parallel to the boiler-room wall. The distance between turbines is fixed by the space necessary to withdraw the generator field. The condenser usually is set at right angles to the turbine axis. Space must be allowed for withdrawing condenser tubes, and also at the side of the turbine for dismantling. Vertical condensers at the side of the turbine or in condenser pits frequently are used.

THE CONDENSER in every case should be placed as close to the turbine exhaust as possible. In small sizes, a bellows type copper expansion joint must be placed between turbine exhaust and the condenser inlet. Large units often are built with no expansion joint, the condenser being bolted directly to the exhaust nozzle. The supporting pads on the condenser shell rest on springs. These are adjusted to carry all, or a major portion of, the weight of the condenser when full of water. They provide a certain amount of flexibility to care for expansion from temperature changes. All water, air pump, condensate pump and atmospheric exhaust connections should have expansion joints when the condenser is bolted firmly to the turbine in this manner. Rubber expansion joints are used for this purpose.

HEADROOM UNDER THE CRANE above engine-room floor must be sufficient to permit lifting off the turbine cover and removing the spindle for repairs or inspection.

PIPING.—The purchaser furnishes piping for cooling water to and from the oil cooler, to and from the regenerator air cooler, for water glands, for steam to and from auxiliary oil pump and for drip connections. Manufacturers' drawings will show where these connections can be made. The manufacturer usually supplies all oil piping, piping to steam-sealed glands and water cooling piping about bearings when such is used.

COOLING AIR for small generators may be taken from below the unit, and after passing through the generator, may be discharged into the engine-room. In some of the older units, air from outside is passed through washers before entering the generator and the discharged air is delivered to the boiler-room. This involves much duct work. In both the above methods, much dirt enters the generator and adheres to the windings. In case of a short-circuit, a fire in the windings usually results from the continued supply of fresh air. Units now are furnished with a closed system of air circulation and with air coolers supplied with condensate or with cooling water from other sources. These eliminate the necessity of cleaning generators, and in case of a short-circuit, the fire is smothered by the exhaustion of the limited oxygen supply in the closed system. Small breathers with viscous air filters provide for changes in volume in this closed circuit.

One manufacturer uses the following formulas to give the approximate amounts of cooling air that are circulated through generators rated at 80% power factor. For 3600-r.p.m. generators from 3000 to 15,000 kw. rating inclusive, cu. ft. per min. of air $= 3.7 \times$ kw. rating/1000. For 1800-r.p.m. generators from 20,000 to 60,000 kw. rating inclusive, cu. ft. per min. of air $= 2.4 \times$ kw. rating/6000.

FLOOR SPACE.—Table 4 gives the approximate overall floor space required by General Electric alternating-current turbo-generators for 80% power factor and with direct-connected exciters. Table 5 gives similar data on Allis-Chalmers Parsons turbines.

Table 4.—Space Required by General Electric Turbo-alternators for 80% Power Factor and Direct-connected Exciters

Kw. Rating, 80% Power Factor	Non-Condensing						Kw. Rating, 80% Power Factor	Condensing					
	Length, ft.	in.	Width, ft.	in.	Height, ft.	in.		Length, ft.	in.	Width, ft.	in.	Height, ft.	in.
10	6	10	2	1	2	1	500	17	2	5	6	5	4
15	7	3	2	1	2	1	600	17	5	5	6	5	4
20	7	3	2	1	2	1	750	17	11	5	6	5	7
25	7	4	3	2	2	8	1,000	18	2	5	11	5	9
35	7	4	3	2	2	8	1,250	18	6	5	11	5	9
50	9	8	3	10	3	5	1,500	20	8	7	6	7	1
75	12	5	3	11	4	10	2,000	21	7	7	6	7	1
100	12	7	3	11	4	10	2,500	22	9	7	6	7	1
150	13	2	4	5	4	10	3,000	24	7	8	7	7	1
200	13	4	4	5	4	10	3,500	25	3	8	7	7	1
300	14	6	4	6	4	10	4,000	26	2	10	2	7	1
400	15	2	5	4	4	11	5,000	29	1	10	10	6	2
							6,000	29	9	11	2	6	8
							7,500	30	9	11	2	6	8
							10,000	33	8	11	8	8	7
							12,500	35	4	11	8	8	7
							15,000	38	3	11	8	8	7

Table 5.—Floor Space Required by Allis-Chalmers Parsons 3600-r.p.m. Turbines

Kw.	Length, ft.	in.	Width, ft.	in.	Height, ft.	in.	Shipping Weight, lb.	Kw.	Length, ft.	in.	Width, ft.	in.	Height, ft.	in.	Shipping Weight, lb.
250	15	11	4	3	5	7	22,400	2000	26	2	9	0	6	0	107,200
500	17	1	4	3	5	7	28,700	2500	27	10	10	1	6	0	115,000
750	18	11	6	11	6	5	47,200	3000	28	7	10	3	6	0	123,000
1000	19	9	6	11	6	5	56,600	4000	29	6	10	6	6	1	134,000
1250	22	9	9	0	4	11	76,100	5000	29	8	10	6	6	1	153,600
1500	24	10	9	0	6	0	88,200	6000	33	8	12	10	6	2	168,000

Care and Operation of Steam Turbines

Manufacturers issue complete instructions for the care and operation of their respective units. The Operating Code Manual for Prime Mover Plant, N.E.L.A., 1929, should be available to every operator. The following notes direct attention to a few operating considerations:

The logical steps in starting a turbine are: First, the condenser circulating pump and the other condenser auxiliaries should be started; next, the auxiliary oil pump on the turbine, and a full oil

supply furnished to all bearings. Cooling water should be turned on the oil cooler. Any new oil added to the system should be poured through several thicknesses of cheese-cloth to remove any chips, cuttings, etc.

The most difficult part of starting is the proper warming-up of spindle and casing. With steam-sealed glands, the vacuum pumps are started and steam turned into the glands. A small amount of steam allowed to pass the throttle valve will, on account of its density, rise to the top of the casing, causing it to heat and expand. The lower half remains filled with cold air and does not change in temperature or form. The result is a distorted cylinder. The same thing occurs on a spindle of Parsons construction. Discs also distort if one-half only is heated. It is evident that warming with a small amount of steam may cause undesirable distortions, which may result in blade rubs. A better method is to suddenly admit enough steam through the throttle valve to revolve the spindle, and to then close the throttle until only sufficient steam enters to keep the spindle turning over slowly. The rotating blades carry the steam around the casing, causing it to heat more evenly and rapidly than in any other way. This should continue until the turbine is evenly heated, before allowing the machine to speed up. All drains must be kept open during the warming process. A blade rub developed in starting probably is due to local distortion, and often may be relieved by allowing the turbine to stand for a short time while the heat in the casing diffuses through the whole body. Another careful start then may indicate that the blades are clear. It is unwise to bring a turbine up to speed without warming, as severe stresses undoubtedly are produced in certain parts.

STARTING UP AND LOADING are important considerations on large turbines. The time required to bring turbines of 20,000 kw. and larger from cold up to speed varies from $1/2$ to 3 hr., and load may be added at the rate of 1000 to 3000 kw. per min. If possible large turbines ought not to be operated at less than 20% of rating.

Large spindles must be warmed and cooled with great care to avoid bowing of the spindle by uneven heating or cooling. Indicators are provided on the shafts of many turbines to show whether the spindle is straight. If a spindle is bowed, it must be brought to rest and allowed to stand until all parts are normal and the bend has disappeared.

Large units are provided with motor-driven turning gear which can continuously revolve the spindle at about 25 r.p.m. during the shut-down period to keep the spindle straight. This gear automatically disengages when starting up. Sometimes oil at high pressure must be supplied to the bearings when starting to establish the oil film.

E. Brown (*Engg.*, Aug. 22, 1930) proposes placing a heat-insulated cover outside the casing, and heating the intervening space electrically to normal operating temperature during the shut-down period to permit instantaneous starting. He states that 0.1 kw.-hr. per sq. ft. of outside cover will maintain the turbine in a hot state. This is equivalent to 0.1% of full-load steam per hour.

When turbines connected to generators, and in parallel with water-power plants, operate for considerable periods at no load or at times are motored, sufficient steam must be admitted past the governing valves to cool and ventilate the blades.

As the turbine speeds up the glands begin to seal and the vacuum builds up. It is well to ascertain if the governor-regulating valve is sufficiently tight to prevent overspeeding with no load and full vacuum on the turbine. The turbine next can be synchronized and the load added.

In operation, the oil supply to the bearings requires constant watching.

SHUTTING DOWN.—The turbine is shut down by reversing the above processes in the regular order. The machine usually is stopped by tripping the emergency valve, to see that it is free and acting properly, or by speeding up the turbine by holding the governor lever until the emergency governor itself acts, at 10% above normal speed.

It is inadvisable to throw large loads suddenly on turbines, as this may cause the boilers to prime and throw water into the superheater. Large, efficient separators before the throttle valve are absolutely essential where wet steam is used.

If the exhaust pipe turns up after leaving the turbine, and rises to a higher level, the pocket formed must be drained. Otherwise it will fill with water and lower the vacuum on the last rows of blades. A tank pump is best for drainage of such a pipe.

INSPECTION.—Every turbine should be opened up and inspected periodically, usually once a year, to note any wear of parts or other troubles. If wet steam is used from boilers that prime or foam, the blades in the first rows may become clogged with material carried over in the steam. Corrosion of blades in rows at the dew point generally is due to impure feed water or to no deaeration. Wet steam may cause erosion of the last rows of blades. Oil reservoirs require cleaning. Water glands scale up, unless condensate only is used. Clearances in the cylinder require checking. Wear on thrust blocks should be noted and corrected. Leakage in throttle and regulating valve seats should be stopped. Care must be taken in reassembling to avoid damage to the blading.

The highest vacuum should be maintained at all times in the condenser exhaust. Frequent use of heavy asphaltum paint on all exhaust joints is a good preventive against air leaks.

Care must be taken in tightening packing on governor valve or automatic valve stems to make sure that these are not made so tight as to cause the valve to stick in service.

Accidents to steam turbines generally are due to one of the following causes: Failure of the oil supply; overspeeding, due to failure of the automatic overspeed governor to act; failure of blades due to fatigue of material from vibration; fouling of blades with foreign parts in the casing; failure of the discs or drum from internal defects; or starting quickly an unevenly-heated and distorted turbine. Some of these conditions can be foreseen and prevented by proper vigilance on the part of the operating engineer.

TURBINE OPERATING DATA for a period of years will be found in *Turbines*, N.E.L.A. and E.E.I. The following terms have been adopted as standards in stating turbine performance:

Period Hours.—Total hours per year; 8760 in 1933.

Service Demand Factor.—The ratio of demand hours to period hours.

Service Demand Availability Factor.—The ratio of service hours to demand hours.

Unit Capacity Factor.—The ratio of kilowatt-hours generated to the product of the unit rating and period hours.

Unit Output Factor.—The ratio of kilowatt-hours generated to the product of the unit rating and service hours.

Unit Operation Factor.—The ratio of service hours to period hours.

Maximum Possible Unit Operation Factor.—The ratio of the sum of the service hours and the reserve hours to the period hours.

Table 6 from *Turbines*, E.E.I., 1934, gives such performance data over a period of years. This shows a steady improvement in those factors associated with availability and reliability and also indicates the outage factors of the complete unit. The effects of the depression after 1930 also can be noted.

OPERATION INDICATORS.—The erection of turbo-generator sets in the open has necessitated a more complete enclosure of all parts and the development of new instruments to show the conditions of operation. Instruments are under development to show the expansion of spindle relative to casing, to detect the degree of vibration or changes in vibration, to indicate the touching or rubbing of parts inside the casing, shaft eccentricity and axial clearance, and to control valves from remote stations. These instruments probably will be placed in all new stations, open or enclosed. See Apparatus for Steam Turbine Control, R. B. Smith, *Elec. Jour.*, Sept., 1932.

Availability and Reliability are important factors in turbine construction and opera-

Table 6.—Average Performance of Units of 20,000 Kilowatts Capacity and Larger

	Period Analyzed					
	1914–1921	1922	1923	1925	1926	1927
Turbine units reviewed	78	74	87	191	153	186
Service demand factor, percent	77.80	74.80	78.40	68.80	70.55	71.11
Service demand availability factor, percent	85.50	71.00	92.30	94.60	92.16	94.62
Unit capacity factor, percent	43.80	48.10	51.60	44.40	49.20	46.13
Unit output factor, percent	66.00	70.60	71.30	68.20	64.44	66.89
Service hours factor, percent	66.50	68.10	72.40	65.10	67.06	67.14
Turbine outage factor, percent	6.93	7.90	7.33	7.29	6.28	6.36
Generator outage factor, percent	2.56	3.10	1.74	1.85	1.70	1.68
Condenser outage factor, percent	1.20	2.50	3.18	3.51	2.65	2.75
Other causes outage factor, percent	0.66	1.80	0.66	0.98	0.57	1.06
Total outage factor, percent	11.35	15.30	12.91	13.63	11.20	11.85
Reserve hours factor, percent	22.05	16.60	14.69	21.25	21.74	21.04
Maximum possible unit operation	88.55	84.70	87.09	86.35	88.80	88.18

	1928	1929	1930	1931	1932	1933
Turbine units reviewed	207	276	324	334	291	310
Service demand factor, percent	72.09	69.43	65.58	60.40	51.01	50.30
Service demand availability factor, percent	96.01	96.27	96.01	96.50	95.73	96.36
Unit capacity factor, percent	48.15	40.57	43.35	38.42	30.52	34.97
Unit output factor, percent	69.44	60.71	68.86	63.65	62.50	72.15
Service hours factor, percent	68.58	66.84	62.96	57.93	48.83	48.47
Turbine outage factor, percent	5.73	5.02	4.44	4.20	3.90	5.22
Generator outage factor, percent	1.13	1.47	1.52	0.95	0.69	0.80
Condenser outage factor, percent	2.77	2.61	2.51	2.16	1.75	1.42
Other causes outage factor, percent	0.57	0.72	0.70	1.06	0.82	0.62
Total outage factor, percent	10.22	9.82	9.17	8.37	7.16	8.06
Reserve hours factor, percent	21.22	23.34	27.87	33.70	44.01	43.47
Maximum possible unit operation	89.80	90.18	90.83	91.63	92.84	91.94

tion, since outages not only cost money but may lead to shut-downs of the entire electrical system. Possible gains in economy sometimes have been sacrificed to obtain more rugged and reliable operating units.

SYNCHRONOUS CONDENSER OPERATION of steam turbo-generator sets often is practiced. Generally the manufacturer refuses to accept responsibility for such an operating condition, although few failures have been assigned to this cause. If the turbine is still coupled to the generator during such operation, particular care must be taken with disc-type turbines to properly ventilate the unit with steam to remove the heat generated by windage in the various stages. This is done by admitting steam through an orifice into the first stage of the casing. This steam, while supplying some of the energy to overcome the mechanical losses of the set, is less than the normal no-load steam. The amount of heat generated, and the location of the hottest stage depends upon the size, type and number of stages and on the density in the turbine casing. The condenser usually is kept in service to maintain a high vacuum. Thermometers are placed on the turbine casing to indicate the rise in temperature when operated as a synchronous condenser. The limiting temperatures are about 500° F. for a steel casing and 400° F. for a cast-iron casing for small units. In large turbines, manufacturers have placed even lower limits.

When the turbine is uncoupled and the generator operated alone as a synchronous condenser, it is started as a motor by being tied-in electrically with another operating unit and brought up to speed.

13. CORRECTION FACTORS FOR TURBINE DATA

AN INCREASE IN STEAM PRESSURES increases the heat drop, which tends to reduce steam consumption at a given load and leads to lower leaving losses. Increased density leads to higher friction losses in the high-pressure section of the turbine. If initial temperature is constant, moisture increases in the low-pressure stages, tending to decrease efficiency. ρ, the ratio of wheel to steam speeds, decreases for each stage but R, the reheat factor, increases. Hence cumulative heat increases. The net result is a slight decrease in efficiency, together with lower steam consumption.

AN INCREASE IN VACUUM increases the heat drop and decreases steam consumption. ρ decreases, tending to lower efficiency. The leaving loss increases. The net result is lowered efficiency, but also decreased steam consumption.

AN INCREASE IN INITIAL STEAM TEMPERATURE decreases density of the high-pressure steam and tends to increase efficiency. It increases the heat drop, and decreases ρ, which tends to decrease efficiency. This often is offset by the decreased moisture in the low-pressure section, although leaving losses tend to increase. The net effect is a slight increase in efficiency, accompanied by a decrease in steam consumption.

CORRECTION FACTORS.—As it may be impossible to reproduce on a plant test exactly the standard conditions specified in the contract, every steam turbine guarantee should contain in the contract, the corrections agreed upon for such variations from standard conditions as may occur on test. The corrections will vary with the various types and sizes of turbines, and with certain assumptions in their designs, and should cover variations in initial pressure, superheat or quality, vacuum and load.

Only the manufacturer can state, with any degree of assurance, reasonable correction factors for his particular design. The purchaser, however, can check these by noting whether there is any appreciable change in the engine efficiency when the corrections are applied.

In the interests of fairness and accuracy, the purchaser should ask for guarantees only under actual every-day operating conditions, such as usual throttle pressure, superheat or quality and vacuum, which he reasonably can obtain on test, and which will, therefore, not need corrections. True values for corrections can be determined only by a series of tests where the conditions are varied one by one, all other conditions being maintained standard, and the results compared.

Correction factors for all possible conditions cannot be stated for lack of space. The following data (Tendencies in Steam Turbine Development, H. L. Guy, *Proc.* Inst. Mech. Engrs., 1929) will indicate the relative magnitude of the various corrections:

The steam or heat consumption at standard conditions is found by multiplying the measured quantity from the test by the correction factor. Standard conditions are 350 lb. per sq. in. gage; 750° F.; 29 in. vac.; four-stage bleeding to heat feed to 300° F. Although not stated in article, these figures probably apply to 40,000-kw., 1500 r.p.m. units.

The correction for initial steam pressure, temperature, and vacuum is much smaller

if performance is expressed as heat consumption than as steam consumption. The correction factors apply to one specific load, the *Most Efficient Load*. Other factors must be provided for partial or full loads, particularly vacuum correction factors which vary appreciably with load. See Goudie, Steam Turbines, p. 621.

Pressure Correction Factors

Pressure, lb. per sq. in. gage	300	325	350	375	400
For steam consumption	0.979	0.99	1.00	1.011	1.022
For heat consumption	0.982	0.991	1.00	1.009	1.018

A straight line variation may be assumed for lesser increases or decreases.

Superheat Correction Factors

Steam temperature, °F	720	735	750	765	780
For steam consumption	0.980	0.989	1.00	1.012	1.025
For heat consumption	0.989	0.995	1.00	1.007	1.014

Vacuum Correction Factors

Vacuum, in.	28.4	28.6	28.8	29.0	29.2	29.4
For steam consumption	0.968	0.979	0.989	1.00	1.01	1.021
For heat consumption	0.977	0.985	0.993	1.00	1.005	1.009

Feed Temperature Correction Factors

Feed temperature, °F	240	260	280	300	320	340	360
For steam consumption	1.041	1.028	1.015	1.00	0.984	0.966	0.945
For heat consumption	0.985	0.992	0.997	1.00	1.001	1.002	1.002

LOAD CORRECTION where necessary is determined by an interpolation or extrapolation of the Willans Line drawn from the results of actual tests and after all other corrections have been applied. The total steam (or heat) consumption for the output for which the correction is desired, may be determined from the formula:

$$M_c = \frac{(M_t - M_a) \times (W_c - W_a)}{(W_t - W_a)} + M_a$$

where W_c = output for which correction is desired; W_t = output of the test requiring correction; W_a = output for an adjacent test employed for determining the load correction; M_c = total steam (or heat) for the corrected output W_c, lb. (or B.t.u.) per hr.; M_t = total steam (or heat) for the output W_t of the test requiring load correction, otherwise corrected to the specified operating conditions, in lb. (or B.t.u.) per hr.; M_a = total steam (or heat) for the output W_a of the adjacent test, corrected to the specified operating conditions employed in determining the correction, lb. (or B.t.u.) per hr.

RULES FOR STEAM TURBINE TESTS are given in the A.S.M.E. Power Test Code. See p. 16–22.

14. TURBINE PERFORMANCE

Steam turbine performance usually is stated in tables giving size, speed, initial steam and exhaust conditions, and pounds of steam per kw.-hr. These tables, while of interest to engineers for reference, have three limitations: 1. It is impossible in a small space to quote tests for every possible steam condition under which turbines of many sizes may operate; 2. Builders seldom allow any tests to be published except the best records made by their equipment, and such tests obviously do not represent average conditions; 3. Turbine builders are prepared to furnish different types of the same size of unit, and frequently of different efficiencies, built to meet quite different conditions. For instance, a simple, cheap unit, with a low ratio of blade speed to steam speed and high leaving losses may be best suited for certain commercial conditions. For other conditions, a more expensive turbine, of more refined construction, with a high ratio of blade speed to steam speed, and with low leaving losses, may be most desirable. The latter will have a much lower steam consumption and higher cost than the first unit.

The following data will serve as a guide in estimating turbine performance or in checking tests:

ENGINE EFFICIENCY.—Operating conditions, particularly on small turbines, vary over such a wide range that tables covering every condition are beyond space limitations. The engine efficiency is used in the following paragraphs as a measure of performance of the smaller units.

The steam consumption of these units readily can be determined by dividing 3412 (the heat equivalent of 1 kw.) by the product of the engine efficiency, η_e at the generator terminals and the adiabatic heat drop from initial steam conditions to exhaust pressure. Thus lb. per kw-hr. = $3412/[\eta_e(h_1 - h_2)]$

The isentropic heat drop $(h_1 - h_2)$ can be found from a Mollier diagram or can be

calculated from the Steam Tables (see p. 5–03). The steam consumption in terms of brake-horsepower can be calculated by substituting 2543 for the horsepower and η_e for the engine efficiency at the coupling. Thus lb. per B. Hp.-hr. = $2543/[\eta_e(h_1 - h_2)]$

Leaving loss also may be considered at the same time as engine efficiency, as it is a measure of the unutilized velocity from the last row of blades.

Since turbines operate best under the conditions for which they are designed, the efficiency ratio under any other set of operating conditions will vary from the ratio at the specified conditions. On account of the high leaving loss from the last rows of blades of high-vacuum turbines, it often happens that a higher efficiency ratio may be obtained at a somewhat lower vacuum than specified, even though the steam consumption per kw-hr. may be higher.

HEAT CONSUMPTION of a turbine is expressed in B.t.u. per hr. See p. 3–72.

British practice expresses turbine performance in terms of thermal efficiency. This is the ratio of 3412 (the heat equivalent of 1 kw.) to the heat consumption of the unit, expressed in B.t.u. per kw-hr.

Another British practice is to state the k value for a unit where

$$k = \Sigma\left(\frac{d}{10}\right)^2 \times$$

where Σd^2 is the sum of the squares of the mean diameters, in., of all rows. The higher the value of k, the greater is the engine efficiency as a rule. Another similar factor called by Kraft (The Modern Steam Turbine, p. 31) the *Parsons coefficient* or *quality factor* $q = \Sigma u^2/h_0$, where Σu^2 = the sum of the squares of the various wheel speeds, ft. per sec.; and h_0 = isentropic heat drop, B.t.u., from initial conditions to final pressure. Σu^2 gives an idea of the bulk of a turbine and consequently of the price of the unit. Kraft gives the following values of engine efficiency at the coupling for various values of q:

q = 5000	7500	10,000	12,500	15,000	17,500
η_c = 69.8%	77.2%	81.5%	84.2%	85.5%	86%

He points out that an increase in q means an increased number of stages, and that a large increase in q gives only a comparatively small increase in efficiency in the higher ranges. Also, the quality factor q may be higher for reaction turbines for the same efficiency than for impulse units.

Table 7 represents the performance of single-stage velocity-type non-condensing turbines for direct connection to the driven apparatus. Blade speeds vary from 300 to 500 ft. per sec. The higher values of engine efficiency refer to the performance under the most favorable steam conditions for that particular design; the lower values refer to performance under less favorable conditions.

Table 7.—Performance of Single-stage, Non-condensing Steam Turbines

Brake Hp.	Engine Efficiency Referred to B.Hp.	Brake Hp.	Engine Efficiency Referred to B.Hp.	Brake Hp.	Engine Efficiency Referred to B.Hp.
25	34 to 44%	125	41 to 48.5%	300	47 to 53%
50	37 to 46	150	42 to 49	400	48 to 54
75	38 to 47	200	44 to 51		
100	40 to 48	250	45 to 52		

The figures in Table 8 represent best efficiencies for single-stage, direct-connected velocity-type steam turbines operating under the conditions stated, as furnished by one American manufacturer. All figures are for dry saturated or slightly superheated steam. Exhaust pressure, 14.7 lb. per sq. in., abs. Lower efficiencies often will be quoted to obtain a cheaper turbine.

Table 8.—Engine Efficiencies Based on B. Hp. of Single-stage, Velocity-type, Non-condensing Turbines

Rated B.Hp.	Inlet Pressure= 300 lb. per sq. in., gage			Inlet Pressure= 150 lb. per sq. in., gage		
	3600 r.p.m.	2700 r.p.m.	1800 r.p.m.	3600 r.p.m.	2700 r.p.m.	1800 r.p.m.
25	34.5	34.0	30.5	36.2	36.0	32.9
50	39.0	38.0	32.6	40.7	40.0	35.0
100	44.5	42.5	33.6	46.2	44.5	36.0
200	49.0	44.5	34.1	50.5	46.5	36.5
400	51.5	45.5	34.4	53.0	47.5	37.0

Blade speed has an important influence on turbine efficiency. Higher blade speeds can be secured either by a relatively large wheel at moderate speed, or a relatively small

wheel at very high speed. In practice, the latter condition has been found most desirable, as the disc friction and blade losses usually are less for the smaller wheel. The very high speed must be reduced through mechanical gearing to the required slow speed of the driven unit.

The figures of Table 9, applying to single-stage, geared, velocity-type turbines, running at blade speeds of 500 to 650 ft. per sec. show by comparison with Table 7 the improvement to be obtained with high blade speeds obtained through small discs and reduction gearing. While these values do not refer to one specific type of turbine in both cases, it would seem that the increase in blade speeds has resulted in an apparent increase in the efficiency ratio of approximately 5%.

Table 9.—Performance of Non-condensing Single-stage, Geared Turbines of High Blade-speed

Brake Hp.	Engine Efficiency Referred to B.Hp.	Brake Hp.	Engine Efficiency Referred to B.Hp.	Brake Hp.	Engine Efficiency Referred to B.Hp.
25	42 to 51%	125	47 to 55%	300	51 to 58%
50	44 to 53	150	48 to 56	400	53 to 59
75	45 to 54	200	49 to 57		
100	46 to 55	250	50 to 58		

For certain purposes, as driving boiler-feed pumps, fans and other station auxiliaries, steam economy of the driving turbine is usually not a matter of great importance, since the exhaust is used to heat feedwater. Rugged construction and reliability of operation, combined with low first cost are usually the controlling factors in the choice of such a unit.

EFFICIENCY RATIO OF RE-ENTRY TURBINES.—Curves are given by Francis Hodgkinson (Historical Review of Steam Turbine Progress, *Elec. Jour.*, 1918) which indicate that the engine efficiency, based on electrical output of non-condensing, direct-connected, single-wheel units of the re-entry type, such as are used for small lighting outfits by contractors is as follows, with 175-lb. gage pressure dry steam: 12% on a 1-kw. unit, 20% on a 5-kw. unit, and 23% on a 10-kw. unit. Geared turbines of the re-entry type give the following probable engine efficiencies, based on electrical horsepower, with 175-lb. gage pressure dry saturated steam, non-condensing: 50-kw., 44%; 100-kw., 47%; 150-kw., 48.5%; 200-kw., 50%.

SMALL IMPULSE TURBINES.—J. Y. Dahlstrand (Characteristic Curves of Steam Turbines, *Power Plant Engg.*, Aug. 1, 1924) discusses certain main characteristics of turbines of the impulse type below 500 kw., where the steam is throttled into the first-stage nozzles. He points out that:

1. The relation between power output and ring pressure before first-stage nozzles approximates a straight line function. 2. The no-load pressure will increase with increased operating speed, and also will rise with increased back pressure. 3. An increase in steam line pressure on a given turbine with a given output, speed and back pressure, will not materially benefit the steam rate of such a turbine. The only benefit results from a slight superheating of the steam through throttling. To realize full benefit from increased steam pressure, the nozzle areas should be reduced to raise ring pressure to the maximum permissible, the nozzle expansion ratios being made correct for the higher pressure. 4. The relation between ring pressure and steam flow through nozzles will closely follow a straight line function, especially at its upper points, unless back pressure is very high; it then will bend along its entire length, following a hyperbolic equation. 5. The steam flow is practically independent of operating speeds with impulse turbines. 6. It is impossible to give definite rules for relations between steam rate or lb. of steam per kw-hr., at full load and at partial loads, but, in general, the following factors will vary but little for any size or type of impulse turbine up to 500 kw.

	1/4 load	1/2 load	3/4 load	Full load
Non-condensing	40%	20%	8%	0
Condensing	20%	10%	4%	0

With throttle governing these factors state the percent by which the steam rate at partial loads exceeds the steam rate at full or most efficient load on such throttle-governed turbines. 7. The power output for a given ring pressure will be influenced to a very small extent by superheating the steam. For a turbine operating condensing there will be practically no change, but for a non-condensing turbine, superheat will increase the power output at the rate of 0.3% for 10° F. superheat for the first 150° F. of superheat. The total steam flow at a given ring pressure of a non-condensing or condensing turbine will decrease at an approximate rate of 1% for 14° F. of superheat for the first 150° F. of superheat. As a result of these two considerations, the steam rate of a non-condensing turbine will decrease about 1% for an average of 10° F. superheat, and that of a condensing turbine will decrease 1% for an average of 14° F. superheat for the first 150° F. of superheat. 8. Moisture will increase the total steam flow through a turbine for a given ring pressure by as many percent as the percentage of moisture present for qualities above 97%. It will decrease the power for a given ring pressure to about the same extent for the same qualities. That is, the steam rate will be increased at the rate of 2% for each percent of moisture for qualities

above 97%. For qualities below 97% the effect will not be quite so great. 9. For steam line pressures of 150 lb. per sq. in. gage and over, small changes in back pressure or vacuum will not affect the steam flow of impulse turbines appreciably.

For non-condensing turbines, power will decrease and steam rates will increase, on an average, by the percentages given for each pound of added back pressure up to 15 lb. gage.

Steam pressure, lb. per sq. in. gage	200	150	125	100	75
Percent decrease in power or percent increase in steam rate	2	1.5	3	4	5

Condensing turbines, designed for 28-in. vacuum, with steam pressures of about 150 to 200 lb. per sq. in. gage, will increase in steam rate at about the following percentages:

Vacuum, in. of mercury	27	26	24
Increase in steam rate	6%	11.5%	18.5%

At partial loads, the increases will be greater. 10. As stated under (3) the steam line pressure will not greatly improve the steam rate, unless ring pressure also is raised to give proper pressure drop over the governor valve. If this change is made, the effect of steam line pressure on steam rate is as follows, 150 lb. per sq. in. gage, being taken as normal.

Non-condensing.

Steam line pressure, lb. per sq. in., gage	200	175	150	125	100	75
Percent change in steam rate	−5	−3	0	+5	+10	+22

Condensing.

Steam line pressure, lb. per sq. in., gage	200	175	150	125	100	75
Percent change in steam rate	−2.5	−1.5	0	+2.5	+5	+10

It will be found that many of these general rules apply also to throttle-governed turbines of greater size than 500 kw.

PERFORMANCE OF NON-CONDENSING TURBINES at 3600 r.p.m. under 1934 plant conditions is given in Table 10 in terms of engine efficiency at the generator terminals in percent. Most efficient load is generally full rating.

Table 10.—Engine Efficiencies of Non-condensing Turbines, 3600 r.p.m.

Size, Kw.	A*	B †	C ‡	Size, Kw.	A*	B †	C ‡
500	53%	58.5%	58 –61%	5,000	....	73%	72 –73%
1000	59%	65%	63 –65%	6,000	....	74%	72.5–74%
2000	61.5%	69%	67.5–69%	8,000	....	75%	....
3000	62.2%	71.5%	70 –71.5%	10,000	....	76%	....
4000	62.5%	72.5%	71.5–72.5%				

* A. From Economic Considerations in the Application of Modern Steam Turbines to Power Generation, A. G. Christie, *Mech. Engg.*, July, 1930, to apply to average non-condensing turbines of various speeds for steam conditions up to 250 lb. per sq. in. gage, 550° F., and atmospheric pressure at exhaust.

† B. From discussion by R. T. Luce, *Mech. Engg.*, April, 1931, p. 275; gives performance of well-designed non-condensing turbines of 3600 r.p.m. for practice in 1931, in operating conditions where advantage is taken of the fact that expansion is confined almost wholly to the superheat region, and of the large quantities of steam required which permits the use of long blades. High values of the speed ratio can be used and leaving losses can be reduced to a minimum.

‡ C. Figures supplied by a leading builder of small turbines, and apply to 1934 plant conditions.

PERFORMANCE OF SMALLER CONDENSING TURBINES at 3600 r.p.m. under 1934 conditions, based upon engine efficiency at the generator terminals at most efficient load, usually at 80% of rating, is given in Table 11.

Table 11.—Engine Efficiencies of Small Condensing Turbines, 3600 r.p.m.

Size, Kw.	A*	B †	C ‡	Size, Kw.	A*	B †	C ‡
250	....	....	55.5%	5,000	73.0%	74–75%	....
500	59%	60 –63%	60.5%	6,000	73.7%	74–75.5%	....
750	62.5%	63 –66.5%	64.0%	7,000	74.2%	....	....
1000	65.0%	66 –68%	65.5%	8,000	74.8%	....	....
2000	69.0%	70 –71.5%	70.0%	9,000	75.2%	....	....
3000	71.0%	72 –73.5%	....	10,000	75.5%	....	....
4000	72.2%	73.5–74.5%	....				

* A. From Economic Considerations in the Application of Modern Steam Turbines to Power Generation, A. G. Christie, *Mech. Engg.*, July, 1930; applies to average single-cylinder, 3600 r.p.m. turbines for steam conditions up to 300 lb. per sq. in. gage, 600° F., 28.5 in. vacuum.

† B. Figures supplied by a leading builder of small turbines, and apply to 1932 plant conditions.

‡ C. Similar figures to B, from a second builder of small units for industrial plants under 1932 steam conditions.

The figures of Table 12, supplied by an American manufacturer, represent best efficiencies for multi-stage, geared, condensing turbines of the impulse type, operating with conditions that roughly approximate: 200 lb. per sq. in. gage, 150° F. superheat, 28 1/2 in. vacuum. These units are suitable for pump drive, blower drive, marine service and other

applications. Turbine speed will be selected to suit the conditions by selecting proper gear ratio. Figures include the gear loss.

Table 12.—Efficiency of Multi-stage, Geared Condensing Turbines

Rated B.Hp.	Engine Efficiency Based on B.Hp.	Rated B.Hp.	Engine Efficiency Based on B.Hp.	Rated B.Hp.	Engine Efficiency Based on B.Hp.
200	52.5	750	64.5	3,000	71.8
300	57.0	1000	66.0	5,000	73.3
400	60.5	1500	68.2	7,500	73.8
500	62.0	2000	69.8	10,000	74.0

VACUUM in all the above and subsequent data is assumed to be measured in inches of mercury, and referred to a 30-in. barometer. Thus a 29-in. vacuum is equivalent to (30 − 29) = 1 in. absolute pressure = 0.491 lb. per sq. in., absolute = 96.67% vacuum.

TESTS OF SMALL TURBO-GENERATORS.—Tables 13 and 14 are from tests of small turbo-generators such as are used in industrial plants and small central stations.

Table 13.—Test of 1000-kw. Turbine, 3600 r.p.m.

Steam Conditions, 150 lb., gage, 450° F., Exhaust 5 lb., gage

Throttle pressure, lb. per sq. in., abs.	164.6	164.6	164.6	164.6	164.6
Throttle temperature, deg. F.	451.5	467.3	464.0	474.9	476.9
Exhaust pressure, lb. per sq. in., abs.	20.0	20.0	20.0	20.0	20.0
Turbine speed, r.p.m.	3612	3613	3609	3617	3610
Total steam, lb. per hr.	9552	11,989	16,004	22,205	27,624
Kw. output	225	325	475	700	910
Steam rate, lb. per kw-hr.	42.5	36.9	33.7	31.7	30.3

Table 14.—Test of 2500-kw. Turbine, 3600 r.p.m.

Steam Conditions, 185 lb., gage, 500° F., 28 in. Vacuum

Throttle pressure, lb. per sq. in., abs.	200.6	200.6	200.6	200.6	200.6
Throttle temperature, deg. F.	502.4	501.8	503.0	499.0	502.0
Exhaust pressure, lb. per sq. in., abs.	0.98	0.98	1.08	1.23	1.47
Turbine speed, r.p.m.	3589	3604	3599	3597	3596
Total steam, lb. per hr.	12,514	14,834	22,998	29,918	36,798
Kw. output	770	990	1,605	2,100	2,560
Steam rate, lb. per kw-hr.	16.25	14.99	14.32	14.25	14.37

Tests of 6000-kw., 3600 r.p.m., Allis-Chalmers, Parsons Turbine.—The following data from a consulting engineer's official test are typical of the results which may be expected from this class of unit. Steam conditions were: 235 lb. per sq. in. gage; 522° F.; and 28 in. vacuum.

Load, kw.	3000	4500	5000	6000
Steam rate, lb. per kw-hr.	12.67	11.15	11.00	11.62

Tests on 7500-kw., 3600 r.p.m. Turbines.—Tests on two different Westinghouse turbines of 7500 kw. capacity, 3600 r.p.m., under different operating conditions are given in Tables 15 and 16.

Table 15.—Test of 7500-kw., 3600 r.p.m. Turbine with Back Pressure

Steam Conditions, 650 lb., gage, 700° F., 150 lb., gage, Back Pressure

Throttle pressure, lb. per sq. in., abs.	662.9	661.9	663.4	663.4	661.7
Throttle temperature, deg. F.	699.0	697.0	694.0	693.0	693.0
Exhaust pressure, lb. per sq. in., abs.	159.5	163.2	162.3	163.6	165.4
Exhaust temperature, deg. F.	438.0	427.0	420.0	417.0	425.0
Kw. load on generator	4217	5865	6697	6992	7589
Total steam, lb. per hr.	153,350	201,120	225,330	231,080	263,500
Steam rate, lb. per kw-hr.	36.36	34.29	33.65	33.05	34.72

Table 16.—Test of 7500-kw., 3600 r.p.m. Condensing Turbine

Steam Conditions, 185 lb., gage, 500° F., 28-in. Vacuum

Throttle Pressure, lb. per sq. in., abs.	192.7	193.7	197.3	193.4	192.8
Throttle temperature, deg. F.	500.7	504.8	517.7	519.5	509.1
Vacuum, in. of mercury	28.03	27.99	28.08	28.06	28.01
Kw. load on generator	1880	3705	5359	6903	7666
Net condensate, lb. per hr.	31,640	52,346	69,788	87,316	100,140
Steam rate, lb. per kw. hr.	19.37	14.12	13.0	12.63	13.1

Efficiencies in Table 16 may be exceeded in modern efficient turbines operating with higher steam temperatures.

Performance of 1800 r.p.m. Condensing Turbines under steam conditions up to 450 lb. per sq. in. gage, 750° F. and 29 in. vacuum is shown in Table 17. (From data in Economic Considera-

tions in Application of Modern Steam Turbines to Power Generation, A. G. Christie, *Mech. Engg.*, July, 1930.) This table presents values of the engine efficiency η_e at the generator terminals at the most efficient load for given name plate ratings of the turbines. In general most efficient load is at about 80% of rating. Thus for a turbine rated at 30,000 kw. the engine efficiency η_e = 77.2% applies to a generator output at most efficient load of 30,000 × 0.8 = 24,000 kw.

Table 17.—Performance of 1800 r.p.m. Condensing Turbines

Rating, kw.	Engine Efficiency, η_e	Rating, kw.	Engine Efficiency, η_e	Rating, kw.	Engine Efficiency, η_e
10,000	74.8	50,000	78.3	90,000	79.4
20,000	76.2	60,000	78.6	100,000	79.5
30,000	77.2	70,000	79.	150,000	80
40,000	78.0	80,000	79.2		

DISTRIBUTION OF LOSSES.—K. Baumann (Some Recent Developments in Large Steam Turbine Practice, *Jour.* Inst. Elec. Engrs., lix, p. 590, 1921), presents the following figures for the parasitic losses of an impulse steam turbine of 28,900 kw. output at the generator coupling at 1500 r.p.m.

a. Mechanical losses—constant kw. losses in per cent of turbine output at coupling.

	Loss Measured in kw. at Coupling	Loss in Percent of Output
Disc friction	315	1.09
Diaphragm leakage losses	174	0.6
High-pressure gland losses	185	0.64
Water and steam seal losses	36	0.12
Bearing losses	109	0.38
Governor and oil pump drive	16	0.05
Total mechanical losses	835	2.89

b. Velocity losses in percent of isentropic heat drop $(h_1 - h_2)$.

Steam chest losses	0.53%
Leaving losses, last blades	2.85
Exhaust losses	0.75
Total steam chest and leaving losses	3.38
Total in percent of turbine output at coupling	3.64
Total losses of turbine in percent of turbine output at coupling	6.53

These losses do not include nozzle or blade losses usually expressed as diagram or blade efficiency.

DATA ON ENGINE EFFICIENCIES will be found in Post War Turbine Development, C. D. Gibb (*Proc.* Inst. Mech. Engrs., 1931). He states the following values of the *internal engine efficiency* of turbines built by C. A. Parsons & Co.; 1000 kw., 67%; 2500 kw., 80%; 5000 kw., 82%; 10,000 kw., 84%; 25,000 kw., 85.5%. He calls particular attention to the gains made in years 1920–1930 in the efficiency of the smaller turbines. This gain is attributed to improved designs of nozzles and blading, to the use of higher values of ρ, the ratio of wheel to steam velocities, to the elimination of eddies in the steam flow by the use of coned blading, to careful design of steam chests and nozzle boxes, and to thin leaving edges on nozzles and blades. A comprehensive table of turbine performances is included in the paper.

The Brown Boveri *Review*, Jan. 1931, states that the largest single-cylinder turbine at 3000 r.p.m. built by that company develops an engine efficiency at the coupling of 81%, with 300 lb. per sq. in. gage steam pressure. Data are presented on a peak load turbine which operates over a range of load from 50% to 150% of nominal rating at an average engine efficiency at the coupling of 81%. The heat consumption varies only 6% from 50% to 150% of nominal rating.

Sweden,
before the [illegible]
at the generator terminals was 85.6%. These are among the best recorded figures and indicate a high efficiency of the turbine blading.

Alden and Balcke (Steam Turbine Practice in the U. S., *Trans.*, A.S.M.E., FSP–55–3a, 1933)

[illegible] the range of 72.5% to [illegible]

Tests of a 40,000-kw. General Electric Turbo-generator Unit.—No steam extracted from turbine. (See Turbines, N.E.L.A., 1930, Test No. 11.) 17-stage turbine, 1800 r.p.m.; generator, 13,800 volts, 3-phase, 60 cycles; exciter, 250 volts at full load, 160 kw., 1800 r.p.m., direct-connected; generator furnished with U-tube fin air cooler. The test results are given in Table 18.

Test of an 80,000-kw. Westinghouse Steam Extraction Turbo-generator Unit with steam extracted at four stages of turbine (See Turbines N.E.L.A., 1931, Test No. 2). Cross-compound unit; combination impulse and reaction type; high-pressure cylinder, single flow design, with a 2-row impulse wheel and 13 rows of Parsons blading; low-pressure turbine, double flow with 9

Table 18.—Summary of Test of 40,000-kw. General Electric Turbo-generator Unit

No steam extracted from turbine

Duration, hr.	3	3	3	3	3	3	3
Barometer (av. of 2 instruments), in. Hg	30.06	30.26	30.51	30.22	29.96	30.22	30.58
Generator							
Generator net output (direct-connected exciter, kw.	15,070	20,240	25,060	29,330	33,420	35,050	40,260
Power factor (by meter), %	81.3	75.5	83.6	85.9	94.2	94.0	96.2
Turbine—Main Unit							
Main steam pressure before throttle, lb. per sq. in., gage	366	355	359	356	353	356	358
Steam pressure after throttle, lb. per sq. in., gage	321	321	320	322	318	324	322
Steam temperature before throttle, °F.	716	709	715	707	710	717	716
Steam pressure after primary valve, lb. per sq. in., gage	147.2	194.5	238.4	273.8	316.7	314.1	312.5
Steam pressure after secondary valve, lb. per sq. in. gage	91.0	120.6	149.7	170.8	198.6	205.5	252.2
Total steam (actual steam conditions), lb. per hr.	154,980	201,810	242,960	282,520	314,620	331,130	387,010
Steam rate (actual steam conditions), lb. per kw-hr.	10.28	9.97	9.70	9.63	9.41	9.45	9.61
Heat supplied per net kw-hr. (actual), B.t.u. per kw.-hr.	13,560	13,120	12,780	12,630	12,380	12,460	12,620
Thermal efficiency (actual), %	25.2	26.0	26.7	27.0	27.6	27.4	27.0
Engine efficiency (actual), %	70.1	72.6	74.4	75.6	77.5	76.2	76.1
Condenser							
Average vacuum (30 in. bar.), in. Hg	29.02	29.01	29.0	28.94	28.95	28.99	28.83
Condensate temperature, °F.	55.3	63.1	66.7	75.1	75.2	72.3	78.0

Table 19.—Summary of Test of 80,000-kw. Westinghouse Turbo-generator Unit

Steam extracted at 4 stages of turbine. Turbine operating under vacuum

Duration, hr.	2	2	2	2	2	2
Barometer, in. of mercury	30.11	30.07	29.99	30.02	30.00	29.99
Turbine—Main Unit						
Steam pressure at throttle, lb. per sq. in., abs.	413.6	412.2	412.5	406.3	403.1	400.8
Steam temperature at throttle, °F.	689.4	681.5	691.8	688.4	681.3	679.4
Vacuum at turbine exhaust, av. (30 in. bar.), in. Hg	28.65	28.67	28.42	28.14	27.87	27.70
Total steam to turbine (actual steam conditions), lb. per hr.	230,830	277,050	509,320	735,040	852,140	970,080
No. 1 Heater						
Steam pressure at heater inlet, lb. per sq. in., abs.	53.9	64.2	115.2	171.3	200.8	227.5
Steam temperature at heater inlet, °F.	460.1	457.4	474.5	541.9	589.0	610.3
Temperature of condensed bleeder steam, assumed equal to saturation temp., °F.	285.8	297.2	338.2	369.2	382.1	392.8
Feedwater temperature entering heater, °F.	246.3	257.5	291.2	316.1	326.9	335.6
Feedwater temperature leaving heater, °F.	288.7	301.0	340.7	370.5	382.9	392.2
Terminal difference, °F.	−2.9	−3.8	−2.5	−1.3	−0.8	+0.6
Arithmetic mean temperature difference, °F.	18.3	18.0	22.3	25.9	27.2	28.9
Feedwater passing through heater, lb. per hr.	236,930	275,820	490,820	732,880	859,150	966,510
Total heat absorbed by feedwater, 1,000,000 B.t.u. per hr.	10.05	12.00	24.30	39.87	48.11	54.70
Heater area, sq. ft.	4214	4214	4214	4214	4214	4214
Coefficient of heat transfer based on arith mean temp. diff., B.t.u. per sq. ft. per hr. per °F.	130	158	259	365	420	

(Table continued on following page)

Table 19—*Continued*

No. 2 Heater						
Steam pressure at heater inlet, lb. per sq. in., abs.	28.3	33.8	59.2	86.4	101.4	116.2
Steam temperature at heater inlet, °F.	337.8	339.8	361.5	408.9	443.8	463.1
Temperature of condensed bleeder steam, °F	251.3	259.9	291.8	317.8	329.1	338.3
Feedwater temperature entering heater, °F.	187.0	195.5	224.2	240.1	247.8	254.5
Feedwater temperature leaving heater, °F.	246.3	257.5	291.2	316.1	326.9	335.6
Terminal difference, °F.	+0.8	−0.3	+0.6	+1.3	+2.0	+3.3
Arithmetic mean temperature difference, °F.	30.5	30.7	34.1	39.3	41.5	43.9
Feedwater passing through heater, lb. per hr.	236,930	275,820	490,820	732,880	859,150	966,510
Total heat absorbed by feedwater, 1,000,000 B.t.u. per hr.	14.05	17.10	32.88	55.70	67.96	78.38
Heater area, sq. ft.	4214	4214	4214	4214	4214	4214
Coefficient of heat transfer based on arith. mean temp. diff., B.t.u. per sq. ft. per hr. per °F.	109	132	229	336	389	424
No. 3 Heater						
Steam pressure at heater inlet, lb. per sq. in., abs.	8.8	11.0	19.5	27.1	31.4	35.8
Steam temperature at heater inlet, assumed, °F.	187.2	197.9	226.6	244.5	266.9	279.6
Temperature of condensed bleeder steam, °F.	187.2	197.9	226.6	244.5	252.9	260.6
Feedwater temperature entering heater, °F.	129.7	134.6	154.5	166.2	172.1	176.9
Feedwater temperature leaving heater, °F.	188.5	197.0	226.6	241.6	249.5	256.2
Terminal difference, °F.	−1.3	+0.9	0.0	+2.9	+3.4	+4.4
Arithmetic mean temperature difference, °F.	28.1	32.1	36.1	40.6	42.1	44.1
Feedwater passing through heater, lb. per hr.	236,930	275,820	490,820	732,880	859,150	966,510
Total heat absorbed by feedwater, 1,000,000 B.t.u. per hr.	13.93	17.21	35.39	55.26	66.50	76.64
Heater area, sq. ft.	4127	4127	4127	4127	4127	4127
Coefficient of heat transfer based on arith. mean temp. diff., B.t.u. per sq. ft. per hr. per °F.	120	130	238	330	383	422
No. 4 Heater						
Steam pressure at heater inlet, lb. per sq. in., abs.	2.0	2.3	3.9	5.4	6.3	7.1
Steam temperature at heater inlet, assumed equal to saturation temperature, °F.	126.1	131.2	152.0	165.5	172.1	177.5
Temperature of condensed bleeder steam, °F.	126.7	132.3	152.2	166.6	173.0	178.2
Feedwater temperature entering heater, °F.	113.8	113.3	115.5	123.2	127.8	130.7
Feedwater temperature leaving heater, °F.	125.7	131.4	151.6	164.9	171.1	175.8
Terminal difference, °F.	+0.4	−0.2	+0.4	+0.6	+1.0	+1.7
Arithmetic mean temperature difference, °F	6.4	8.9	18.4	21.4	22.7	24.3
Feedwater passing through heater, °F.	235,71	275,33	496,90	732,880	855,260	967,480
Total heat absorbed by feedwater, 1,000,000 B.t.u. per hr.	2.80	4.98	17.94	30.56	37.03	43.63
Heater area, sq. ft.	4737	4737	4737	4737	4737	4737
Coefficient of heat transfer based on arith. mean temp. diff., B.t.u. per sq. ft. per hr. per °F.	92	118	206	302	344	380
Economy						
Generator net output (corr. to operating steam conditions), kw.	19,270	24,320	46,940	68,290	77,060	86,660
Steam rate (actual steam conditions), lb. per kw-hr.	12.57	11.93	11.26	11.49	12.05	12.24
Heat consumption (actual steam conditions), B.t.u. per kw-hr.	13,800	12,890	11,770	11,630	11,990	12,040
Overall thermal efficiency (actual steam conditions), %	24.7	26.5	29.0	29.4	28.5	28.4

Parsons rows on each side. The turbine has three governor control valves arranged to carry 45,000 kw. with the primary valve, 60,000 kw. with the primary and secondary valves, and 80,000 kw. with the primary, secondary, and tertiary valves, when operated bleeding under design steam conditions. Each turbine runs at 1800 r.p.m. All bleeder heaters are of the closed type. Each turbine drives a 40,000-kw. generator, 90% power factor, 13,800 volts, 3-phase, 60 cycles. The 300-kw. exciter is direct-connected to the high-pressure turbine. Each generator has a 15,000-sq. ft. U-fin air cooler, to use condensate, together with a 6255-sq. ft. U-fin air cooler to use raw river water. The results of the tests together with bleeder heater data are given in Table 19. A surge tank is placed between No. 4 and No. 3 heaters, and this accounts for differences in the feed-water passing through No. 4 heater and through the other heaters in series.

ble 20. st D

			.p.m.	Load, kw.	nitial Abs. Pr lb. p sq. i	mp., g. F.			al ed- er np. F.	St l k Ot	Referen Notes.
		0	,000				9	lb., abs.			VDI; Apl. 10, 17
		0	,600					.14			NELA, Turbines, 25
		0	,000				0	lb., abs.			World Power Co To
		0	,000					.91			*The Engr.*, Jan. 2 926
		0	,000				1	lb., abs			VDI; Apl. 10, 1926
NB.		0	,000					.89			VDI; Aug. 28, 1926
...		0	,600					.0			*Power Plant Engg.*, Feb.
NB		0	,000					.44			World Power Conf., Tok
		0	,000					.47			*The Engr.*, June 11, 1926
		0	,600					.00			NELA, Turbines, 1930
		0	,600					.78			NELA, Turbines, 1925
		0	,600					.81			*Engg*, July 10, 1925
			,600					.55			NELA, Turbines, 1929
			,000					.06			*The Engr.*, June 11, 1926
			,000					.06			VDI; Apl. 10, 1926
	7,	0			284			.65			*Engg*, Oct. 2 , 1926
	7,	0			241			lb., gag			NELA, Tur nes, 1929
	7,	0			428			lb., abs			VDI; Jan. , 1930
	7,	0		,070	257			b., gag			NELA, Tur nes, 1929
	8,0	0		8,237	213			.32			VDI; Apl. , 1926
	0,0	0		0,970	206			.33			VDI; Apl. 10, 1926
	0,0	0		0,286	239			.97			NELA, Turbines, 1930
	0,0	0		8,228	212			.54			*Engg*, Nov. 14, 1924
	0,0	0		0,068	392			.02			*Trans.* ASME, 1934
2	1,0	0		0,975	197			.35			*Engg*, Nov. 4, 1927
2		0		1,931	217			.20			*Power*, Nov. 12, 1929
2		0		2,520	201			.89			NELA, Turbines, 1925
2		0		1,931	217			.20			World Power Conf., Tokyo, 192
2		0		0,594	202			.90		0	*Elektricitats Wirt*, June 30, 1932
		0		3,897	272			.46		0	*Power*, Jan. 27, 1925
	,000		,800	,80						0	NELA, Turbines, 1930
	,000		,800	4,07						9	*Trans.* ASME, 1934
	,000		500	1,16						1	NELA, Turbines, 1929
	,000		800	4,74						1	NELA, Turbines, 1927
	,000		800	,86						0	NELA, Turbines, 1925
			,500							0.7	*Engg*, Nov. 21, 1924
			,000							8.8	*Archiv f. Warme.* and *Dam*

Stor] IB	7	6,00	000	9			....	8.23	. ..	The Eng	15,
AC; ..	8	20,00	800	0			28	....	1, 0	Power I	g.,
CAP ..	8	20,00	000	0			80	9.94	1, 3	The Eng	15, 192
BB; ..	8	20,00	000	8			79	10.2	2, 0	The Eng	18, 192
GE; ..	0	20,00	800	0			....	9.18	. ..	NELA,	, 1930
West	0	0,000	0	21,100			....	11.60	4, 0	NELA, Turbines,	1930
GE;	n	0,000	0	20,060			....	12.03	4 0	NELA, Turbines,	1930
GE;		0,000	5	15,892			....	12.17		NELA, Turbines,	1925
West		0,000	0	16,339			....	9.83		NELA, Turbines,	1925
West		0,000	0	20,276			....	10.81		NELA, Turbines,	1925
		5,000	0	22,490			....	11.04		NELA, Turbines,	1927
		5,000	0	19,930			....	10.02		The Engr., Mar. 15,	19
		7,500	0	25,461		,	....	9.57		Engg, Feb. 12,	1926
		0,000	0	22,541		1	....	9.87		NELA, Turbines,	1925
		0,000	0	24,522		14	....	9.89		NELA, urbines,	1925
GE;		0,000	0	26,150	585	)	....	10.68		NELA, Turbines,	1927
GE;		0,000	0	25,200	688	8	10	....		NELA, Turbines,	1929
AC;		0,000	0	24,696	697	2	....	9.53		NELA, Turbines,	1929
SSW		0,000	0	18,400	761.	6	....	8.74		Mech. Engg., Nov.	1932
BTH		0,000	0	31,287	588		....	11.05		Engg., Jan. 18,	1924
AC;		0,000	0	26,394	750	9	....	9.49		NELA, urbine,	1933
AC;		0,000	0	23,097	70	14	0	....			
CAP		0,000	0	....			0	9.52		,	
MV;		5,000	0	28,382			9.6	9.117		11,	
MV;		5,000	0	28,382			0	9.12		11,	
GE;		5,000	0	32,046			....	9.77		bines,	
West; N		5,000	0	27,018			....	9.81		bines,	
AC; NB		5,000	0	34,760	12		....	9.53	..	bines,	
GE; NB		6,000	0	31,990	85		....	10.87	0		
CAP; 3S		6,000	0	....	00		4	9.44	15	g.,	
BB; 2S...		6,000	0	24,940	89		8	7.39	0	B. Rev., Jan. 1	
GE; NB..		7,800	0	28,120	18		....	9.22	..	LA, Turbines,	5
SACM; N		0,000	0	30,236	8		....	10.35		LA, Turbines,	5
West; NB		0,000	0	31,850			....	10.10		LA, Turbines,	7
West; 4S.		000	0	39,630			6	11.25		LA, urbines,	9
West; NB		00	0	36,960			....	9.71	810	NELA, Turbines,	9
GE; 4S...		000	0	32,980		02	5	10.57	120	NELA, Turbines,	0
GE; NB..		000	0	33,420		05	....	9.41	380	NELA, Turbines,	0
MV; 4S...		000	0	34,578		66	6	10.06	914	Engg., May 23,19	
West; 4S.		250	0	38,160		67	0	11.23	040	NELA, Turbines,	9
West B		00	0	39.714	634	76	....	9.87	778	LA, Turbines,	7
West B			0		633	00	....	9.99	870	LA, Turbines,	9
West					616	98	8	11.27	980	LA, Turbines,	9

Table 20.—Turbine Test Data—*Continued*

Type See notes below	Date of Test	Rating, kw.	R.p.m.	Load, kw.	Initial Conditions: Abs. Pressure, lb. per sq. in.	Initial Conditions: Temp., deg. F.	Exhaust Pressure, in. Hg, Abs.	Final Feed-water Temp. deg. F.	Lb. Steam per kw-hr. Output	B.t.u. per kw-hr. Output	Reference See notes below
GE; 4S	1929	50,000	1,800	38,368	324.6	698	1.04	318	10.23	10,842	NELA, Turbines, 1929
GE; 3S	1929	50,000	1,200	45,700	409.3	673	1.02	230	....	11,089	NELA, Turbines, 1929
GE; NB	1929	50,000	1,200	39,810	296.7	608	1.27	....	10.05	13,210	NELA, Turbines, 1929
AC; 4S	1929	50,000	1,800	43,410	634.0	677	1.21	260	9.37	10,330	NELA, Turbines, 1929
GE; NB	1932	50,000	1,800	34,200	387.7	700	1.00	....	8.82	....	NELA, Turbines, 1932
STAL; NB	1932	50,000	1,500	31,163	233.6	801	0.88	....	8.41	....	Official Tests
CAP; 3S	1930	50,000	1,500	....	615	800	1.0	280	7.4	9,280	*Engg.*, Aug. 5, 1932
CAP; 3S	1926	50,000	{1,800 / 720}	40,000	550	{750 / R700}	0.75	....	8.19	10,038	*The Engr.*, May 11, 1934
GE; NB	1930	60,000	1,500	40,150	371.7	719	1.05	....	9.06	12,010	NELA, Turbines, 1930
GE; 3S	1930	65,000	1,800	43,658	373.8	727	1.5	279	9.89	11,022	NELA, Turbines, 1930
AC; 3S	1931	65,000	1,800	48,000	646.0	{713 / R717}	0.81	....	....	9,632	NELA, Turbines, 1931
GE; NB	1930	75,000	1,800	50,350	332.7	716	1.14	....	8.98	....	NELA, Turbines, 1930
GE; 2S	1930	75,000	1,800	48,050	340.7	691	0.87	284	9.86	11,450	NELA, Turbines, 1930
West; 4S	1931	80,000	1,800	63,960	421.0	688	1.86	370	11.49	11,630	NELA, Turbines, 1931
AEG; NB	1928	80,000	1,500	51,559	485.0	768	0.53	....	8.56	....	VDI; Aug. 4, 1928
BB; NB	1930	85,000	1,500	87,858	191.0	689	1.85	....	10.10	....	VDI; June 21, 1930
West; 2S	1931	110,000	1,800	76,930	433.7	698	1.24	254	10.09	11,450	NELA, Turbines, 1931
GE; 3S	1932	110,000	1,800	81,800	1232.4	725	1.49	381	....	9,642	NELA, Turbines, 1933
AC; 4S	1932	115,000	1,800	70,944	638.0	723	1.00	333	....	9,262	NELA, Turbines, 1933
GE; NB	1932	160,000	1,500	88,100	398.2	756	1.74	....	8.75	11,360	NELA, Turbines, 1933
GE; 2S	1932	160,000	1,800	126,650	407	712	1.00	259	9.25	10,550	NELA, Turbines, 1933
GE; 2S	1932	160,000	1,800	97,300	419	685	1.22	245	9.35	10,650	NELA, Turbines, 1933
BB; NB	1931	160,000	{1,800 / 1,200}	60,832	279.7	601	0.99	....	10.10	....	NELA, Turbines, 1931
West; NB	1931	165,000	1,800	68,771	275.7	726	0.99	....	9.29	....	NELA, Turbines, 1931

Abbreviations in the table: Under "Type" is given an abbreviation for the manufacturer and the type of turbine. Thus the first unit "BB; HP" designates a Brown Boveri & Co. turbine operating high pressure. The symbols are as follows:

BB = Brown Boveri & Co. (Switzerland). AC = Allis-Chalmers Mfg. Co. STAL = Svenska Turbinenfabriks Aktiebolaget Ljungstrom, (Sweden). O = Maschinenfabrik Oerlikon, (Switzerland). EB = Erste Brunner Maschinenfabrikgesellschaft, (Czecho-Slovakia). AEG = Allgemeine Elektricitats-Gesellschaft, (Germany). GE = General Electric Co., (America). How = Jas. Howden & Co., (Scotland). LMAN = Ljungstrom turbine built by Maschinenfabrik Augsburg Nürnberg, (Germany). GMA = Gorlitzer Maschinenfabrik Actiengesellschaft, (Germany). RW = Richardson, Westgarth & Co., (England). EW = Escher Wyss & Co., (Switzerland). BTH = British Thomson-Houston Co., (England). Stork = Machinefabriek Gebr. Stork & Co., (Holland). CAP = C. A. Parsons & Co., (England). West. = Westinghouse Elec. & Mfg. Co., (America). MV = Metropolitan-Vickers Co., (England). SSW = Siemens-Schuckert-Werke, (Germany). SACM = Société Alsacienne de Construction Mécaniques. HP = High-pressure unit. NB = Condensing unit, no extraction heaters. 2S = Condensing unit with two stages for extraction heating of feedwater. 3S = Unit with three stages for extraction heating. 4S = Unit with four stages for extraction heating. R = Reheat Temperature. VDI = *Zeitschrift des Vereins Deutscher Ingenieure*. *Engg.* = *Engineering*, London. NELA = Publications of National Electric Light Association. In 1933, this association was consolidated with the Association of Edison Illuminating Companies to form the Edison Electrical Institute. *The Engr.* = *The Engineer*, London.

EFFECT OF AGE ON EFFICIENCY OF TURBINES.—Data in *Turbines*, N.E.L.A., 1932, indicate that replacements of blading and turbine overhauls tend to restore turbines to their original efficiency. It is difficult to draw any general conclusion as to effect of age over an extended period since variations in individual cases are considerable. The loss does not appear to be more than 1% per year, which may be offset by current renewals and maintenance. The last rows of blades on many units may be badly eroded by moisture, yet the efficiency is affected little. This suggests the use of rugged, well rounded blade shapes to resist erosion.

TURBINE TEST DATA from the sources noted have been gathered in Table 20. The figures quoted represent the steam rate or heat rate of the turbine at most efficient load.

15. STEAM TURBINE CALCULATIONS

Steam turbine calculations may be made for the purposes of: *a*, design; *b*, to estimate turbine performance with extraction heaters or bleeder connections; *c*, to check test results. The methods of making these calculations are varied, since the procedure is not standardized. Many assumptions are made to simplify the work. The following methods yield results satisfactory for estimating and checking purposes.

CHARACTERISTIC CURVES.—Three characteristic curves are needed to analyze turbine performance. The first of these, the *Willans Line*, curve *A*, Fig. 33, indicates the relation between total steam per hour and load. This line may be assumed for throttle governing to be straight between no load and *most efficient load*. Above most efficient load, the form of the line depends upon the overload valve arrangement, and the points at which overload steam is admitted to the turbine. In any case, the slope of the line differs from that below most efficient load. With nozzle governing, the Willans Line should consist of a series of straight lines at slightly different slopes, with steps at each valve opening. For estimating purposes a straight line up to most efficient load may be substituted for this series, or the series of stepped lines may be used if data are available for their estimation.

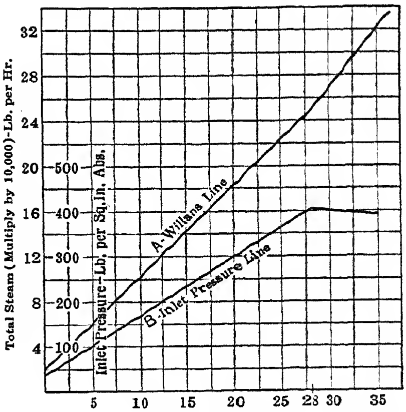


FIG. 33. Willans and Inlet Pressure Lines for 35,000-kw. Turbine

When guarantees of steam consumption, or when performances of the given turbine or of a similar unit are not available, the Willans Line may be plotted as follows: The heat drop from initial conditions to final pressure can be found. From Tables 7 to 12, or from Table 17, the engine efficiency at most efficient load, based on generator output (or B.Hp.), can be selected. At most efficient load of a turbo-generator,

$$\text{Total steam} = \frac{3412}{\text{Engine efficiency} \times \text{Total heat drop}} \times \text{Most efficient load in kw.}$$

The no-load total steam with generator at full voltage may be approximated from Table 21, which expresses no-load steam in percent of total steam at most efficient load for various name plate nominal ratings. These figures apply to single-cylinder turbines with most efficient load at 80% of the name plate ratings for units 500 kw. and above. Most efficient load in small non-condensing turbines is frequently full load.

Non-condensing No-load Steam depends largely on steam conditions. For units of 500 kw. and less, assume no-load steam, y = 20 to 25% of total steam at most efficient load.

No-load steam depends upon turbine design and may vary considerably from the values given in Table 21. For multi-cylinder turbines, the no-load factor may be greater than stated, on account of the additional bearings.

The preceding values of the no-load factor y, apply only to straight condensing turbines. Few data are available on values for regenerative and reheating turbines. Morse

(Power Plant Engineering and Design) suggests another factor ψ to find no-load steam. ψ is the product of no-load steam and isentropic heat drop from initial throttle conditions to exhaust pressure. Morse (p. 362) finds that ψ depends on the size of unit, and the values in Table 22 are taken from his data, based on recent tests.

Table 21.- No-load Steam Consumption of Turbines, Percent of Total Steam

Name Plate Rating, kw. condensing..	1000	2000	3000	4000	5000	10,000	15,000	20,000	25,000	30,000	40,000	50,000 and over
% of Total Steam at Most Efficient Load..........	13.8	12.5	11.8	11.3	10.9	9.9	9.2	8.8	8.6	8.4	8.2	8.0

Table 22.—Values of ψ

Rated capacity of unit, thousands of kw...	20	30	50	60	70	80	90	100
ψ, millions of B.t.u. per hour.............	6.5	10.4	17	18.8	19.9	20.8	21.8	22.8

When the heat drop $(h_1 - h_2)$ is known, the no-load steam becomes $w_n = \psi/(h_1 - h_2)$ lb. per hr.

The Willans Line now can be drawn between no-load and most efficient load. At full load the steam rate depends on the design of turbine and exceeds that at most efficient load. It may for estimating purposes be assumed 5% in excess of that at most efficient load. The total steam at full load then can be computed and the remainder of the total steam line can be drawn.

Example:—A 35,000-kw. turbine, 1800 r.p.m., single-cylinder, complete expansion, operates with 400 lb. per sq. in. gage, 750° F., and 29 in. vacuum. Assume most efficient load = 0.8 × 35,000 = 28,000 kw. Heat drop from 415 lb. per sq. in., abs., 750° F. to 29 in. vac., entropy, s, = 1.6586 = 1389.5 − 891.3 = 498.2 B.t.u. per lb. Engine efficiency at generator terminals at most efficient load of 28,000 kw. for turbine of 35,000 kw. rating, η_e = 77.6%. ∴ Steam rate at most efficient load = 3412/(498.2 × 0.776) = 8.82 lb. per kw-hr. output. Total steam at most efficient load = 28,000 × 8.82 = 246,960 lb. per hr. No load factor = 8.3%; no load steam = 0.083 × 246,960 = 20,497 lb. per hr. Full load steam rate = 1.05 × 8.82 = 9.26 lb. per kw-hr. ∴ Full load total steam = 9.26 × 35,000 = 324,100 lb. per hr. The Willans Line is shown in Fig. 33 as curve *A*.

The Inlet Pressure Curve is the second characteristic curve. In throttle-governed turbines it shows the relation between the pressure before the first stage and the load. It has been confirmed by test that the inlet pressure is nearly proportional to the flow of steam through the turbine. The steam supply as shown by the Willans Line is related to load. Hence inlet pressure can be plotted against load. Inlet pressure at most efficient load is less than throttle pressure by the amount of the pressure losses through throttle valve, strainer, and governor valve. This loss ranges from 3 to 5% of the absolute initial pressure. Below most efficient load, inlet pressure at any load is proportional to total steam flow. Above most efficient load, added steam passes through the throttle valve with added losses. These throttling losses above most efficient load can be assumed to vary as the square of total steam, since velocity through these areas is proportional to the total steam.

Example:—In the 35,000-kw. unit of the preceding example, let the throttle and governor drop at most efficient load = 3% of abs. initial pressure = 0.03 × 415 = 12.5 lb. per sq. in. Inlet pressure at most efficient load = 415 − 12.5 = 402.5 lb. per sq. in., abs. Inlet pressure at no-load = (20,497/246,960) × 402.5 = 33.4 lb. per sq. in., abs. Throttling and governor loss at full load = $(324{,}100/246{,}960)^2$ × 12.5 = 21.5 lb. per sq. in. ∴ Full load inlet pressure = 415 − 21.5 = 393.5 lb. per sq. in., abs. Inlet pressure line plotted on Fig. 33 as curve *B*.

With nozzle-governed turbines, the pressure before the nozzle is presumed to remain constant. However, the pressure before the second stage will vary with steam flow up to most efficient load, and can be plotted and estimated in the same way as inlet pressure, provided one knows or can assume the first-stage pressure at any given flow.

The Condition Curve is the third characteristic curve. It represents the loci of points on a Mollier diagram which approximately represent the steam condition at the entrance to each stage and the final true end point. The exact condition curve which may be needed for design purposes only can be plotted from a step-by-step calculation of the nozzle, blade, disc and leakage losses of each stage, consideration being given to carry-over velocities. For estimating purposes, and for checking performance, such an accurately drawn curve is not necessary. Experience has shown that approximated curves serve for ordinary needs, such as for estimates of station performance with regenerative heating.

In the case of a complete expansion turbine with throttling governor, it is assumed that the steam is throttled at constant total heat from the steam pressure and temperature (or quality) before the throttle valve to the inlet pressure. The inlet pressure points

therefore can be readily located upon the Mollier diagram. Fig. 35 shows these points plotted from the data on Fig. 33, assuming throttle conditions at all loads of 415 lb. per sq. in., abs., 750° F. Should temperature at the throttle decrease with load due to superheater characteristics, the new throttle condition could be used as the starting point for the throttling to inlet pressure. However, it would be necessary to plot Willans Lines and new inlet pressure lines for the changed steam conditions in order to determine the inlet pressure under the new throttle conditions.

Manufacturers usually will furnish either condition curves for given loads or data on the conditions of the steam at various bleeder and reheating points, and at exhaust from the last row of blades, as well as data on the leaving losses. This information is considered as part of the performance data on the unit. These condition points can be readily plotted on a Mollier diagram, and the condition curves drawn through them. Similarly, if test data are at hand indicating steam conditions at various points, and from which tests the true end point can be found, condition curves can be drawn through these points.

An Estimate of Performance of any turbine, needed for preliminary studies, can be made by plotting condition curves on the basis of certain assumptions. In the case of a turbine with throttling governor, the probable engine efficiency of a unit at most efficient load, if operated straight condensing, can be assumed from data in Tables 7 to 12 or Table 17. The steam rate based upon generator output can be computed from the assumed engine efficiency. The method of finding the probable true end point is outlined in Conditions at Turbine Exhaust, by Christie and Colburn (*Turbines*, N.E.L.A., 1932), from which the following paragraphs are abstracted.

The steam in expanding must furnish energy to supply the net output at generator terminals plus all generator and mechanical losses plus the energy equivalent to the leaving losses.

The manufacturer usually furnishes data upon generator efficiency. If such data are not at hand, the following approximate formulas may be used which express present-day (1933) generator efficiency at nominal or name plate rating. Generator efficiency in percent at nominal rating of unit, which rating is expressed in kw., is

For 1800 r.p.m. turbines $\eta_g = 98.85 - \{9/(\sqrt{\text{rating}/1000})\}$

For 3600 r.p.m. turbines $\eta_g = 96.0 - \{2.5/(\sqrt{\text{rating}/1000})\}$

Corrections for partial loads in percent of full load efficiencies are approximately:

Generator load, Percent of rating	100	80	75	50	25
1800 r.p.m.	100%	99.6%	99.5%	98.3%	95%
3600 r.p.m.	100%	99.3%	99.0%	96.5%	92.5%

The *coupling kw.* can be found by dividing the output by the generator efficiency.

Example:—Referring to the 35,000-kw. unit noted above, the generator efficiency at full load $\eta_g = 98.85 - (9/\sqrt{35{,}000/1000}) = 97.33\%$. At most efficient load, generator efficiency $\eta_g = 97.33 \times 0.996 = 96.94\%$. Coupling kw. = (28,000/0.9694) = 28,885 kw. at most efficient load. Coupling kw. at other loads can be found in a similar manner.

THE MECHANICAL LOSSES of a turbine consist of friction in bearing and thrust devices, power to drive governor, oil pumps, and water glands, if used, and any other mechanical losses. Radiation is of a small order, except in small auxiliary turbines, and being in the nature of heat abstracted from the steam, can, for purposes of computation, be included in the mechanical losses.

Mechanical losses depend almost entirely upon the speed of the unit, and may be considered constant at all loads, since the speed is maintained substantially constant. They are expressed as a percentage of the nominal rating of the generator, and also can be supplied by the manufacturer. When not so supplied they may be estimated by the following approximate formulas for single-cylinder units up to 100,000 kw.

1800 r.p.m. turbines,

$$\text{Loss in percent of nominal rating} = 1.5 - 1.15 \log_{10} (\text{rating}/10{,}000)$$

3600 r.p.m. turbines,

$$\text{Loss in percent of nominal rating} = 4/(\sqrt{\text{rating}/1000})$$

Mechanical losses added to coupling kw. give *internal kw.* that is, the net power that must be developed upon the shaft just inside the gland casing.

Example: The mechanical losses in the 35,000-kw., 1800 r.p.m. unit would be

$$1.5 - 1.15 \log_{10} (35{,}000/10{,}000) \times (35{,}000/100) = 306 \text{ kw.}$$

These losses are constant at all loads. ∴ Internal kw. at most efficient load = 28,885 + 306 = 29,191 kw. Internal kw. at other loads can be found by adding 306 kw. to the coupling kilowatts at that load.

The heat to internal work h_e in B.t.u. per pound of steam at a given load now can be found.

$$h_e = 3412 \times I\ \text{kw.}/w$$

where I kw = total internal kw. and w = total lb. of steam per hr. passing through the turbine.

Of the total heat h_1 that enters the throttle, h_e is all that passes out of the casing as work; the remainder goes to the condenser with the exhaust steam. The heat to exhaust per lb. of steam is $h_o = h_1 - h_e$ as shown on Fig. 34. h_o is known as the *exhaust point*.

EXAMPLE: Heat to internal work at most efficient load on 35,000 kw. turbine = $(3412 \times 29{,}191)/246{,}960 = 403.4$ B.t.u. per lb. Heat to exhaust $h_o = 1389.5 - 403.4 = 986.1$ B.t.u. per lb.

WETNESS AT EXHAUST is an important factor in leaving losses, erosion, resuperheating, and condenser design. It readily can be found from the Mollier diagram when the total heat h_o is determined.

Guy (Tendencies in Steam Turbine Development) presents curves from which the following data are taken. These enable a quick estimate to be made of exhaust conditions. Table 23 indicates the steam temperature in degrees F. at the throttle required to provide the given percent moisture at exhaust for stated internal efficiencies η_i.

Table 23.—Relation between Throttle Temperature and Pressure and Moisture in Exhaust Steam

Throttle Pressure, lb. per sq. in., gage	$\eta_i = 78\%$ Percent Moisture at Exhaust							$\eta_i = 83\%$ Percent Moisture at Exhaust						
	8	9	10	11	12	13	14	8	9	10	11	12	13	14
	Temperature at Throttle, deg. F.													
200	675	.....	.....	.....	.....	.....	.....	755	720	680	650	.....	.....	.....
300	740	710	675	.....	.....	.....	.....	840	795	755	720	680	650	.....
400	800	760	725	690	660	.....	.....	900	855	815	775	740	700	665
500	840	800	770	730	700	670	.....	950	900	860	825	785	750	715
600	875	830	800	765	730	700	670	990	940	900	860	825	785	750
800	930	890	860	825	790	760	725	.....	.....	970	925	885	850	815
1000	980	935	900	870	835	800	775	.....	.....	.....	980	935	900	865
1200	.....	980	945	910	875	845	810	.....	.....	.....	.....	980	940	910
1400	.....	.....	980	940	910	875	840	.....	.....	.....	.....	.....	980	940

THE LEAVING VELOCITY AND EXHAUST LOSS occurs in the exhaust hood between the last wheel exit and the exhaust flange to the condenser. See Leaving Velocity and Exhaust Loss in Steam Turbines, E. L. Robinson, *Trans.* A.S.M.E., FSP–56–10, July, 1934. The loss consists of both kinetic energy loss and pressure loss through the hood, since, according to the Power Test Code, vacuum is measured at the exhaust flange. This loss increases rapidly with load, and also varies with location around the wheel annulus. The heat equivalent to the loss produces no useful work and passes to the condenser as part of h_o. The energy equivalent to this loss comes from the expansion of the steam from initial conditions to exhaust pressure. The steam condition at the end of the condition curve is represented by point h_{Bo}, Fig. 34, where $h_o - h_{Bo}$ = leaving velocity and exhaust losses.

The leaving velocity and exhaust loss sometimes called simply the *leaving losses* may be expressed in B.t.u. per lb. flow to the condenser, as a percent of adiabatic heat drop, or as a percent of the total energy theoretically available for conversion to switchboard power. In practice, it varies from 4 to 36 B.t.u. per lb. flow to the condenser or from 1 to 7.5% of the adiabatic heat drop. See Turbines, N.E.L.A., 1932.

The manufacturer should furnish an estimate of leaving velocity and exhaust loss, as this can be determined from the turbine design. Robinson states that with a particular exhaust operating at fixed steam conditions, the leaving velocity and exhaust loss increases roughly as the square of the quantity of steam flowing to the condenser. With a particular exhaust passing a fixed flow, increasing the total available energy in the higher stages of the turbine by improved steam conditions, correspondingly reduces the percentage loss in the exhaust. With a fixed percentage loss in a particular exhaust, the power may be increased, by improved steam conditions, as the $^3/_2$ power of the total available energy by increasing the flow to the condenser.

When data are not available, the average annulus velocity, equivalent to the leaving velocity and exhaust loss, may be approximated by multiplying the weight flow to the condenser by the specific volume at the exhaust flange and dividing by the annulus area of the last blade row. If A_o = area of last blade annulus, sq. ft.; W = lb. of steam per sec. to condenser; v_{so} = specific volume and quality at pressure at exhaust flange, cu. ft.

per lb.; the average annulus velocity, $V_o = Wv_{so}/A_o$. The approximate leaving velocity and exhaust loss $h_L = 1.25\ (V_o/223.8)^2$. The condition of the steam at the true end-point in Fig. 34 may be assumed as $h_{Bo} = h_o - h_L$.

EXAMPLE: The leaving velocity and exhaust loss at most efficient load of the 35,000-kw. unit is 10 B.t.u. $\therefore\ h_{Bo} = 986.1 - 10 = 976.1$ B.t.u. per lb.

This gives an assumed quality leaving the last row of blades of 88.65% or 11.35% moisture. Leaving velocity and exhaust loss at full load, neglecting specific volume change, equals

$$(324{,}100/246{,}960)^2 \times 10 = 17.2 \text{ B.t.u. per lb.}$$

This method gives lower values than the actual leaving loss at loads below about 1/3 load, due to too low entering steam velocities. These calculated values may be increased arbitrarily 25 to 50% at 1/4 load and less.

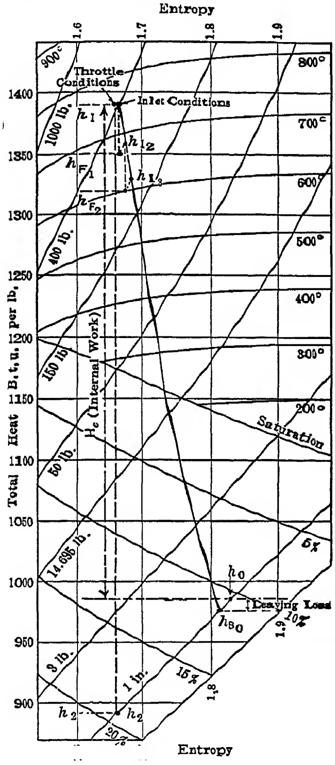


FIG. 34. Condition Curve

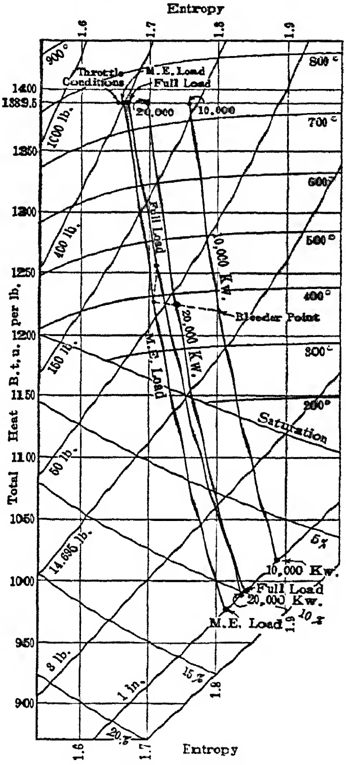


FIG. 35. Conditions Curves for Various Loads

The true end points are the lower ends of the condition curves for the various loads. The exact form of the curve between the initial point and the end point depends upon the design and efficiency of the various sections of the particular machine. For purposes of performance analysis, the curve can be approximated with sufficient accuracy on the Keenan-Mollier diagram by laying a straight edge between the two end-points and mid-way marking a point to the left of the straight edge and 3/8 in. distant. A smooth curve through the initial point, this marked mid-point and the end-point can be taken as the approximate condition curve.

In the case of a turbine with nozzle governing, the first nozzle is throttled until the load requires its full capacity, then the next nozzle is throttled, etc. As each nozzle

reaches full capacity, the initial point of the condition curve for the first stage becomes the same. The end-points for the condition curve of this first stage will vary, depending upon the efficiency and capacity of the turbine. When one nozzle is throttling the steam and one or more other nozzles are giving full delivery, there must be two expansion lines for the first stage. Thereafter a single condition curve suffices. But the turbine efficiency must be known in order to plot the complete condition curve.

To find the leaving velocity and exhaust loss on bleeder turbines, the total steam to the condenser and its condition must be known. This condition generally is not known unless data are available from the manufacturer on the relative efficiencies of various sections of the turbine.

The moisture in the exhaust steam and at the true end point can be estimated from the Mollier diagram. European designers aim to keep this moisture content less than 10%. American turbines have operated with moisture contents as high as 14%.

Table 24 presents the calculations necessary to find the true end points at a series of loads for the 35,000-kw. turbine considered in the examples given above.

These data of initial points and end points are plotted on a Mollier diagram in Fig. 35, and condition curves have been drawn for the various loads.

Table 24.—Calculation of End Points of Condition Curves

Load, kw.	Full, 35,000	Most Efficient, 28,000	20,000	10,000
1. Total steam (from Willans Line), lb. per hr.	324,100	246,960	182,260	101,380
2. Inlet pressure (from Inlet Pressure Line), lb. per sq. in., abs.	393.5	402.5	297.0	165.2
3. Generator efficiency	97.33	96.94	96.11	93.14
4. Coupling kw.	35,960	28,885	20,809	10,736
5. Mechanical losses, kw.	306	306	306	306
6. Internal kw.	36,266	29,191	21,115	11,042
7. Heat to internal work, B.t.u. per lb. of steam, (Item 6×3412)/Item 1	381.8	403.4	395.	371.6
8. Heat at throttle, B.t.u. per lb.	1389.5	1389.5	1389.	1389.5
9. Heat at exhaust point, B.t.u. per lb., (Item 8−Item 7)	1007.7	986.1	994.2	1017.9
10. Leaving loss, B.t.u. per lb.	17.2	10.0	5.5	2.0
11. True end-point, B.t.u. per lb., (Item 9−Item 10)	990.5	976.1	988.7	1015.9

THE CONDITION CURVE EFFICIENCY, η_{cc}, of a turbine with no extraction or reheating, is the ratio of the heat to work, as shown by the condition curve, to the total heat drop. In Fig. 34 the condition curve efficiency

$$\eta_{cc} = (h_1 - h_{B_O})/(h_1 - h_2)$$

THE INTERNAL EFFICIENCY, η_I of such a turbine is the ratio of the heat to work on the shaft, just inside the casing, to the total heat drop. In Fig. 34 the internal efficiency

$$\eta_I = (h_1 - h_o)/(h_1 - h_2) = \eta_{cc} - L_L$$

where L_L is the leaving loss expressed as a percent of the total heat drop.

EROSION of the inlet edges of blades by water droplets in the low pressure steam frequently has been encountered. Christie and Colburn (Turbines, N.E.L.A., 1932) show that in a pure impulse turbine, the moisture content of the steam entering the last blade row is greater than at the outlet or true end-point condition. In a Parsons turbine the reverse is the case, with moisture content highest at the true end-point. Since the last blade rows on most large impulse turbines now are given a certain degree of reaction, it generally may be assumed that the moisture content at the true end-point is the greatest that prevails in the turbine.

STAGE EFFICIENCY is the ratio of the heat to work in a stage of a turbine, measured on the shaft, to the isentropic heat drop for that particular stage. Nozzle and blade losses together with disc and idle blade loss and leakage must be considered. In general $\eta_s = \eta_n\eta_b\gamma$, where η_s = stage efficiency; η_n = nozzle efficiency; η_b = diagram efficiency; η_{cc} = condition curve of efficiency; γ = a factor to cover disc, idle blade leakage, and radiation losses. In large turbines γ is large, about 98%, for units with full peripheral admission and moderate blade speeds when labyrinth packings are tight. On small machines with partial admission, high wheel speeds and large leakage, γ is smaller. In Fig. 34 the stage efficiency $\eta_s = (h_1 - h_{I_2})/(h_1 - h_{F_1})$ for the first stage, and similarly for other stages.

It is well known from thermodynamics that the heat returned as reheat in a stage (as

for instance $(h_{I2} - h_{F1})$ in the first stage of Fig. 34) increases the available heat to produce work in succeeding stages. Hence the total heat available to produce work in the several stages, $\Sigma(h_1 - h_{F1}) = \Sigma h_r$ will exceed the total heat drop $(h_1 - h_2)$.

The ratio, $\Sigma h_r/(h_1 - h_2)$ is known as the *Reheat Factor*, *R*.

$$(h_1 - h_2) \qquad (h_1 - h_2) \qquad (h_1 -$$

where η_s = average stage efficiency.

Also $\eta_I = \eta_{cc} - L_L = \eta_s R - L_L$. Hence, with the condition curve found, it is possible to estimate the average stage efficiency, provided the reheat factor R is known. The calculation of R is difficult, since the stage efficiency in the saturation region generally is assumed to decrease 1% for each 1% of additional wetness. Kraft (The Modern Steam Turbine) suggests that the stage efficiency at any point in the saturated region can be found by multiplying the efficiency for the superheated region by the quality at the given point, and that tests have verified this assumption. Hence step-by-step methods must be used to find exact reheat factors. Table 25 has been taken from Report on Reheat Factors, E. L. Robinson (*Mech. Engg.*, Feb., 1928, pp. 1155–6). These reheat factors may be used for estimating purposes. The table is calculated for constant stage efficiency of 80% and initial superheats, 0 to 400° F. It is closely approximate for initial pressures from 20 lb. per sq. in., abs. to 700 lb. per sq. in., abs. These factors are based on Keenan's Steam Tables and Mollier Diagram. The table gives reheat factors R for various isentropic heat drops for steam expansion with an infinite number of stages.

Table 25.—Values of Reheat Factors, *R*

Isentropic Heat Drop, B.t.u. per lb.	Saturation	100° F. Superheat	200° F. Superheat	300° F. Superheat	400° F. Superheat
	Reheat Factor				
50	1.006	1.013	1.011	1.010	1.009
100	1.011	1.021	1.024	1.021	1.019
150	1.017	1.023	1.036	1.034	1.030
200	1.022	1.026	1.038	1.048	1.042
250	1.027	1.030	1.039	1.051	1.056
300	1.033	1.035	1.041	1.052	1.065
350	1.038	1.040	1.044	1.053	1.064
400	1.043	1.045	1.048	1.054	1.064
450	1.049	1.050	1.052	1.057	1.065
500	1.054	1.055	1.057	1.060	1.066
550	1.060	1.061	1.062	1.064	1.068

$$\text{Reheat Factor} = \frac{\text{Actual Heat to Work on Condition Curve}}{\text{Average Stage Efficiency} \times \text{Isentropic Heat Drop}} = \frac{}{\eta_s (h_1 - h_2)}$$

With any particular number of stages the heat recovery $(R - 1)$ is decreased by a fraction nearly equal to the reciprocal of the number of stages, *i.e.*, by a fraction $(n - 1)/n$, where n = number of stages. Thus with 500 B.t.u. adiabatic heat drop and 300° F. superheat, with 80% stage efficiency, the reheat factor for a 17-stage machine would be $R_{17} = 1 + \{16\,(R - 1)/17\} = 1 + \{16 \times 0.060/17\} = 1.056$. Also with any other stage efficiency, the heat recovery is greater or less, nearly in direct proportion, as the stage loss is greater or less. If the stage efficiency were 70% instead of 80% the reheat factor with an infinite number of stages with 500 B.t.u. adiabatic heat drop and 300° F. superheat is $R_{70\%} = 1 + \left(\frac{- 0.70}{- 0.80}\right)(R_{80\%} - 1) = 1 + \frac{0.3}{0.2} \times 0.060 = 1.090$. On a 17-stage machine this becomes 1.085.

CUMULATIVE HEAT.—In a multi-stage turbine the sum of the isentropic heat drops Σh_r in the individual stages is greater than the adiabatic heat drop $(h_1 - h_2)$, from initial conditions to final pressure, by the reheat factor R. That is, $\Sigma h_r = R(h_1 - h_2)$. This is called the cumulative heat. In steam turbine design, it is often necessary to determine the heat actually available for work at any point on the condition curve. When the stage efficiency is known and H_s is the total heat to work from initial conditions to the given point, then cumulative heat $H_c = H_s/\eta_s$. This cumulative heat then is distributed among the various stages and leads to a proper design. See Goudie Steam Turbines, p. 581, for methods of distributing cumulative heat and for determining specific volumes at the various points.

Extraction Calculations

Most large turbines now are provided with extraction points from which steam is bled for regenerative feedwater heaters. These heaters vary from one to five in number. If economizers are used, only one or two feed heaters are added. If air preheaters only are installed, usually three or more heaters are used.

When steam is thus extracted at intermediate stages from a turbine, more steam must enter the throttle for a given generator output than if the turbine were operating straight condensing. Reynolds (Factors Affecting the Gain from Feedwater Heating, *Elec. Jour.*, June, 1929) gives in Table 26 such increases, expressed as percent of throttle flow for the same output for non-extraction operation, due to stage extraction for feedwater heating, and steam extracted for feedwater heating as a percent of total throttle flow when operating with extraction. Steam conditions: throttle pressure, 300 lb. per sq. in., gage; throttle temperature 622° F.; vacuum, 29 in. Hg.

Table 26.—Increase in Steam Flow and Steam Required for Extraction Heating
Expressed as Percentage of Throttle Flow for Non-extraction Operation

Final Feed-water Temp., ° F.	Extraction Stages					Extraction Stages				
	1	2	3	4	5	1	2	3	4	5
	% Increase in Steam Flow					Steam Extracted—% of Throttle Flow				
180	4.0	3.5	....	....	....	9.6	9.9	....	....	....
200	5.5	4.5	4.2	....	....	11.3	11.6	11.9	....	....
220	7.5	6.0	5.5	5.0	....	13.0	13.2	13.5	13.8	....
240	9.0	7.5	6.5	6.0	5.5	14.6	14.9	15.2	15.5	15.7
260	11.0	9.2	8.0	7.5	7.0	16.0	16.4	16.8	17.1	17.3
280	....	11.0	10.0	9.2	8.5	....	18.0	18.3	18.6	18.9
300	....	13.2	11.7	11.0	10.0	....	19.3	19.8	20.1	20.4
320	....	....	13.5	13.0	12.0	....	....	21.4	21.7	22.1
340	....	....	17.0	15.5	14.5	....	....	22.8	23.2	23.6
360	....	....	....	18.5	17.0	....	....	....	24.7	25.1
380	....	....	....	21.5	20.0	....	....	....	26.0	26.5
400	....	....	....	25.0	23.5	....	....	....	27.6	28.1

The percent increase in steam flow will be slightly lower with higher steam pressures and slightly higher with lower steam pressures. For estimating purposes, the values in Table 26 may be multiplied by 1.05 for 200 lb., by 0.95 for 400 lb., by 0.915 for 600 lb., and 0.88 for 1200 lb.; all pressures in lb. per sq. in. gage. Similar corrections apply to the percentage of steam extracted. The latter data may be used to estimate steam to condenser, the leaving losses, and condenser capacity.

The condition curve is modified by this additional throttle steam, as increased steam flow through the high-pressure stages leads to increased efficiency in these sections. Also decreased flow to the condenser results in lesser leaving loss. The general result of bleeding a turbine for regenerative reheating is to increase its overall efficiency, provided the stages are designed for operation under extraction conditions.

To plot exact condition curves, manufacturers can furnish data on the pressures, temperatures, or quality, and heat contents at the various extraction points and at exhaust or true end-point. With leaving losses also given, the condition curves at the various loads can be plotted. Lacking such data, an approximation to the condition curves can be made, as outlined in preceding sections, by assuming the efficiency of a similar size of turbine operating under the same steam and exhaust conditions.

Performance data generally are expressed in terms of final feedwater temperature leaving the last heater. The choice of this temperature in the design of a plant depends upon the number of heaters and the personal judgment of the engineer. In general, at most efficient load this is about 70% to 80% of the saturation temperature at boiler pressure when no economizer is used.

Hendrickson and Vesselowsky (A Thermal Study of Available Steam-power-plant Heat Cycles, *Trans.* A.S.M.E., FSP–56–4, April, 1934) present thermodynamic formulas and methods for computing the performances of 30,000-kw. units on various cycles and with different pressures and temperatures at throttle. Their results are shown by curves from which deductions on performance can be drawn.

R. L. Reynolds (Calculation of the Gain from Feedwater Heating, *Elec. Jour.*, April, May, June, 1929) presents simple methods of calculation and many data on performance. He has recomputed these data on the basis of Keenan's Steam Tables and Mollier Diagram, also considering more recent power plant developments. Tables 27 to 32, which will prove useful in making plant estimates, are taken from the Reynolds' curves.

Table 27 shows the final feedwater temperature from the last heater, which gives

maximum reduction in heat consumption with multi-stage extraction. Steam conditions: Throttle pressure, 200 to 1200 lb. per sq. in., gage; throttle temperature, 750° F.; vacuum, 29 in. These temperatures vary only a few degrees with changes in throttle temperature.

Table 27.—Final Feedwater Temperatures with Extraction Heating

Steam Pressure, lb. per sq. in., gage	Number of Stages of Extraction							
	1	2	3	4	5	6	8	10
	Deg. F.	Deg. F.	Deg. F.	Deg. F.	Deg. F.	Deg. F.	Deg. F.	Deg. F.
200	222	260	286	305	318	327	339	344
300	233	273	300	320	333	342	355	361
400	243	285	312	332	346	357	370	377
500	252	295	324	344	358	369	384	392
600	260	307	335	354	369	380	396	406
700	268	316	345	364	379	391	407	419
800	275	326	355	374	390	402	419	432
900	282	335	364	384	400	412	430	445
1000	289	344	373	393	409	422	442	457
1100	295	353	382	402	419	432	452	469
1200	302	361	390	411	428	442	463	481

The reduction in heat consumption with multi-stage extraction, in percent of that with no extraction, and with the feed temperatures of Table 27, are given in Table 28, using the same steam conditions as given for Table 27.

Table 28.—Reduction of Heat Consumption with Extraction Heating

Expressed as a percentage of the heat consumption with non-ext

Steam Pressure, lb. per sq. in., gage	Number of Stages of Extraction							
	1	2	3	4	5	6	8	10
	Percent	Percent	Percent	Percent	Percent	Percent	Percent	Percent
200	5.15	6.80	7.70	8.35	8.85	9.20	9.70	9.90
300	5.60	7.35	8.35	9.05	9.55	9.95	10.50	10.70
400	5.90	7.80	8.90	9.65	10.25	10.65	11.20	11.45
500	6.25	8.25	9.40	10.20	10.80	11.20	11.80	12.10
600	6.55	8.65	9.85	10.65	11.30	11.75	12.35	12.65
700	6.70	9.00	10.25	11.10	11.75	12.20	12.85	13.15
800	6.90	9.35	10.60	11.50	12.15	12.60	13.30	13.65
900	7.25	9.65	10.95	11.85	12.50	13.00	13.70	14.05
1000	7.45	9.90	11.25	12.10	12.80	13.30	14.00	14.40
1100	7.60	10.15	11.50	12.30	13.05	13.60	14.35	14.80
1200	7.75	10.30	11.75	12.65	13.30	13.85	14.60	15.10

The effect of a rise in steam temperature is to decrease these gains slightly; a decrease in steam temperature increases the gain slightly. For estimating purposes, deduct 0.2% for each 100° F. rise above 750° F., and add 0.23% for each 100° F. decrease below 750° F. The percent reduction in heat consumption does not vary much for a range of feed temperature on either side of those given in Table 27. This is shown by the values in Table 29. In practice the tendency is to choose feed temperatures below the maximum rather than above.

Table 29 indicates the reduction in heat consumption due to feedwater heating to various final temperatures expressed as a percent of non-extraction performance.

Table 29.—Reduction in Heat Consumption Due to Feedwater Heating

Expressed as a percentage of non-extraction operation. Steam conditions: Throttle pressure, 400 lb. per sq. in., gage; throttle temperature, 750° F.; vacuum, 29 in.

Final Feed Temperature, deg. F.	Number of Stages of Extraction				Final Feed Temperature, deg. F.	Number of Stages of Extraction			
	1	2	4	10		1	2	4	10
	Percent	Percent	Percent	Percent		Percent	Percent	Percent	Percent
180	5.20	5.65	5.90	6.30	320	5.05	7.65	9.65	11.00
200	5.60	6.40	6.80	7.25	340	....	7.45	9.65	11.25
220	5.85	7.00	7.55	8.10	360	....	7.10	9.55	11.40
240	6.00	7.45	8.25	8.85	380	....	....	9.35	11.45
260	5.95	7.70	8.80	9.55	400	....	....	9.00	11.35
280	5.75	7.85	9.20	10.10	420	....	....	....	11.20
300	5.45	7.80	9.50	10.60	440	....	....	....	10.95

Since stage pressures increase or decrease with the amounts of steam passing through the stage, the saturation temperature of the steam at the extraction points, and consequently the final feedwater temperature, will increase or decrease with load. The aver-

age large turbine has its feed heating system designed to give best performance at a most efficient load of 75 to 80% of rating. Reynolds gives in Table 30 the figures for a unit designed for most efficient load at 75% rating. Table 30 shows the influence of load on the reduction in heat consumption as compared with non-extraction conditions with 4-stage feed heating. Steam conditions are the same as given for Table 29.

Table 30.—Influence of Load on Reduction of Heat Consumption, 4-stage Heating

Expressed as a percentage of the heat consumption with non-extraction operation

Load on turbine, percent of rating	30	40	50	60	70	75	80	90	100
Percent reduction in heat consumption	4.85	6.35	7.60	8.55	9.30	9.65	9.85	10.25	10.55

Since the heat consumption in non-extraction operation increases above most efficient load, operation with extraction heaters as indicated by the above gains will lead to heat consumptions at full loads, relatively less in proportion to most efficient load than in the case of non-extraction performance. This leads to a flatter heat consumption curve.

Reynolds shows that a decrease in vacuum from 29 in. to 27 in. decreases the gain from extraction operation, but increases the most favorable final feed temperature. Table 31 shows the influence of vacuum on the reduction in heat consumption as compared to non-extraction operation when using 4-stage feed heating. Steam conditions, same as given for Table 29.

Table 31.—Influence of Vacuum on Reduction of Heat Consumption with 4-stage Heating

Expressed as a percentage of the heat consumption with non-extraction operation

Final Feed-water Temp., deg. F.	Vacuum 27 in.	28 in.	29 in.	29.5 in.	Final Feed-water Temp., deg. F.	Vacuum 27 in.	28 in.	29 in.	29.5 in.
	Percent	Percent	Percent	Percent		Percent	Percent	Percent	Percent
260	7.60	8.10	8.75	9.45	340	8.85	9.35	9.65	10.05
280	8.10	8.65	9.20	9.80	346	8.85	....	....	....
300	8.45	9.00	9.50	10.00	360	8.80	9.20	9.55	9.90
320	8.70	9.20	9.65	10.10	380	8.70	9.05	9.30	9.60
324	....	....	....	10.13	400	8.45	8.75	8.95	9.20
332	....	....	9.70	....					

SELECTION OF HEATERS.—The economic selection of the proper number of heaters requires an evaluation of the savings, giving consideration to average use factor and average load throughout the useful life of the turbine, the cost of fuel, and whether or not economizers are used. Against this are charged the fixed costs on the investment in the added heater, valves, and piping, repairs and maintenance on this equipment, and the cost of added power for pumping due to the friction head added by the heater. In many cases this balance will indicate that the savings from adding more than 3 or 4 heaters are not warranted by the returns on this investment.

When the final feed temperature from the last heater is fixed, the temperatures leaving other heaters can be selected. The easiest method is to divide the temperature rise equally between heaters. But in actual cases, the feed heating system is complicated by drainage from previous heaters, by evaporators and their condensers, by steam jet air pumps, and by the heat from the boiler feed pump. These modify the amount of steam bled at the various heaters, and particularly tend to decrease that bled at No. 1 heater. Reynolds shows a curve from which Table 32 is taken to indicate that best performance is obtained when equal quantities of steam are bled at all heaters. Table 32 shows the effect of location of heaters on heat consumption for 4-stage extraction by indicating the quantity bled at various heaters.

Table 32.—Effect of Location of Extraction Heaters on Heat Consumption

Steam conditions: Throttle pressure, 300 lb. per sq. in., gage; throttle temperature, 622° F.; vacuum, 29 in.

Stage of Extraction	Heat Consumption, B.t.u. per kw-hr. 11,962	11,944	11,925	11,953
	Steam Bled, lb. per hr.			
4	15,560	13,520	11,100	7,980
3	12,440	12,300	11,100	10,820
2	9,250	11,180	11,100	10,530
1	7,530	7,750	11,100	15,360

There is no rule to determine the distribution of bleeder points and trial and error must be employed. Several approximate methods are in use.

If generator air coolers, oil coolers, heat exchangers on drips, and the exhaust of steam

jet air pumps, all are used to heat the feed, the temperature entering the first heater will be considerably higher than hotwell temperature. It may be assumed for preliminary calculations at most efficient load and 29 in. vacuum to be about 100° F. Having chosen the number of heaters and the desired final feed temperature, the temperature leaving each heater can be fixed, or if equal quantities of steam are bled, the feed temperature is found by calculation.

The terminal differences in feed heaters varies from 5° to 10° F., depending on the amount of heater surface, and usually is 6° F. With this fixed, the saturated steam temperature can be determined and hence the saturated steam pressure in the heater. Some feed heaters for the high-pressure bleeder points, where superheated steam is bled from the turbine, have been furnished with a counter-current section to utilize the superheat, which in some cases has heated the feedwater to the saturation temperature of the steam in the heater or even higher.

A Pressure Drop occurs in the piping, non-return valves, and gate valves between turbine casing and heater, and in the turbine between the blading and the extraction nozzle. This frequently has been assumed to be 10% of the absolute pressure at the bleeder point at all loads. An analysis of N.E.L.A. test data indicates that pressure drops, shown in Table 33, between extraction points on the condition curve and the heaters may be assumed at most efficient load.

Table 33.—Pressure Drop between Extraction Points and Heater

Pressure at extraction point on condition curves, lb. per sq. in., abs.		10	20	30	40	50		70			100	120	140	160
Percent of pressure drop	15.6	14.5	12.8	11.6	10.6	9.7	9.0	8.1	7.5	7.0	6.4	5.8	5.4	5.

The major portion of the pressure drop is caused by the non-return valve, and this becomes a factor of decreasing importance with increase in pressure.

At other loads the pressure drop from the extraction points on the condition curve to the heaters, can be found by the formula $p = K w^2 v$, where p = pressure drop, lb. per sq. in.; w = steam flow, lb. per hr.; v = specific volume at extraction point on condition curve, cu. ft. per lb.; K = a constant determined by applying the above formula to the assumed pressure drop, flow and specific volume at most efficient load. Such computations are somewhat involved, and for estimating purposes, approximate pressure drops in percent can be estimated at other than most efficient loads as proportional to the total steam flow through the throttle. Thus in the preceding example, if the pressure at a given extraction point on the condition curve at most efficient load is 60 lb. per sq. in., the pressure drop to extraction point, from Table 33, is $60 \times 0.09 = 5.4$ lb. per sq. in., when 246,960 lb. per hr. are flowing through the throttle. At full load, the throttle flow is 324,100 lb. per hr. The pressure on the condition curve at the extraction point would be $60 \times 324{,}100/246{,}960 = 78.7$ lb. per sq. in. The approximate pressure drop to the heaters, in percent, would be $9 \times 324{,}100/246{,}960 = 11.8\%$. For preliminary calculations the pressure drop at full load may be taken as $78.7 \times 0.118 = 9.3$ lb. per sq. in.

Reynolds (Calculation of the Gain from Feedwater Heating, *Elec. Jour.*, April, 1919) uses a different approximate method by allowing for a pressure drop between extraction point and heater, which he assumes as equivalent to 2° F. drop in saturation temperature.

The pressure of the steam in the turbine casing now can be computed, and when plotted on the condition curve, the total heat in the steam at the extraction point can be found.

The pressures at other loads at the high-pressure extraction point are proportional to the total steam flows at the throttle, and for estimating purposes may be assumed in the same proportion at the other bleeding points. The true pressures at bleeder points below the high-pressure heater are proportional to the actual flows at each point, but these cannot be determined until later computations have been made. Slight errors in extraction point pressures have comparatively little effect upon the final unit performance.

The location of these pressures on the respective condition curves indicates the heat content of the steam at the various bleeder points. The computations now can be reversed and the temperatures of feed leaving each heater at each load can be found.

Table 34 shows these calculations for the 35,000-kw. turbine assumed above and similar calculations for other loads. It also shows the method of finding the actual pressure in the heater. These pressures on the condition curve are marked on Fig. 35.

Standard practice numbers the heaters from the exhaust. Thus No. 1 heater is the first one that the condensate enters.

The heaters may be drained: *a*, Through traps to the succeeding heater in *Cascade*, and finally to the condenser; *b*, by a return pump to the feed system at each heater; or *c*, by cascading to No. 2 heater, which is often made a deaerating heater, and mixing with the feed; the drains from No. 1 heater pass through a heat exchanger to the condenser. H. L. Guy (Tendencies in Steam Turbine Development, *Proc.* Inst. Mech.

Engrs., 1931) shows that (*b*) will decrease heat consumption over (*a*) with 600 lb., 700° F. steam conditions by about 0.87%; while (*c*) betters (*a*) by 0.7%. With 350 lb., 700° F. steam conditions, the gains of (*b*) and (*c*) over (*a*) are 0.97% and 0.69%, respectively, for the conditions assumed. System (*b*) involves many small drip pumps, often of low capacity, and at the high-pressure heaters these must pump against full boiler feed pressure. These added auxiliaries decrease the reliability and availability factors of the turbine. Scheme (*c*) or a modification of this arrangement is generally preferred.

Table 34.—Steam Conditions in 35,000-kw. Turbine

Load, kw.	Full, 35,000	Most Efficient, 28,000	20,000	10,000
Total Steam (from Willans Line), lb. per hr.	324,100	246,960	182,260	101,380
Pressure on condition curve at extraction point, lb. per sq. in., abs.	78.7	60	44.2	24.6
Percent pressure drop to heater	11.8	9	6.6	3.7
Pressure drop to heater, lb. per sq. in.	9.3	5.4	2.9	0.9
Pressure in heater, lb. per sq. in., abs	69.4	54.6	41.3	23.7

The Calculation of the Steam Extracted at each bleeder point is a heat balance calculation. Thus assume, for example, No. 3 heater of a series of four heaters.

$$b\,(h_3 - m_3 - h_{f3}) + a\,(h_{f4} - h_{f3}) = (x + w)\,(h_o - h_i)$$

Solve for b = lb. of steam per hr. to be bled, where h_3 = total heat at bleeder point on the condition curve; m_3 = radiation loss, B.t.u. per lb. of bled steam (usually 3 to 5 B.t.u.); h_{f3} = heat of saturated liquid at bleeder heater pressure; a = lb. of steam per hr. cascaded from previous heater; h_{f4} = heat of saturated liquid at pressure of preceding heater; x = make-up feedwater, lb. per hr.; w = lb. of steam per hr. entering turbine; h_o= heat of feedwater at temperature leaving No. 3 heater; h_i = heat of feedwater at temperature entering No. 3 heater; all heat quantities are in B.t.u. per lb.

In the above calculation the temperature of the drips was assumed to be saturation temperature at that heater pressure. Some counter-current heaters reduce this temperature considerably below saturation temperature and these lower temperatures can be used in the heat balance equations.

Where pumps lift the drip back into the feed line, w must be correspondingly decreased on the heater from which the drips are pumped, as well as on lower-temperature heaters.

Heat Transfer Rates in Bleeder Heaters depend upon the velocity of the water through the tubes. As shown by Yellott's statement (Turbines, N.E.L.A., 1932) heat transfer rates in a given heater may vary from 400 B.t.u. at 2 ft. per sec. water velocity; 600 B.t.u. at 4 ft. per sec., to 750 B.t.u. at 6 ft. per sec. High water velocities permit the use of less heater surface. Regarding the effect of average film temperatures, if the heat transfer rate at 4 ft. per sec. velocity is taken as 400 B.t.u. at 100° F. average film temperature, the rate should be 500 B.t.u. at 150° F., 600 B.t.u. at 200° F., and 700 B.t.u. at 250° F. Usual water velocities vary from 4 to 6 ft. per sec. The pressure drop on the feedwater side of the heater varies from 4 to 12 lb. per sq. in. per heater.

Bleeder heaters with an outlet temperature up to 250° F. generally are installed on the suction side of the boiler-feed pump. The condensate pump is selected to deliver the condensate through these heaters at a suitable pressure to the suction of the boiler-feed pump. Non-return valves in the bleeder steam connections prevent flooding the turbine if a tube ruptures in one of the closed heaters.

Frequently, one of the feedwater heaters serves as a *deaerating heater* to remove oxygen and gases from the feedwater. In case this is an open mixing heater, the temperature of the condensate leaving this heater will be the saturation temperature corresponding to the heater pressure. When boiler pressures do not exceed 600 lb., the boiler-feed pump removes the water from No. 2 when this is an open heater. Heaters beyond this point must be designed on the water side to withstand full boiler pressure. With 1500 lb. boiler pressure, a booster pump is placed at No. 2 heater and the boiler-feed pump may be placed beyond the final heater.

Allowance must be made for the heat added to the feedwater from the losses in centrifugal condensate, booster, and boiler-feed pumps. It may be assumed that 10% of the losses disappear as radiation and 90% is carried away by the feedwater. On this assumption, the temperature rise in deg. F. due to pump losses is

$$\left(\frac{1}{\eta_p} - 1\right)$$

where η_p = pump efficiency as a decimal; P = total pressure added by pump, lb. per

sq. in.; d = specific volume of feedwater entering pump, cu. ft. per lb.; w_1 = total feedwater pumped, lb. per hr.

The Efficiency of Centrifugal Condensate Pumps may vary from 25 to 60%, depending upon working conditions. Centrifugal boiler-feed and booster pumps have efficiencies, at full load, of 50 to 75% depending upon pressure and temperatures. These efficiencies decrease at partial loads.

The Evaporator to provide distilled water for boiler-feed make-up usually forms an element in the extraction system. A single-stage evaporator generally is used, taking steam from No. 2 or No. 3 extraction point, and exhausting into an evaporator condenser placed on the feed line between the preceding heater and the heater at the extraction point from which steam is taken. Aside from radiation losses, the evaporator and its condenser merely serve as the first stage of feed heating at the extraction point from which steam is taken to the evaporator. (See pp. 3–36 to 3–48 for evaporator data and calculation.) Make-up in a modern station varies from 0.5 to 2% of the total steam to the turbine. In industrial plants, it may be much higher and may even exceed possible evaporator capacity when all condensate does not return from the bleeder services.

Heat transfer in evaporator coils depends both upon temperature head and on the vapor temperature and degree of purity of the water evaporated. Thus with 25° F. temperature head, heat transfer rates of 425 B.t.u. at 100° F. vapor temperature, 500

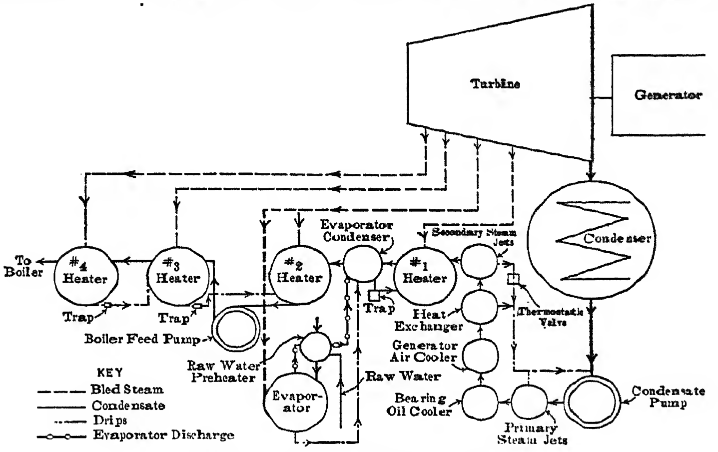


Fig. 36. Typical Extraction Plant Layout.

B.t.u. at 200° F. and 550 B.t.u. at 300° F. might be expected. Generally the total heating surfaces in all stages of multiple effect evaporators are made the same. See Yellott's statement in Turbines, N.E.L.A., 1932.

In some plants, steam is withdrawn at bleeder points to furnish station heating, to operate steam-jet air removal pumps on the condenser, to operate steam-driven booster pumps, etc. These cases can be computed by giving careful consideration to all heat quantities involved.

Generally, the feed leaving the condensate pump passes first through the condenser for the primary steam jets of the air pump. Knowing the total steam required by the jets and the total heat in this steam, the temperature rise of the feedwater can be computed.

The feed next passes through the *generator air cooler* where the use of clean condensate is desirable, as it eliminates the need of cleaning and will not corrode tubes. Few data have been published to show the relation which the heat recovered in the generator air cooler bears to the heat equivalent of the total electrical losses. Schoenherr's statement in Turbines, N.E.L.A., 1932, indicates that at 29.5 in. vacuum, 92.5% of the electrical losses are recovered by the feedwater; at 29 in. vacuum, 84.5%; at 28.5 in. vacuum, 76%. Passing the condensate through the generator air cooler reduces the steam bled at the first bleeder point, and results in a final saving in heat rate of unit of 0.25 to 0.5%.

The feedwater next passes through the *oil cooler*. Data indicate that heat equivalent

to the total mechanical losses is absorbed by the feedwater in this heater. Undoubtedly some of the missing heat in the generator losses and heat from the turbine itself passes by conduction to the bearings and is removed by the oil.

If a *heat exchanger* is used on drips, this can be calculated by assuming a terminal difference of 10° F. between the cooled drips and entering condensate, and calculating the resultant temperature rise of the condensate.

The feed finally passes through the condenser of the secondary air jets of the vacuum pump, where temperature rise can be calculated as for the primary jets.

At light loads, there may be insufficient condensate to properly cool the generator air. In such a case a thermostatically controlled valve may by-pass feedwater beyond the oil cooler to the condenser, where it flashes and thereby increases the flow to the generator air cooler to obtain the desired generator air temperature. Fig. 36 shows a typical extraction layout. Further information upon the computation of the heat balance on extraction turbines can be found in the following:

Brown and Drewry, Economy Characteristics of Stage Feedwater Heating by Extraction, *Trans.* A.S.M.E., xlv, p. 713, 1923; Harding, Steam Power Plant Engineering; Hyde and Guigon, A Method for Calculating Central Station Heat Balance, *Power*, July 3, 1928; Reynolds, Calculation of the Gain from Feedwater Heating, *Elec. Jour.*, April, 1919; Wheeler, A Typical Power Plant Heat Balance Layout, *Powerfax*, Autumn, 1931. A Thermal Study of Available Steam-Power Plant Heat Cycles, Hendrickson and Vesselowsky, *Trans.* A.S.M.E., FSP–56–4, April, 1934.

OPERATION OF EXTRACTION TURBINES.—Extraction turbines are started, operated, and shut down with all bleeder heaters connected in service. The connection to the turbine usually contains a gate valve and a non-return check valve. Some turbines have been designed so that at the time of peak load, bleeder heaters may be shut off and all of the steam passed to the exhaust, increasing the leaving loss and decreasing the plant efficiency by feeding colder water to the boilers. Christie and Viessman (Low Cost Peak-Load Capacity Bleeder Turbines, *Mech. Engg.*, Feb., 1931) show that the use of hot water accumulators to provide feed to the boilers over the peak when the bleeder heaters are cut off, leads to increases in station capacity of about 10% with 3 bleeders, 17% with four bleeders, and 21% with 5 bleeders. These gains are obtained with no increase in boiler or furnace capacity over those when the bleeders are in service. Station economy is also maintained nearly constant. These installations require a low investment, are simple and reliable in operation, and are economical in fuel consumption.

16. REHEATING TURBINES

REHEATING the steam from the high-pressure section of a turbine before admitting it to the low-pressure section has two objects: 1. To decrease moisture at exhaust; 2. To increase overall plant efficiency. Warren and Blowney (The Increase in Thermal Efficiency Due to Resuperheating, *Trans.* A.S.M.E., xlvi, p. 563, 1924) show that the stage efficiency of the saturated section of a turbine decreases 1.15% for each additional 1% of moisture in the steam. Reheating provides more expansion in the superheat region with corresponding gains in efficiency. Reheating is done with live steam, with boiler flue gases, or with a combination of these two methods. Station economy is highest when reheating is done by boiler flue gases. The need of reheating becomes less as permissible initial steam temperatures increase. For instance with 700° F. throttle temperature, reheating is necessary at all pressures above 500 lb. per sq. in., gage, to maintain moisture at exhaust at 12%. With 850° F., reheating is not necessary until above 700 lb. per sq. in. pressure is reached. With 1000° F. and 85% engine efficiency, reheating is unnecessary below 1250 lb. per sq. in. gage pressure.

Reynolds has shown (Reheating, *Southern Power Jour.*, 1932) that the efficiency of a turbine has little effect on the gain to be expected from reheating. The number of stages

Table 35.—Reduction in Heat Consumption Due to Reheating

Steam Conditions: 1200 lb. per sq. in., gage; 750° F. inlet temp.; 750° F. reheat temp.; 29-in. vacuum.

Operating Condition	Pressure entering reheater which produces maximum reduction, lb. per sq. in., abs.	Reduction in heat consumption, percent
Straight condensing	170	6.75
1-Stage feed heating	195	5.65
2-Stage feed heating	210	4.95
3-Stage feed heating	215	4.65
4-Stage feed heating	220	4.55
5-Stage feed heating	225	4.45

of extraction does affect the gains. Table 35 shows the reduction in heat consumption due to reheating.

The Pressure Drop over the Reheater is assumed as 5% of the absolute pressure leaving the high-pressure section. Table 35 indicates that 4- or 5-stage extraction reduces the gain about $^1/_3$, but this does not include the gain due to regenerative feed heating. The gain from both reheating and regenerative feed heating may be taken as 95% of the sum of the individual gains. This maximum gain is obtainable only over a narrow range of reheating pressure. For example, with 4-stage bleeding, the gain is 4.5% or better with reheat pressures of 175 to 225 lb. per sq. in., abs. Usually the pressure entering the low-pressure section varies with load. The efficiency of the low-pressure section is a maximum at only one load and reheating pressure, and this load is the most efficient load for the complete unit. Hence for loads above or below most efficient load, the improvement due to reheat will be lessened. While further gains might be obtained by additional stages of reheating, these add to the expense of the plant and involve operating complications. More than one stage of reheating is not warranted with pressures of 2000 lb. per sq. in., abs., or less, and 850° F. at throttle.

THE GAIN FROM REHEATING depends upon the temperature to which the steam is reheated. See Table 36, by Reynolds.

Table 36.—Reduction of Heat Consumption Due to Reheating—4-stage Feed Heating

Steam conditions: 1200 lb. per sq. in. gage; 750° F. inlet temperature; 29-in. vacuum; 5% pressure drop over reheater.

Temperature of reheat, deg. F.	Pressure entering reheater which produces maximum reduction in heat consumption, lb. per sq. in., abs.	Reduction in heat consumption, percent	Reduction in steam flow to condenser due to reheat, percent	Reduction in heat absorbed by condenser circulating water due to reheating, percent
850	250	6.4	20.0	10.5
750	220	4.55	16.0	7.7
650	200	2.75	11.8	4.7
550	175	1.35	7.9	2.35
450	150	0.40	4.3	0.70

The temperature of saturated steam at 1200 lb. per sq. in. gage is about 569° F. Allowing a terminal difference in a live-steam reheater of 19° F., the reheat temperature to the low-pressure section would be 550° F., on which the maximum reduction of heat consumption with 4-stage extraction is 1.35% at most efficient load. At other loads the gain is less. This gain may be entirely wiped out if the pressure drop through the reheater should reach 17%. All gains are affected by this pressure drop through the reheater. Hence the need of large piping and passages and of few baffles.

Reynolds states that the maximum gain from reheating decreases but slightly with increased pressure. Thus, with 4-stage extraction, 750° F. inlet and reheat temperatures, and with 29 in. vacuum, the maximum gain with 600 lb., gage, steam pressure is 4.9%; at 1200 lb., gage, it is 4.55%. The decrease appears less rapid at higher pressures.

Since more heat per pound is converted into work with reheating, less steam enters the throttle for a given load, and less passes to the condenser. The percent reductions at the stated reheat pressures are given in Table 36. A corresponding reduction can be made in the size of the condensate pump, feedwater heaters, feedwater piping and boiler feed pump. But reheating involves the rejection of more heat per pound of steam to the condenser. The percent reduction in the heat absorbed by the circulating water also is shown in Table 36 for reheat effect only. A corresponding reduction can be made in condenser and circulating pump sizes.

Reheating introduces problems of reheat temperature control, of regulation when load suddenly is removed and of complicated station layout. With increased steam temperatures, simple regenerative plants are preferred, despite the thermal advantages of reheat.

For further information see: Guy, Tendencies in Steam Turbine Development, *Proc.*, Inst. Mech. Engrs., 1929; Guy, The Economic Value of Increased Steam Pressure, *Proc.* Inst. Mech. Engrs., 1927; Baumann, Some Considerations Affecting the Future Development of the Steam Cycle, *Proc.* Inst. Mech. Engrs., 1930.

17. ECONOMIC SELECTION OF OPERATING CONDITIONS

PRESSURES AND TEMPERATURES for steam turbines have changed greatly over the last fourteen years (1921 to 1935), and are being increased. Steam temperatures of 750° F. to 850° F. are in use and turbines are offered for 925° F. As the result of a two-year experience with a 10,000-kw. turbine, operating with initial temperature of 1000° F., Thompson and Van Duzer (High Temperature Steam Experience at Detroit, *Trans.* A.S.M.E., FSP-56-9, July, 1934) state that a plant now can be built to operate on 1000° F.

and that reliable service can be expected. Carter and Ellenwood (The Thermal Performance of the Detroit Turbine Using Steam at 1000° F., *Trans.* A.S.M.E., FSP–56–8, July, 1934) estimate that a 50,000-kw. turbine with 1200 lb. per sq. in., 1000° F., 29 in. vacuum, no reheating, and about 12% moisture at exhaust, should operate close to 8600 B.t.u. per kw.-hr. of net generator output. Turbines form only 15 to 20% of total station cost, and consideration must be given to load factor, coal cost, and cost of complete plant in making the selection. In general, the larger the capacity, the lower the unit cost. Boilers now operate at 3200 lb. per sq. in., and pressures of 6000 lb. per sq. in. are being tried experimentally.

While stage efficiencies tend to decrease with high pressures, the reheat factor increases and the overall efficiency of the turbine changes but little. F. Hodgkinson (Steam Turbine and Condensing Equipment, *Elec. Jour.*, Dec., 1924) gives engine efficiencies of 80.5% for 400 lb. unit with no reheat, and 79.5% for 1500 lb. unit with reheat. These values are increased in present day units.

EXHAUST CONDITIONS.—The lowest exhaust pressure gives the greatest heat drop and largest potential power per pound of steam. It also entails long low-pressure blading, or in a given casing either limits output or increases leaving velocity and exhaust losses. In any specific case the weighted average cooling water temperature should be determined from an analysis of water temperatures throughout the year and the loads at such times. This weighted average temperature may vary from the mean temperature by several degrees. When the weighted temperature is found, the corresponding vacuum may be estimated by allowing for a temperature rise (usually 10 to 15° F.) and a terminal difference leaving the condenser (usually 9 to 12° F.). When this vacuum is found the economic rating of a given casing can be chosen or blade lengths adjusted to the steam volume.

METALS FOR HIGH TEMPERATURE.—Steel and other metals decrease rapidly in tensile strength at temperatures above 750° F. Data have been published in many papers and books on this decrease of strength. Besides the references on creep given below, see also Symposium on Effect of Temperature on the Properties of Metals, A.S.M.E. 1931, and bibliography included in this symposium.

A further factor is *creep,* the name given to that indefinite elongation that occurs at higher temperatures under constant load at stresses much below the elastic limit of the material. This rate of flow is stated in terms of hours. S. H. Weaver (The Effect of Temperature on Materials Required in Turbine Design, *Gen. Elec. Rev.*, Nov., 1930) shows that the limiting creep stress on a 0.23% carbon steel at 900° F. is 14,200 lb. per sq. in. A load of 10,300 lb. per sq. in. produces a flow of 1% in 10,000 hrs.; a 1% flow results from 8,000 lb. per sq. in. load in 100,000 hrs. while a 6000 lb. per sq. in. load produces an elongation of 0.1% in 100,000 hrs. Certain designers have chosen the allowable rate of creep as 0.01% per year. This permits turbines to be built safely with material now available for 900° F. and possibly for 1000° F.

Particular attention must be given to the matter of creep in bolting material, in disc and diaphragm construction and in piping connections. Data on various materials can be found in Norton's The Creep of Steel at High Temperatures, McGraw-Hill Book Co.; Kanter & Spring's Long Time or Flow Tests of Carbon Steels at Various Temperatures, A.S.T.M., 1928; McVetty, Working Stresses for High Temperature Service, *Mech. Engg.*, Mar., 1934; Baumann, Some Considerations Affecting the Future Development of the Steam Cycle, *Proc.* Inst. Mech. Engrs., 1930. In this article, Baumann gives permissible creep rates per hr. as follows: 1. Turbine discs pressed on shafts 10^{-9}. 2. Bolted flanges of turbine cylinders 10^{-8}. 3. Steam piping, welded joints, boiler tubes 10^{-7}. 4. Superheater tubes, 10^{-6}. These stresses must be well below the yield point. He suggests a factor of safety of 3 based on the yield point at the working temperature.

PROBABLE STATION CONDITIONS will be: With no reheat, 450 lb. per sq. in., 750° F. for stations with 10,000- to 50,000-kw. units, with $4 per ton coal, and with load factor under 50%. Single-cylinder units with up to 4 stages of bleeding will be used. For stations with 1- and 2-cylinder units of from 25,000 to 125,000 kw., and no reheating, 750 lb. per sq. in., 850° F. Newer stations of this type may use 1000° F. and no reheat. For certain high-pressure stations with high load factor, 1200 lb. per sq. in., 800° F., and reheat to 950° F., with 4 to 5 bleeders. This plant costs from 5 to 10% more than a 400-lb. plant, but is about 13% more economical in fuel. If fuel cost and load factor are such in 450-lb. plant that fuel charges equal fixed charges, it pays to consider high-pressure plant.

Single-cylinder units in sizes up to 100,000 kw. are preferred, and are economic for 750 lb. per sq. in., 850° F. When cross-compound units are used, the high-pressure units generally operate at 3600 r.p.m. The capacity of the high-pressure unit depends on whether steam or gas reheat is used.

G. A. Gaffert (High-pressure Steam and Binary Cycles as a Means of Improving

Power Station Efficiency, *Trans.* A.S.M.E., FSP–56–11, Oct., 1934) considers the possibilities of these various cycles. His results, given in Table 37, are based on the following assumptions: *a.* An overall efficiency ratio of 82% for turbines of 30,000 to 50,000 kw. capacity; *b.* a maximum of 11% moisture at exhaust at full load; *c,* terminal differences on feedwater heaters of 5 to 20° F. for feed temperatures of 100 to 525° F.; *d,* steam generator efficiencies of 85%, including air preheater (if used); *e,* pressure drops of 10% between boiler and turbine, bleed points and their respective heaters, reheater piping and reheater; *f,* radiation loss of 2% from bleed point to heater, and 3% for reheating lines; *g,* normal auxiliary power allowances, with 20 kw.-hr. per ton as power for pulverizing coal; *h,* feedwater heated in equal temperature steps to a maximum of 75 to 80% of saturation temperature corresponding to throttle pressure when most economical number of feedwater heaters is employed; *i,* an overall efficiency ratio of 75% for the mercury vapor turbine; *j,* a terminal difference of 30° F. across the mercury condenser-boiler between mercury condensate and steam vapor temperatures. With diphenyloxide, a difference of 30° F. was assumed for the condenser-boiler at 25 lb. exhaust pressure for the diphenyloxide.

Table 37.—Plant Performances for Steam and Binary Cycles

Initial and Reheating Pressure* and Temperature Cycles Final Vacuum = 29 in. in all cases	Number of Points of Steam Extraction B.t.u. per kw.-hr. of Station Output			
400 lb.- 800° F. steam	12,800	12,600	12,460	12,400
400 lb.- 900° F. steam	12,460	12,280	12,160	12,100
400 lb.-1000° F. steam	12,180	11,960	11,840	11,760
600 lb.- 800° F. steam	12,180	11,930	11,830	11,760
600 lb.- 900° F. steam	11,900	11,700	11,620	11,590
600 lb.-1000° F. steam	11,720	11,500	11,430	11,390
900 lb.-1000° F. steam	11,300	11,100	11,000	10,930
1200 lb.-1000° F. steam	11,050	10,800	10,680	10,580
1200 lb.- 800° F.-R 200 lb.-800° F.†	11,300	11,060	10,880	10,800
1200 lb.-1000° F.-R 200 lb.-1000° F.†	10,630	10,400	10,260	10,220
2500 lb.- 800° F.-R 500 lb.-800° F.†	11,080	10,850	10,680	10,580
2500 lb.-1000° F.-R 500 lb. 1000° F.†	10,300	10,080	9,930	9,860
3226 lb.-800° F.-R 900 lb.-800° F. 2nd R-200 lb.-800° F.†	10,760	10,550	10,440	10,360
3226 lb.-1000° F.-R 900 lb.-1000° F. 2nd R-200 lb.-1000° F.†	9,880	9,730	9,620	9,550
DPO.‡ 146 lb.-750° F.- 25 lb. Exh.§ Steam 730 lb.-800° F. R-100 lb.-800° F.†	10,920	10,780	10,700	10,680
DPO.‡ 210 lb. -800° F.- 25 lb. Exh.§ Steam 730 lb.-1000° F.	10,630	10,550	10,480	10,460
Mercury 46 lb.-800° F.- 4 in. Exh.§ Steam 500 lb.-800° F.	9,720	9,640	9,600	
Mercury 95 lb.-900° F.- 4 in. Exh.§ Steam 500 lb.-800° F.	9,300	9,210	9,170	
Mercury 180 lb.-1000° F.- 4 in. Exh.§ Steam 500 lb.-800° F.	8,900	8,850	8,830	
Mercury 200 lb.-1020° F.- 4 in. Exh.§ Steam-Mercury Superheated at 500 lb.-800° F.	8,700	8,630	8,600	

* Pressures in lb. per sq. in., abs. † R = Reheat conditions. ‡ DPO. = Diphenyloxide.
§ Exh. = Exhaust pressure in lb. per sq. in. or inches of mercury, absolute.

Gaffert concludes that higher steam pressures economically are justified, and that the steam cycle has not reached its limit; that when metals become available for temperatures over 1000° F., the mercury-steam binary cycle is the only feasible one; and that when thermal advantages and capital costs are considered, there is little choice between mercury-steam, diphenyloxide-steam and high pressure steam cycles, assuming low fuel costs and 800° F. initial temperature.

VARIABLE-PRESSURE OPERATION is possible with boilers operating at or above the critical pressure at constant temperature, such as the Benson. A description of this boiler appears in *The Engr.*, April 21, June 23, 1933. The specific volume at approximately constant temperature varies inversely as the pressure. A turbine for 2250 lb. pressure will pass about four times as much steam as one for 550 lb. pressure and the same temperature, with some loss in vacuum. The heat drop and heat utilized remain nearly constant in both cases. Hence, it appears possible to govern the load on the turbine by varying the boiler pressure and thus the inlet pressure. This would eliminate the use of throttling or nozzle governing valves.

THE LIMITING FACTOR IN TURBINE CAPACITY is the permissible area of the annulus of the last blade row. The amount of steam that can be discharged to the condenser is determined by the allowable leaving loss, and hence the allowable leaving velocity from this last row.

Kearton (Steam Turbines, p. 258) develops a formula for the mean diameter at the

exhaust end of an impulse turbine with no extraction from which the following is derived:

$$D = 21.46\sqrt{\frac{-}{\eta}} \qquad \times$$

where D = mean diameter, in.; x_E = quality of steam at outlet of blade; v_E = specific volume of steam at exhaust pressure, cu. ft. per lb.; ρ = ratio of blade speed to steam speed for last row; η_g = engine efficiency referred to generator output; $(h_1 - h_2)$ = isentropic heat drop from initial conditions to final pressure; m = allowable ratio of blade length to mean diameter (maximum value in 1934, $m = 0.35$); c = thickness coefficient, *i.e.*, ratio of clear opening at outlet of blades to pitch of blades (often assumed = 1.0); α = nozzle angle for last stage; kw. = kilowatt output of generator; N = r.p.m.

For a Parsons turbine, $D =$

where w = total pounds of steam per sec. passing last blade row, and α_1 = outlet angle of last blade row, other notation being as before.

The maximum permissible annulus area largely is determined by the permissible tip speed and blade length, with maximum tip speed of 1257 ft. per sec. (1934). Increased tip speeds contribute to increased economy.

Blade length depends upon blade strength, and the method of blade fastening. Improved blade designs permit higher tip speeds, and thereby increase area of the annulus.

Baumann blades are used by several companies to increase the capacity of the last row. One or more partitions are placed across both orifices and blades of several of the last rows. The outer section in each case expands the steam to exhaust pressure and delivers it to the exhaust pipe. The inner sections by-pass the steam to the succeeding orifices where expansion takes place. By this means the actual last row annulus area may be increased 60% by one extra Baumann row, 120% by two extra rows and 180% by three extra rows.

The greater the allowable leaving loss the greater is the turbine capacity. For instance, if the leaving loss can be doubled, the turbine capacity is increased about 40%. Doubling the leaving loss does not signify an equivalent decrease in turbine efficiency, for increased steam flow decreases disc, gland, and leakage losses, and increases the high-pressure blade lengths, while the mechanical losses per lb. of steam also decrease. Hence, the turbine itself may decrease in efficiency by only half the amount of the added leaving losses. The selection of the economic leaving loss for a given turbine, and hence its economic rating, depends on certain factors extraneous to the turbine, as load factor, daily and annual load curves, coal cost, station cost, and cooling water temperatures. The load factor should be the average throughout the whole useful life of the turbine. N.E.L.A. records show that this is comparatively low. Daily and annual load curves determine the nature of the loading, the point of best economy, and the permissible sacrifice of efficiency at occasional full load. Coal costs are highly important, as they fix the value of increased efficiency with decreased leaving loss. Station cost influences the amount of money that may be spent on the turbine. Water temperatures influence vacuum. For instance, if 28-in. vacuum only can be obtained, practically twice the weight of steam can be passed through the last blade row as with 29-in. vacuum and the same leaving velocity.

A given turbine casing can have a low rating for low leaving loss and maximum economy, or a high rating with high leaving loss and decreased economy. Foreign practice favors low leaving loss, not exceeding 15 B.t.u. per lb. American design formerly followed this practice. Turbines, N.E.L.A., 1932, gives a table which shows that at full rating, leaving losses vary from 6.6 to 33 B.t.u. per lb. Balcke and Alden (Steam Turbine Plant Practice in the U. S., *Trans.* A.S.M.E., FSP-55-3*a*, 1933) suggest for maximum load, winter conditions, and a 29 in. vacuum, a leaving loss of 25 to 30 B.t.u. per lb. of steam. American practice in last row annulus varies from 1.2 to 2.5 sq. ft. per 1000 kw. rated capacity, with the trend towards the lower figure. Bleeding for regenerative heating of the feedwater tends to decrease the steam to exhaust, and thereby increase possible rating of a given casing. Reheating has a similar effect in decreasing leaving loss.

Part of the kinetic energy of the leaving velocity is converted into pressure in the exhaust passage to the condenser. Practice allows area of exhaust openings to be 2.5 to 5 sq. ft. per 1000 kw. of rated capacity. With high leaving losses, large condenser outlets are not necessary, and a saving may be effected in the cost of this item.

INCREMENT LOADING.—The incremental rate of a turbine at any given output is the slope of the input-output curve at the point corresponding to the output. The

best combined efficiency of turbines carrying load in parallel is obtained when these are operated at outputs corresponding to equal incremental rates. See The Theory of Incremental Rates, Stienberg and Smith, *Elec. Eng.*, March-April, 1934.

THE ECONOMIC SELECTION OF A LARGE STEAM TURBINE is based on the above considerations of economy and cost. The extra costs of decreased unit efficiency, resulting from increased leaving loss, must be balanced against the fixed charges on the added cost of units of higher efficiency with lower leaving loss. This is fully discussed and a method of computation outlined by Christie (Economic Considerations in the Application of Modern Steam Turbines to Power Generation, *Mech. Engg.*, Aug.-Sept., 1930).

FLOOR AREA in sq. ft. per 1000 kw. is given in Christie's and Warren's papers, World Power Conference, Berlin, 1930, and presented in Table 38.

Table 38.—Floor Area per 1000 kw. of Steam Turbines

Size, kw.	Type of Turbine					
	3600 r.p.m. units	1800 r.p.m. single cylinder	Vertical compound	2 cyl. tandem 1800 r.p.m.	2 cyl. cross compound	3 cyl. cross compound
	Sq. ft.	Sq. ft.	Sq. ft.	Sq. ft.	Sq. ft.	Sq. ft.
2,000	95	....	....	....	....	....
5,000	52	....	....	....	....	....
10,000	42	50 to 60	....	....	....	....
20,000	....	37 to 48	....	....	....	....
40,000	....	27 to 36	22	30	....	....
60,000	....	20 to 27	18	25	....	....
80,000	....	17 to 24	14	20	....	....
100,000	....	12 to 23	12.5	17	25	....
150,000	....	....	....	12.5	17	23
200,000	....	....	....	....	13	17

WEIGHTS OF STEAM TURBINES for condensing units for representative turbo-generator installations as shown in Table 39, are from Christie's World Power Conference paper, Berlin, 1930.

Table 39.—Weights of Steam Turbines for Condensing Units

Size, kw.	1,000	2,000	5,000	10,000	20,000	40,000	60,000	80,000	
Weight, lb. per kw., 3600 r.p.m.	40	31	23	19					
Weight, lb. per kw., 1800 r.p.m.				29	24	20	17	15	13

THE SELECTION OF INDUSTRIAL TURBINES depends upon the power demand, the efficiency warranted, use factor of unit, and its cost. If the plant has large low-pressure steam requirements, a high-pressure turbine may serve as a reducing valve from high-pressure boilers. When all steam is utilized in industry, the electric energy is generated at all loads under average conditions at about 4600 B.t.u. per kw.-hr., or roughly, $1/3$ lb. of coal per kw.-hr. A complete analysis of this problem is given by Flanders in Turbine Arrangements for Supplying Industrial Power (*Power*, June 7, 1932).

Limited cooling water at the industrial plant may warrant a turbine for an extra high pressure to lessen the heat to condenser per kilowatt of plant output. These, and other economic influences, indicate increasing use of higher pressures and temperatures in industrial plants. Many old industrial turbines, of uneconomical design, could be replaced by modern efficient units with the old alternator. Savings up to 17% are possible with no change in steam conditions. An additional saving of 5% may be effected by regenerative feedwater heating with new units. With same quantity of steam as used by the old turbines, a greater electrical output can be obtained.

The Selection of a Bleeder Turbine involves a careful study of the conditions in the plant for which it is intended. The calculation of the performance of a bleeder turbine is somewhat complicated and can be represented best by a set of performance curves, based on the local plant conditions. These curves must be supplied by the builder of the turbine. See Campbell, The Field and Limitations of Extraction Turbines, *Elec. Jour.*, Nov., 1927; Gove, Picking the Turbine to Fit the Job, *Elec. Jour.*, Mar., 1933.

18. MERCURY TURBINES

Mercury, as a power generating fluid, has high boiler temperatures for moderate pressures (see Table 8, p. 5-16) and does not decompose at any reasonable temperature. Mercury vapor can be utilized in a properly designed turbine, where it is expanded in stages to the exhaust pressure. The condensing temperatures at usual vacua (see Table 8, p. 5–16) are so high that it can be used to evaporate water at a pressure suitable for

steam turbine operation. Hence stations can be built to operate upon the mercury vapor-steam cycle. This combination leads to high economy in the plant. See data in Table 37.

The mercury vapor-steam plant of the Hartford Electric Light Co., of 10,000 kw. capacity, developed power at the switchboard for a year at an average fuel consumption of 10,250 B.t.u. per kw.-hr.

The mercury-turbine at Hartford is a 5-stage impulse unit, and operates at 70.7 lb. per sq. in. gage pressure, 28.5 in. vacuum. More stages will be used on new units with higher mercury pressures. The wheel speed of the Hartford unit is about 300 ft. per sec. The turbine wheels are overhung on the outer end of the generator shaft in order to save gland. The engine efficiency, based on shaft horsepower of the Hartford mercury bine, is said to be about 77% at full load, and only a little less at half load.

The mercury condenser is arranged for a temperature head of 20° F. between condensing mercury and boiling steam.

Besides the Hartford station, two other plants have been installed (1934). One at Schenectady, N. Y., of 20,000 kw. capacity delivers power to the New York Power and Light Co. and sends 325,000 lb. of steam per hr. to the works of the General Electric Co. The other plant, also of 20,000 kw. capacity, is being built at the Kearny Station of the Public Service Electric and Gas Co. of N. J., and delivers steam to turbines now in place, in which 33,000 kw. will be generated, thus making a total output of 53,000 kw. for the combined mercury-steam unit. Table 40 gives comparative operating conditions for the three stations.

Table 40.—Operating Conditions of Mercury Vapor-Steam Turbine Stations

	Hartford	Kearny	Schenectady
Load on mercury turbine, kw.	10,000	20,000	20,000
Speed of mercury turbine, r.p.m.	720	900	900
Total steam from unit, lb. per hr.	129,000	325,000	325,000
Steam pressure, lb. per sq. in., gage	275	365	400
Steam temperature, deg. F.	680	750	760
Mercury vapor, pressure at turbine, lb. per sq. in., gage	70.7	125	125
Mercury vapor temperature at turbine, deg. F.	884.5	958	958
Vacuum of mercury condenser, in. Hg, abs.	1.5	3.0	3.0
Temperature of mercury vacuum, deg. F.	440	485	485

The mercury-vapor plant appears to be economically justified on a system where fuel costs are high or where the unit with its accompanying steam turbine can be given a constant heavy loading throughout the year.

Section 9

CONDENSING AND COOLING EQUIPMENT

By Robert Thurston Kent

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CONDENSING AND COOLING EQUIPMENT

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CONDENSING EQUIPMENT

TYPES OF CONDENSERS.—The two general types of condensers are: 1. Direct contact condensers, in which the steam to be condensed comes in direct contact with the condensing water. They include jet, barometric, and ejector condensers. 2. Surface condensers, in which steam and condensing water circulate on opposite sides of a metallic condensing surface.

1. DIRECT CONTACT CONDENSERS

THE JET CONDENSER, Fig. 1, comprises a head into which the steam is delivered and where it is condensed by water sprays, a pump for the removal of condensate and condensing water, and a vacuum pump or steam-air jet ejector for removing air and non-condensible gases. Sometimes the same pump is used for removing both air and water. The jet condenser may be used where there is an ample supply of condensing water, particularly if this is of the proper quality for boiler feeding. Condensing water is drawn into the head by means of the partial vacuum therein, and no circulating pump is required.

Vacuum Breakers are required on jet condensers to admit air to the condenser body if water rises in it to a predetermined level. If permitted to rise above this level, the water might flood and damage the prime mover. The vacuum breaker consists of a float that opens an air valve, thereby destroying the vacuum and stopping the flow of condensing water.

Cost.—The jet condenser is low in first cost, but its operating expense is higher than that of other types of condensers.

Maximum Suction Lifts for the condensing water should be from 15 to 18 ft. If greater, an overload on the condenser would tend so to decrease the vacuum that insufficient condensing water would be drawn for operation. Fig. 4 is a chart issued by the Elliot-Erhart Company showing permissible condenser overloads at various suction lifts

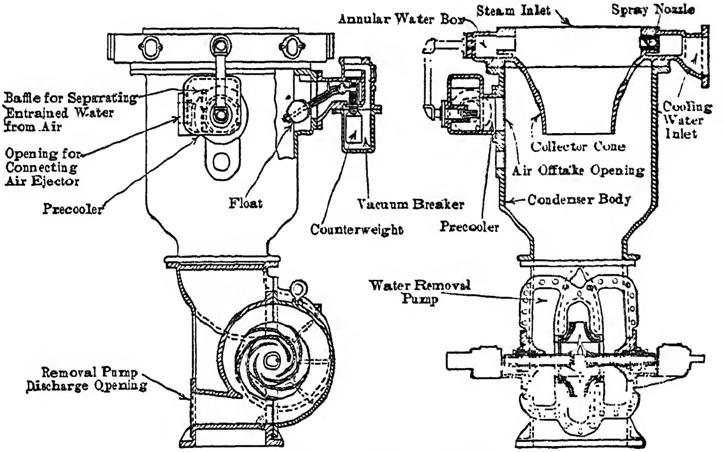


FIG. 1. Westinghouse Jet Condenser

The chart is based on condensing water at 70° F. At lower temperatures the permissible overload is greater.

The use of the chart is as follows: Let L = load on condenser, lb. of water per hr., at suction lift m and vacuum v; l = proportionate load at lift m and vacuum v; X and X_a = actual load on condenser at incipient and absolute instability, respectively; x and x_a = proportionate load at point of intersection of curve of suction lift m, with lines of incipient and absolute instability respectively. Then

$$X = L(x/l) \quad \text{and} \quad X_a = L(x_a/l).$$

Example.—Load L = 50,000 lb. per hr.; vacuum, 27 in.; suction lift 18 ft. Proportional load (v = 27; m = 18) = 140; proportional load of incipient instability (m = 18) = 199. Load of incipient instability X = 50,000 (199/140) = 71,071 lb.

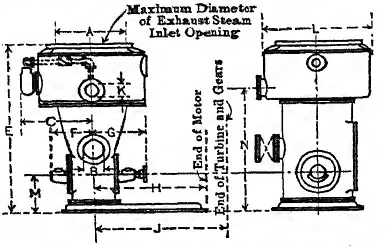


Fig. 2. Dimensions of Wheeler Low-level Jet Condensers

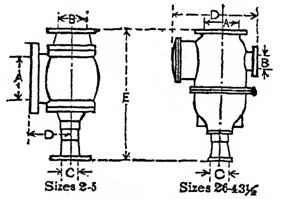


Fig. 3. Dimensions of Schutte and Koerting Barometric Jet Condensers

Table 1.—Dimensions of Low Level Vertical Cylindrical Jet Condensers

(C. H. Wheeler Mfg. Co., Philadelphia, 1935)

Size	Dimensions, in. (See Fig. 2.)											
	A	*B*	*C*	*E*	*F*	*G*	*H*	*J*	*K*	*L*	*M*	*N*
JC	16	4	25	82	19	23	51	100	4	28	14	66
JD	20	5	28	88	20	24	58	102	4	34	17	70
JE	24	6	31	92 1/2	24	28	64	105	5	42	18 1/2	72
JF	28	7	33	97 1/2	24	28	67	111	6	48	20 1/2	74
JG	30	8	38	107	25 1/2	30	76	117	7	54	21	82
JH	36	9	47	112 1/2	27	32	78	120	8	62	22 1/2	84
JK	42	10	52	118 1/2	29	34	80	129	9	72	24 1/2	87
JL	48	12	56	127	32	37	90	132	10	80	29	92
JM	54	14	58	139 1/2	33	38	91	135	12	84	31 1/2	102
JN	60	16	62	146	34	39	97	138	14	92	33	104
JP	72	18	67	162	35	40	108	138	16	102	35	116

Table 2.—Capacities and Dimensions of Barometric Multi-jet Condensers

(Schutte & Koerting Co., Philadelphia, 1935)

Size No.	Max. Water Capacity, gal. per min.	Diameter of Connections, in. (See Fig. 3)			Overall Dimensions, in. (See Fig. 3)		Weight, lb.
		A	B		D	E	
					ft. in.	ft. in.	
2	50	5	2 1/2	2	0– 7 1/2	2–2 1/4	
3	85	6	3	2 1/2	0– 8 1/4	2–7 1/4	
4	130	8	4	3	0– 9	3–1 5/8	
5	210	10	4	4	0–10	3–8 3/4	
26	250	12	5	4	2– 6		1000
27	300	12	5	4	2– 6 1/2		1000
28	350	12	5	4	2– 6 1/2	4–4 5/8	1000
29	400	12	5	4	2– 6 1/2	4–4 5/8	1200
30	450	12	5	4	2– 6 1/2	4–4 5/8	1200
31	550	14	6	5	2–10 3/4	5–1 1/4	1600
32	650	16	6	5	2–11 1/4	5–1 1/4	1800
33	800	18	8	5	3– 3	5–6 1/4	2500
34	900	18	8	5	3– 3	5–6 1/4	2500
34 1/2	1000	18	8	5	3– 3	5–6 1/4	2500
35	1100	20	8	6	3–10 1/2	6–6 5/8	3200
35 1/2	1250	20	8	6	3–10 1/2	6–6 5/8	3200

Size No.	Max. Water Capacity, gal. per min.	Diameter of Connections, in. (See Fig. 3)			Overall Dimensions, in. (See Fig. 3)		Weight, lb.
		A	B	C	D	E	
					ft. in.	ft. in.	
36	1,400				3–10 1/2	6– 6 5/8	3,500
36 1/2	1,600				3–10 1/2	6– 6 5/8	3,500
37	1,800				4–10	8– 9 3/8	6,000
37 1/2	2,100				4–10	8– 9 3/8	6,000
38	2,400			10	4–11	8– 9 3/8	6,500
38 1/2	2,800					8– 9 3/8	6,500
39	3,300					10–10 3/8	8,600
39 1/2	4,000				5– 7 3/4	10–10 3/8	8,600
40	5,000				6– 7	11– 8	12,000
40 1/2	6,000				6– 7	11– 8	12,000
41	7,000				7– 8	12– 6	15,000
41 1/2	8,000				7– 8	12– 6	15,000
42	9,000				8– 4	13– 4	18,000
42 1/2	10,000				8– 4	13– 4	18,000
43	12,000				9– 0	14– 2	22,000
43 1/2	14,000				9– 0	14– 2	22,000

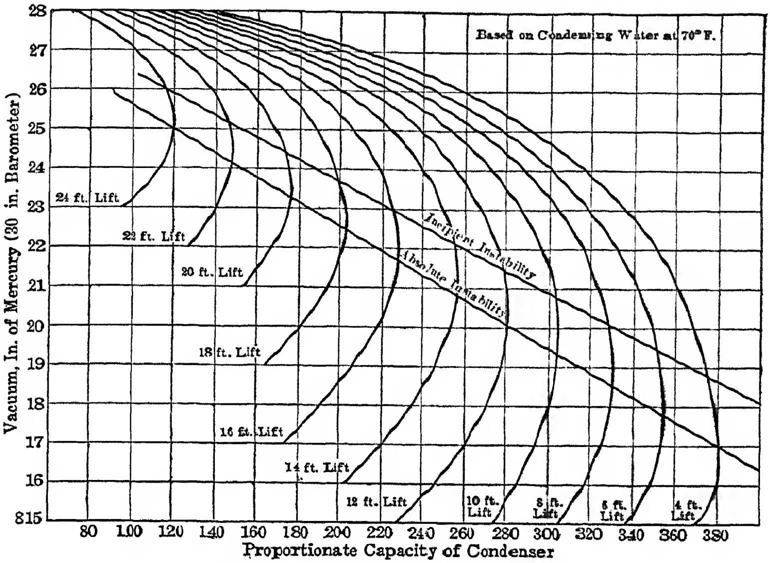


FIG. 4. Stability Chart for Jet Condensers

THE BAROMETRIC CONDENSER consists of a condenser head similar to that of a jet condenser, mounted at the top of a discharge pipe whose length is at least 34 ft. above the level of the hotwell. The condensate and condensing water falling into the tail pipe draw out the entrained air. As a rule, no vacuum pump is required with this type of condenser although sometimes a dry air-pump may be used to remove such air as may not be removed by the descending water column. If the condensing water inlet is not over 20 ft. above the source of supply, condensing water will be drawn into the condenser by the partial vacuum, as in a jet condenser, and no circulating water pump will be necessary.

Fig. 5 shows the Schutte & Koerting multi-jet barometric condenser. The condensing water is discharged through a series of small nozzles *A* into a combining tube *B* and an extension *C*, consisting of several sets of tapered rings. The water jets are directed into the throat *D* where they unite to form a single jet. The vapors flow through the annular passes of the combining tube and there are condensed. The water jets entrain the air and non-condensible gases and discharge them into the barometric tail pipe. This type of condenser has had wide application in connection with evaporators, vacuum pans, dryers, stills, etc. Table 2 gives dimensions and capacities.

THE EJECTOR CONDENSER operates in a manner similar to a steam ejector. In the Koerting Eductor condenser, built by Schutte & Koerting Co., Philadelphia, condensing water enters through a series of nozzles at a pressure of 5 to 9 lb. per sq. in. and condenses the steam. The condensate and steam enter the discharge pipe at high velocity and discharge at atmospheric pressure, air and non-condensible gases being entrained and discharged with the jet. Water capacities of these condensers range from 15 to 750 gal. per min. Steam capacities, at 26-in. vacuum, range from 150 to 750 lb. per hr. Condensing water is supplied by a circulating pump, no vacuum pump being required.

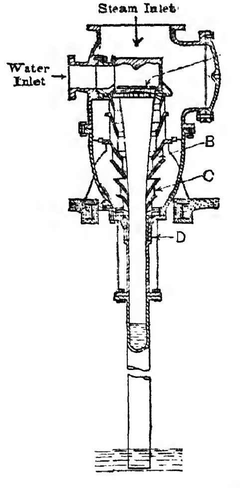


FIG. 5. Schutte and Koerting Barometric Jet Condenser

Table 3.—Properties of Saturated Steam at Pressures below Atmospheric Pressure
(Compiled from Keenan's Steam Tables)

Temperature, deg. F, T_s	Vacuum, in. of Mercury, (30-in. Bar.)	Pressure, abs.		Specific Volume, cu. ft. per lb.	Heat Content		Temperature, deg. F, T_s	Vacuum, in. of Mercury, (30-in. Bar.)	Pressure, abs.		Specific Volume, cu. ft. per lb.	Heat Content	
		Lb. per sq. in.	In. of Mercury (Mercury at 32° F.)		Of Liquid, B.t.u.	Of Vapor, B.t.u.			Lb. per sq. in.	In. of Mercury (Mercury at 32° F.)		Of Liquid, B.t.u.	Of Vapor, B.t.u.
32	29.819	0.0887	0.1806	3301	0.00	1073.4	86	28.747	0.6153	1.253	527.7	54.00	1098.0
33	29.812	.0923	.1879	3178	1.01	1073.9	87	28.707	.6352	1.293	512.0	55.00	1098.4
34	29.804	.0961	.1956	3059	2.01	1074.3	88	28.665	.6555	1.335	497.1	56.00	1098.9
35	29.796	.1000	.2036	2946	3.02	1074.8	89	28.623	.6765	1.377	482.5	57.00	1099.4
36	29.788	.1041	.2119	2835	4.03	1075.3	90	28.579	.6980	1.421	468.5	58.00	1099.8
37	29.779	.1083	.2205	2731	5.03	1075.7	91	28.534	.7201	1.466	454.9	58.99	1100.2
38	29.770	.1126	.2293	2632	6.04	1076.2	92	28.487	.7429	1.513	441.8	59.98	1100.7
39	29.762	.1171	.2384	2536	7.04	1076.6	93	28.440	.7662	1.560	429.1	60.98	1101.2
40	29.752	.1217	.2478	2445	8.05	1077.1	94	28.391	.7902	1.609	416.8	61.97	1101.6
41	29.742	.1265	.2576	2357	9.05	1077.6	95	28.341	.8149	1.659	404.9	62.96	1102.0
42	29.732	.1315	.2677	2272	10.05	1078.0	96	28.289	.8403	1.711	393.3	63.96	1102.5
43	29.722	.1367	.2783	2190	11.05	1078.5	97	28.236	.8663	1.764	382.2	64.95	1102.9
44	29.711	.1420	.2891	2112	12.05	1078.9	98	28.182	.8930	1.818	371.4	65.94	1103.4
45	29.700	.1475	.3003	2037.2	13.05	1079.4	99	28.126	.9205	1.874	361.0	66.94	1103.8
46	29.688	.1532	.3119	1965.3	14.06	1079.8	100	28.068	.9487	1.932	350.8	67.93	1104.2
47	29.676	.1591	.3239	1896.1	15.06	1080.3	101	28.010	.9776	1.990	341.1	78.92	1104.7
48	29.664	.1652	.3363	1829.7	16.06	1080.8	102	27.949	1.0072	2.051	331.6	69.92	1105.1
49	29.651	.1715	.3492	1765.9	17.06	1081.2	103	27.887	1.0377	2.113	322.4	70.91	1105.6
50	29.638	.1780	.3624	1704.8	18.06	1081.7	104	27.824	1.0689	2.176	313.6	71.91	1106.0
51	29.624	.1848	.3763	1645.3	19.06	1082.1	105	27.759	1.1009	2.241	305.0	72.91	1106.4
52	29.609	.1918	.3905	1588.3	20.06	1082.6	106	27.692	1.1338	2.308	296.6	73.90	1106.9
53	29.595	.1989	.4050	1534.6	21.06	1083.0	107	27.623	1.1675	2.377	288.6	74.90	1107.3
54	29.580	.2063	.4200	1482.4	22.06	1083.5	108	27.553	1.2020	2.447	280.8	75.90	1107.8
55	29.566	.2140	.4357	1431.8	23.06	1084.0	109	27.480	1.2375	2.520	273.2	76.89	1108.2
56	29.548	.2219	.4518	1383.5	24.05	1084.4	110	27.406	1.274	2.594	265.8	77.89	1108.6
57	29.532	.2300	.4683	1337.4	25.05	1084.8	111	27.331	1.311	2.669	258.6	78.89	1109.1
58	29.515	.2384	.4854	1292.8	26.05	1085.3	112	27.251	1.350	2.749	251.7	79.89	1109.5
59	29.497	.2471	.5031	1249.6	27.05	1085.8	113	27.172	1.389	2.828	245.0	80.89	1110.0
60	29.479	.2561	.5214	1208.0	28.05	1086.2	114	27.091	1.429	2.909	238.6	81.89	1110.4
61	29.460	.2654	.5404	1167.9	29.05	1086.7	115	27.007	1.470	2.993	232.3	82.89	1110.8
62	29.440	.2749	.5597	1129.7	30.05	1087.1	116	26.922	1.512	3.078	226.2	83.88	1111.2
63	29.420	.2848	.5799	1092.5	31.05	1087.6	117	26.834	1.555	3.166	220.4	84.88	1111.7
64	29.400	.2949	.6004	1057.1	32.04	1088.0	118	26.742	1.600	3.258	214.5	85.88	1112.1
65	29.378	.3054	.6218	1022.7	33.04	1088.5	119	26.651	1.645	3.349	209.0	86.88	1112.6
66	29.356	.3162	.6438	989.6	34.05	1089.0	120	26.595	1.692	3.445	203.6	87.88	1113.0
67	29.334	.3273	.6664	957.8	35.04	1089.4	121	26.459	1.739	3.541	198.4	88.88	1113.4
68	29.310	.3388	.6898	927.1	36.03	1089.8	122	26.360	1.788	3.640	193.3	89.88	1113.8
69	29.286	.3506	.7138	897.6	37.03	1090.3	123	26.258	1.838	3.742	188.3	90.88	1114.3
70	29.261	.3628	.7387	869.0	38.03	1090.8	124	26.154	1.889	3.846	183.5	91.88	1114.7
71	29.238	.3754	.7623	841.4	39.03	1091.2	125	26.048	1.941	3.952	178.9	92.87	1115.1
72	29.209	.3883	.7906	815.0	40.02	1091.7	126	25.938	1.995	4.062	174.4	93.87	1115.5
73	29.182	.4016	.8177	789.4	41.02	1092.1	127	25.828	2.049	4.172	170.0	94.87	1116.0
74	29.154	.4153	.8456	764.8	42.02	1092.6	128	25.714	2.105	4.286	165.8	95.87	1116.4
75	29.125	.4295	.8745	740.9	43.01	1093.0	129	25.596	2.163	4.404	161.6	96.86	1116.8
76	29.096	.4440	.9040	718.0	44.01	1093.5	130	25.478	2.221	4.522	157.64	97.86	1117.2
77	29.065	.4590	.9345	695.8	45.01	1093.9	131	25.356	2.281	4.644	153.74	98.86	1117.6
78	29.034	.4744	.9659	674.5	46.00	1094.4	132	25.230	2.343	4.770	149.91	99.86	1118.0
79	29.001	.4903	.9983	653.8	47.00	1094.8	133	25.101	2.406	4.899	146.22	100.86	1118.5
80	28.968	.5067	1.032	633.8	48.00	1095.3	134	24.971	2.470	5.029	142.66	101.85	1118.9
81	28.934	.5236	1.066	614.5	49.00	1095.7	135	24.837	2.536	5.163	139.17	102.85	1119.3
82	28.899	.5409	1.101	595.9	50.00	1096.2	136	24.700	2.603	5.300	135.80	103.85	1119.7
83	28.862	.5588	1.138	577.9	51.00	1096.6	137	24.560	2.672	5.440	132.51	104.85	1120.2
84	28.825	.5772	1.175	560.5	52.00	1097.1	138	24.417	2.742	5.583	129.33	105.84	1120.6
85	28.787	.5960	1.213	543.8	53.00	1097.6	139	24.271	2.814	5.729	126.22	106.84	1121.0

Table 3.—Properties of Saturated Steam at Pressures below Atmospheric Pressure.—*Cont.*

m, of Mercury. -in. Bar.)	Pressure, abs.		Specific Volume, cu. ft. per lb.	Heat Content				Pressure, abs.		Specific Volume, cu. ft. per lb.	Heat Content	
	Lb. per sq. in.	In. of Mercury. (Mercury at 32° F.)		Of Liquid, B.t.u.	Of Vapor, B.t.u.			Lb. per	In. of Mercury. (Mercury at 32° F.)		Liquid, B.t.u.	Of Vapor, B.t.u.
24.122	2.887	5.878	123.22	107.84	1121.4	173	16.935	6.417	13.065		140.85	1134.9
23.969	2.962	6.031	120.29	108.84	1121.8	174	16 634	6.565	13.366	57.03	141 85	1135.3
23.813	3.039	6.187	117.43	109.84	1122.2	175	16.326	6.716	13.674	55.82	142.86	1135.7
23.652	3.118	6.348	114.63	110.84	1122.6	176	16.015	6.869	13.985	54.66	143.86	1136.1
23.489	3.198	6.511	111.94	111.84	1123.0	177	15.697	7.025	14.303	53.52	144.86	1136.5
23.322	3.280	6.678	109.31	112.84	1123.5	178	15.373	7.184	14.627	52.41	145.86	1136.9
23.153	3.363	6.847	106.77	113.84	1123.9	179	15.046	7.345	14.954	51.34	146.87	1137.3
22.978	3.449	7.022	104.27	114.84	1124.3	180	14.710	7.510	15.290	50.28	147.87	1137.7
22.801	3.536	7.199	101.86	115.84	1124.7	181	14.368	7.678	15.632	49.25	148.87	1138.1
22.619	3.625	7.381	99.51	116.84	1125.1	182	14.019	7.849	15.981	48.24	149.87	1138.5
22.434	3.716	7.566	97.27	117.84	1125.5	183	13.663	8.024	16.337	47.26	150.87	1138.9
22.245	3.809	7.755	95.00	118.85	1126.0	184	13.303	8.201	16.697	46.30	151.87	1139.3
22.051	3.904	7.949	92.83	119.85	1126.4	185	12.934	8.382	17.066	45.36	152.87	1139.7
21.854	4.001	8.146	90.72	120.85	1126.8	186	12.560	8.566	17.440	44.45	153.88	1140.1
21.652	4.100	8.348	88.66	121.85	1127.2	187	12.179	8.753	17.821	43.56	154.88	1140.5
21.447	4.201	8.553	86.66	122.85	1127.6	188	11.790	8.944	18.210	42.69	155.88	1140.9
21.235	4.305	8.765	84.69	123.85	1128.0	189	11.395	9.138	18.605	41.84	156.89	1141.3
21.021	4.410	8.979	82.80	124.85	1128.4	190	10.992	9.336	19.008	41.01	157.89	1141.7
20.001	4.518	9.199	80.94	125.85	1128.8	191	10.581	9.538	19.419	40.19	158.90	1142.1
20.579	4.627	9.421	79.15	126.85	1129.2	192	10.161	9.744	19.839	39.40	159.90	1142.5
20.351	4.739	9.649	77.40	127.85	1129.6	193	9.734	9.954	20.266	38.62	160.90	1142.9
20.119	4.853	9.881	75.69	128.85	1130.0	194	9.298	10.168	20.702	37.86	161.91	1143.3
19.881	4.970	10.119	74.02	129.85	1130.4	195	8.856	10.385	21.144	37.12	162.91	1143.7
19.639	5.089	10.361	72.40	130.85	1130.9	196	8.408	10.605	21.592	36.40	163.92	1144.1
19.392	5.210	10.608	70.82	131.85	1131.3	197	7.952	10.829	22.048	35.69	164.92	1144.5
19.140	5.334	10.860	[illegible].28	132.85	1131.7	198	7.488	11.057	22.512	35.00	165.93	1144.9
18.883	5.460	11.117	67.78	133.85	1132.1	199	7.016	11.289	22.984	34.33	166.93	1145.2
18.621	5.589	11.379	66.31	134.85	1132.5	200	6.535	11.525	23.465	33.67	167.94	1145.6
18.354	5.720	11.646	64.89	135.85	1132.9	202	5.548	12.010	24.452	32.39	169.95	1146.4
18.081	5.854	11.919	63.49	136.85	1133.3	204	4.526	12.512	25.474	31.17	171.96	1147.2
17.814	5.990	12.196	62.14	137.85	1133.7	206	3.469	13.031	26.531	30.01	173.97	1148.0
17.519	6.130	12.481	60.81	138.85	1134.1	208	2.376	13.568	27.624	28.90	175.98	1148.7
17.230	6.272	12.770	59.52	139.85	1134.5	210	1.246	14.123	28.754	27.83	177.99	1149.5
						212	0.079	14.696	29.921	26.82	180 00	1150 2

Table 4.—Weight and Volume of Water Vapor per Pound of Dry Air

Pump Suction Temp. deg. F.	29-in. Vacuum		28.5-in. Vacuum		28-in. Vacuum		27.5-in. Vacuum		27-in. Vacuum		26.5-in. Vacuum		26-in. Vacuum	
	Lb.	Cu. Ft.	Lb.	Cu. Ft.	Lb.	Cu. Ft.	Lb.	Cu. Ft.	Lb.	Cu. Ft.	Lb.	Cu. Ft.	Lb.	Cu. Ft.
40	0.201	491.8	0.119	291.1	0.085	208.0	0.066	161.8	0.054	132.4	0.046	112.1	0.036	97.1
45	.268	544.7	.156	317.7	.110	223.9	.085	173.2	.070	141.2	.059	119.1	.051	103.0
50	.343	603.7	.192	338.3	.134	235.0	.102	180.6	.083	145.9	.070	122 7	.060	105.8
55	.482	688.8	.255	365.2	.174	248.5	.132	188.3	.106	151.6	.089	126.8	.076	109.0
60	.680	820.2	.333	401.0	.220	265.4	.164	198.3	.131	158 3	.109	131.7	.094	112.8
65	1.026	1048.0	.442	451.2	.282	287.5	.207	211.0	.160	163 2	.135	137.7	.115	117.3
70	1.764	1531.7	.606	525.5	.366	317.1	.262	227.1	.204	176.9	.167	144.8	.141	122.6
75	......	....	.873	645.7	.485	358.8	.336	248.4	.257	190.0	.208	153.8	.175	129.2
80	.....	....	1.376	870.6	.665	420.9	.437	277.6	.327	207.1	.261	165.1	.217	137.3
85	......	....	2.646	1437.2	.963	523 1	.589	319.8	.424	230 2	.331	179.9	.272	147.6
90	......	....	......	....	1.533	717.5	.822	384.8	.562	262.9	.427	199.7	.344	161.0
95	......	....	......	....	3.044	1230.6	1.233	498.4	.773	312.5	.563	227.6	.443	179.0
100	......	....	......	....	......	....	2.125	744.2	1.130	395.7	.770	269.5	.583	204.4
105	......	....	......	....	......	....	5.432	1653.5	1 848	562 6	1.114	339.0	.797	242.5
110	......	....	......	....	......	....	......	....	3.921	1040.3	1.775	470.9	1.147	304.3
115	......	....	......	....	......	....	......	....	......	....	3.697	857.3	1.860	431.3
120	......	....	......	....	......	....	......	....	......	....	......	....	3 887	790 0

QUANTITY OF CONDENSING WATER REQUIRED.—Let H = B.t.u. in exhaust steam at temperature T_s; h = B.t.u. in condensate at temperature T_c; W = weight of exhaust steam per hr., lb.; Q = quantity of condensing water required per hr., lb.; T_i = initial temperature of condensing water, deg. F.; T_c = temperature of condensate deg. F.; T_f = final temperature of condensing water. Heat surrendered by the steam = $W(H-h)$. Heat absorbed by the condensing water = $Q\,(T_f - T_i)$, and $Q = W(H-h)/(T_f - T_i)$. Theoretically $T_c = T_s$, but due to the presence of air and to imperfect mixing T_c should be taken from 10 to 15 deg. lower than T_s. $T_f = T_c$, whence

$$Q = W\,(H-h)/(T_c - T_i) \quad . \quad . \quad . \quad . \quad . \quad . \quad . \quad . \quad [1]$$

The value of $(H-h)$ may be taken as 943 B.t.u. when exhaust steam is assumed to contain 10% moisture with 29-in. vacuum, and 933 B.t.u. with 28-in. vacuum. For approximate calculations $(H-h)$ frequently is assumed as 1000 B.t.u., but the use of the actual values of H and h is more accurate.

Table 5.—Volume and Partial Pressure of 1 Lb. of Saturated Air at Various Vacua and Temperatures

Pump Suction Temperature, deg. F.	Pressure in lb. per sq. in., absolute. Volume in cubic feet.													
	29-in. Vacuum. 0.4910 lb. Pressure		28.5-in. Vacuum. 0.7365 lb. Pressure		28-in. Vacuum. 0.9820 lb. Pressure		27.5-in. Vacuum. 1.2275 lb. Pressure		27-in. Vacuum. 1.4730 lb. Pressure		26.5-in. Vacuum. 1.7185 lb. Pressure		26-in Vacuum. 1.9640 lb. Pressure	
	Pressure	Volume	Pressure	Volume	Pressure	Volume	Pressure	Volume	Pressure	Volume	Pressure	Volume	Pressure	Volume
40	0.3639	491.8	0.6148	291.1	0.8603	208.0	1.1058	161.8	1.3513	132.4	1.5968	112.1	1.8423	97.1
45	.3435	544.7	.5890	317.7	.8345	223.9	1.0800	173.2	1.3255	141.2	1.5710	119.1	1.8165	103.0
50	.3130	603.7	.5585	338.3	.8040	235.0	1.0495	180.0	1.2950	145.9	1.5405	122.7	1.7860	105.8
55	.2770	688.8	.5225	365.2	.7680	248.5	1.0135	188.3	1.2590	151.6	1.5045	126.8	1.7500	109.0
60	.2349	820.2	.4804	401.0	.7259	265.4	0.9714	198.3	1.2169	158.3	1.4624	131.7	1.7079	112.8
65	.1856	1048.0	.4311	451.2	.6766	287.5	.9221	211.0	1.1676	163.2	1.4131	137.7	1.6586	117.3
70	.1282	1531.7	.3737	525.5	.6192	317.1	.8647	227.1	1.1102	176.9	1.3557	144.8	1.6012	122.6
75	......	......	.3070	645.7	.5525	358.8	.7980	248.4	1.0435	190.0	1.2890	153.8	1.5345	129.2
80	......	......	.2298	870.6	.4753	420.9	.7208	277.6	0.9663	207.1	1.2118	165.1	1.4573	137.3
85	......	......	.1405	1437.2	.3860	523.1	.6315	319.8	.8770	230.2	1.1225	179.9	1.3680	147.6
90	......	......	......	......	.2840	717.5	.5295	384.8	.7750	262.9	1.0205	199.7	1.2660	161.0
95	......	......	......	......	.1671	1230.6	.4126	498.4	.6581	312.5	0.9036	227.6	1.1491	179.0
100	......	......	......	......	......	......	.2788	744.2	.5243	395.7	.7698	269.5	1.0153	204.4
105	......	......	......	......	......	......	.1266	1653.5	.3721	562.6	.6176	339.0	0.8631	242.5
110	......	......	......	......	......	......	......	......	.2030	1040.3	.4485	470.9	.6940	304.3
115	......	......	......	......	......	......	......	......	......	......	.2485	857.3	.4940	431.3
120	.	......	......	......	......	......	......	......	......	......	......	......	.2720	790.0

2. SURFACE CONDENSERS

The standard type of surface condenser comprises a chamber containing a large number of brass tubes, from $^5/_8$ to $1\,^1/_4$ in. diam., through which flows the condensing water. Steam flows around the tubes at high velocity and is condensed. Condensate, air, and other non-condensible gases are removed by a vacuum pump. For high vacuums, air and other gases are removed by a dry vacuum pump, and the condensate by a separate hotwell pump. The tubes are arranged so that the water makes usually one or two passes through them, although in some cases it may make as many as five passes. Air coolers usually are provided to further cool the entrained air and reduce its volume, and to condense any condensible gases present. These may be placed in the condenser chamber or be external to it.

CONDENSING SURFACE REQUIRED.—Let S = outside area of condenser tubes, sq. ft.; Q = quantity of condensing water, lb. per hr.; T_1 and T_2 = respectively, initial and final temperatures of condensing water, deg. F.; U = overall coefficient of heat transfer per sq. ft. per hr. per deg. F. temperature difference; t_m = mean temperature difference, deg. F., between temperature T_s of the steam and temperature of the condensing water. Then

$$\div \log_e \frac{}{(T_s - T_2)} \quad . \quad . \quad [3]$$

G. A. Orrok (*Trans.* A.S.M.E., xxxviii, 1916, p. 467) gives for the value of U the empirical formula

$$U = 325V^{0.6}/T_m^{1/8} \quad . \quad . \quad . \quad . \quad . \quad . \quad . \quad . \quad . \quad . \quad [4]$$

where V = velocity of water through the tubes, ft. per sec., other notations as above.

W. H. McAdams (*Trans.* A.S.M.E., xlviii, 1926, p. 1233) in an analysis of Orrok's data derived the empirical formula $1/U = r_v + r_t + r_s + (1/BV^{0.8})$, where r_v, r_t, and r_s = respectively, the individual thermal resistance, per unit area of condensing surface on the vapor side, of the tube itself, and of any scale or other deposit on both sides of the tube, and V = a coefficient depending on the dimensions of the tube and on the temperature. On a clean tube of Admiralty metal, 1 in. outside diameter, with a steam temperature of 136° F. and an average water temperature of 66° F. this formula reduces to

$$1/U = 0.000401 + (1/248V^{0.8}) \quad [5]$$

In practice V ranges from 4 to 10 ft. per sec. Corresponding values of $V^{0.6}$ and $V^{0.8}$ are

V = 4	4.5	5	5.5	6	6.5	7	7.5	8	8.5	9	9.5	10
$V^{0.6}$ = 2.31	2.47	2.63	2.77	2.93	3.07	3.21	3.35	3.48	3.61	3.74	3.86	3.98
$V^{0.8}$ = 3.03	3.33	3.62	3.91	4.19	4.47	4.74	5.01	5.27	5.54	5.80	6.16	6.31

Values of t_m and $t_m^{1/8}$ are given in Table 6.

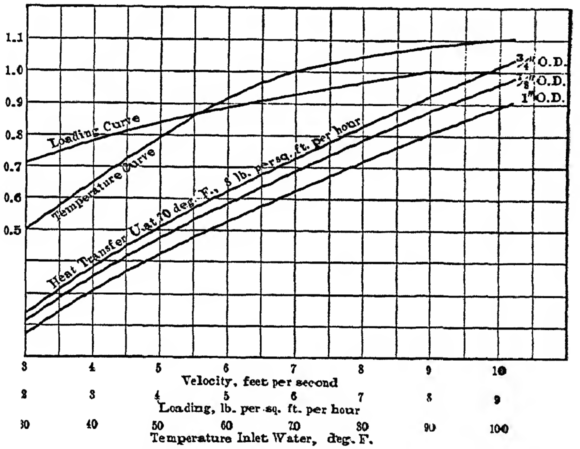


FIG. 6. Curves of Values of U for Surface Condensers

The curves, Fig. 6 (*Power*, Sept., 1932) have been adopted by the leading condenser manufacturers for values of U that can be used on a commercial basis for the design of surface condensers for use with steam turbines. They take into account the effect on the value of U of the velocity and temperature of the inlet water, tube diameter, and fouling of the tubes. Figs. 7 and 8 are curves adopted by the same manufacturers for determining loss of head in the tubes and water box losses, the latter including entrance and exit losses of the tubes. The curves of Fig. 6 give lower values of U than does the Orrok formula.

Number and Size of Tubes.—Geo. A. Orrok (*Trans.* A.S.M.E., xxxviii, 1916, p. 467) gives the following: Let N = number of passes in the condenser, n = number of tubes in one pass, d = outside diam. of tubes, in., t = thickness of tubes, in., l = total length of all passes, *i.e.*, total length of water travel in the condenser, ft., a = full area through tubes for the passage of water, sq. ft. Then

$$a = {}^1/_4\pi(d - 2t)^2/144 = 0.00545\,(d - 2t)^2, \quad [6]$$

$$V = Q/(3600 \times 62.4 \times na), \quad [7]$$

$$n = Q/1223V(d - 2t)^2. \quad [8]$$

The number of passes N depends on the ratio $(l/d) = R$. Values of R in practice range from 25 to 50, and $N = l/R$.

Tube diameters range from $^3/_4$ in. to $1\,^1/_4$ in. The smaller diameters are preferable as the rate of heat transfer is higher in a small tube. With 1-in. tubes, 1 sq. ft. of condenser cross-section will contain from 40 to 80 tubes, depending on tube spacing and

gement. If B = number of tubes per sq. ft., A = area of condenser cross-section, and L = length of condenser, ft. $A = Nn/b$, and $L = l/N$ + length of water boxes.

EXAMPLE.—Determine the dimensions of a surface condenser to condense 175,000 lb. of steam per hr. at a vacuum of 28.5 in.; inlet water temperature t_i, 75° F.; velocity, V = 8 ft. per sec. in the tubes; loading, 5 lb. per sq. ft. per hr. Assume temperature of condensate, t_c = 81.75° F., diam. d of tubes = $^3/_4$ in., thickness t, No. 18 B.W.G. = 0.049 in.

Solution.—From Table 3, T_s = 91.75; H = 1100.6; t_s = 91.75, h = 49.75.

$$Q = 175{,}000 \times (1106.5 - 49.75)/(81.75 - 75) = 27{,}244{,}250$$

From Fig. 6, $U = 762 \times 1.03 \times 0.88 = 690.7$. From formula [3],

$$t_m = \frac{81.75 - 75}{\log_e \{(91.75 - 75)/(91.75 - 81.75)\}} = 13.086$$

Surface = $S = Q\,(t_c - t_i)/Ut_m$ = 27,244,250 (81.75 − 75)/690.7 × 13.086 = 20,346 sq. ft.
No. of tubes = $n = Q/1223\ V\,(d - 2\,t)^2$ = 27,244,250/1223 × 8 (0.75 − 0.098)² = 6550
Length = l = 3.82 S/dn = 3.82 × 20,346/0.75 × 6550 = 15.82 ft.
If ratio of length to diameter of tubes R = 25,
Number of passes $N = l/R$ = 15.82/25 = 0.632, indicating one pass.
If tubes per sq. ft. of condenser cross section B = 110,
$A = Nn/B$ = 1 × 6550/110 = 59.55 sq. ft.

CONSTRUCTION DETAILS.—Shells usually are made of steel for land service, and of steel or copper for marine service. Shells are ribbed on the outside to resist collapsing pressure, and steel or copper shells are stiffened with angles.

Tubes usually are of brass, Muntz metal or Admiralty metal. The latter is used for salt water. A more recent development (1935) is arsenical copper. Tube thickness ranges from No. 16 to No. 20 B.W.G. and diameters from $^5/_8$ to 1 $^1/_4$ in. The larger tubes are used where water is dirty and liable to clog the tubes. Usual practice has been to set the tubes in brass ferrules screwed into the tube plates, and packed with lacing or wicking. In other cases, tubes were set in ferrules at one end and expanded into the tube sheet at the other end. Current practice (1935) tends toward expanding the tubes into the tube sheet at both ends, and allowing for expansion by flexure of the tubes, a special arrangement of the intermediate supporting plates being necessary. Another construction uses one fixed tube plate and one floating tube plate, an expansion joint being placed between the tube plate and the shell.

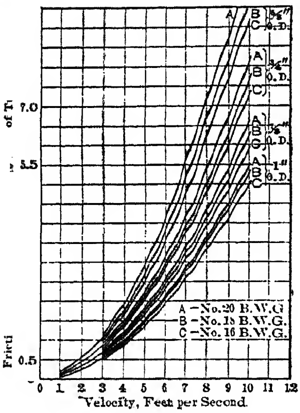


FIG. 7. Loss of Head in Tubes of Surface Condensers

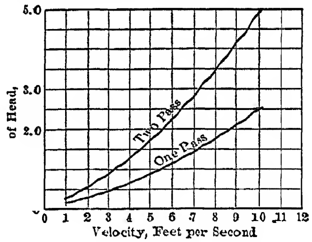


FIG. 8. Water Box Losses in Surface Condensers

Tube Spacing must provide room for ferrules, if these are used, and leave sufficient metal between holes for strength. The maximum number of tubes per square foot of tube sheet is

Pitch of tubes, in.	1	1 $^1/_{16}$	1 $^1/_8$	1 $^3/_{16}$	1 $^1/_4$	1 $^5/_{16}$	1 $^3/_8$	1 $^7/_{16}$	1 $^1/_2$	1 $^9/_{16}$	1 $^5/_8$
Tubes per sq. ft.	172	154	137	122	110	99	89	82	76	70	65

Table 6.—Values of $t_m^{1/8}$

t_m	$t_m^{1/8}$	t_m	$t_m^{1/8}$	t_m	$t_m^{1/8}$	t_m	$t_m^{1/8}$	t_m	$t_m^{1/8}$
2	1.095	10	1.334	18	1.435	26	1.503	34	1.554
4	1.189	12	1.364	20	1.454	28	1.517	36	1.565
6	1.251	14	1.391	22	1.472	30	1.530	38	1.576
8	1.298	16	1.414	24	1.488	32	1.542	40	1.586

Tube Sheets usually have been of rolled Muntz metal or brass. Where tubes are expanded into both tube sheets the Foster-Wheeler Corp. recommends the use of copper-bearing steel tube sheets. Thickness of tube sheets ranges from 1.1 to 1.5 tube diameters. Large areas should be stayed to prevent collapse.

Supporting Plates for the tubes, usually of cast iron, should be spaced about 60 to 70 tube diameters. The holes should be 1/8 in. larger diameter than the tubes.

Water Passages.—Dimensions of water connections should be based on a velocity of 8 ft. per sec., and of hotwell pump suction on a velocity of 4 ft. per sec. Air pump connection should be twice the size of the hotwell pump suction.

Setting.—Condensers should be set with a slope of 1 in. in 10 to 15 ft. to drain the tubes when shut down. A small hole should be drilled in the water box partition to enable the water boxes to drain. Circulating water outlet should be above the highest point of tubes. The air pump should be as close as possible to the condenser with short, direct pipe connections. Air suction should be connected at the coldest part of the condensers and steam flow directed to it.

Condenser design should be such that the steam penetrates every portion of the tube bank. Both longitudinal and transverse distribution must be considered. Dead air pockets, and blanketing of tubes by air are to be avoided. Steam passages should be of decreasing area toward the suction, to maintain practically uniform velocity of steam throughout the tube bank. The construction should be such as to cause the temperature of the condensate to be as close as practicable to the vacuum temperature. Baffles or special arrangements of tubes should protect the lower tubes from excessive condensate dripped from the upper ones.

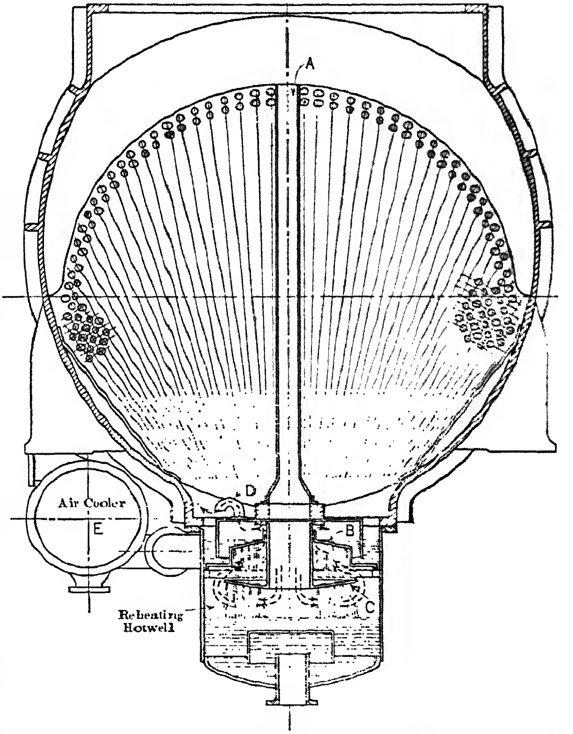


Fig. 9. Cross Section of Foster-Wheeler Single Pass Surface Condenser with Reheating Hotwell and External Air Cooler

Fig. 9 shows sections of a Foster-Wheeler condenser with a reheating hotwell and external air cooler. The tubes are arranged in straight lines diverging from the bottom. Steam enters at the top into a steam belt surrounding the upper part of the tube bank and flows through the straight lanes of diminishing area to the air off-takes at the bottom. A pipe *A* between the two tube banks conveys a portion of the steam to the hotwell, where it flows through two curtains of condensate falling from perforated plates *B* and *C*, being condensed thereby and raising the temperature of the condensate to within about 1° F. of vacuum temperature. Entrained air is vented through pipe *D*. The air off-takes are connected to the external air cooler *E*, the cooling surface of which ranges from 5 to 15% of the cooling surface of the main condenser. The water velocity in the air cooler is less than 1 ft. per sec.

Table 7 shows dimensions of standard small and medium size surface condensers

installations of 1000 kw. and over usually are designed for the conditions

to be

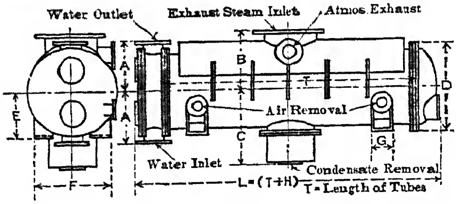


Fig. 10

–Dimensions of Two-pass Surface Condensers, Steel Plate Shell
(C. H. Wheeler Mfg. Co., Philadelphia, 1935)

Size	Dimensions, in. (See Fig. 10.)							
	A	*B*	*C*	*D*	*E*	*F*	*G*	*H*
20	15	18	22	25	11	24	6	13
22	17	20	24	28	12	24	6	15
25	19	21	26	30 1/4	13	24	6	16
28	20	24	27	33 1/2	16	26	6	20
30	21	24	34	36	18	30	8	23
33	23	23	36	39	18	32	8	25
36	24 1/2	30	40	42	20	36	10	28
39	27	29	48	46	21	36	10	28
42	28	33	54	49	22	42	10	30
45	31	34	54	52	24	42	12	31
48	33	36	54	55 1/2	26	48	12	35
54	36	40	58	62	28	50	12	38
60	40	48	66	68	32	54	15	42
66	42	48	64	74	35	60	18	46
72	45	54	69	80	38	66	18	50
76	48	58	73	84	40	66	18	53
80	48	60	75	88	42	72	18	58
86	54	66	83	96	44	84	18	60

AIR IN SURFACE CONDENSERS.—In addition to air entrained in the feedwater, surface condensers contain air due to leakage through joints subject to vacuum. This leakage may range from 1 to 25% of the volume of the feedwater, but average leakage is from 5 to 10%. G. A. Orrok (*Trans.* A.S.M.E. xxxiv, 1912, p. 713) shows the effect of leakage on the vacuum of three units as follows.

Leakage, cu. ft. of air per min. . .	10	20	30	40	50
Size of unit		Vacuum In. of Mercury			
8700 kw	28.25	27.85	27.4	27.0	26.6
8500 kw	28.4	28.1	27.8	27.5	27.2
4000 kw	28.7	28.35	28.0	27.65	. . . .

The air leakage was measured at atmospheric pressure. Mr. Orrok says that the leakage in condensers of from 20,000 to 50,000 sq. ft. should be less than 5 cu. ft. of free air per min. Sources of greater leakage should be found and corrected. The leakage should be added to the air entrained in the feedwater when determining the size of air pump.

The vacuum in condensers may be increased by the application of deaerators, which remove practically all the air entrained in the feedwater. The air in the condenser then is limited to that introduced by leakage.

CLEANLINESS FACTOR IN CONDENSER PERFORMANCE.—As deposits on the surface of condenser tubes greatly affect heat transmission through them, and hence affect the performance of the condenser, a determination of the cleanliness of the condenser is essential in evaluating its performance under given conditions. This is particularly important in tests to determine conformity with guarantees. P. H. Hardie and W. S. Cooper (*Trans.* A.S.M.E. RP-55-3, 1933) define cleanliness factor as the ratio of heat transmission of a fouled tube to that of a new tube through which water has passed only during the time necessary to start the test, the tubes being supplied with water at the same inlet temperature and subjected to the same conditions on the steam side. Also.

water velocities through the tubes must be equal. In determining cleanliness factor, representative groups of tubes throughout the condenser are selected, and independently supplied with water from the same source as the main supply. Inlet and exit temperatures are measured for each tube, and the velocity in each tube is calculated from the discharge of calibrated nozzles on each tube. The heat transmission of old and new tubes is computed by

$$U = (WC/S)\,\log_e\,(T_a/T_b) \quad . \quad . \quad . \quad . \quad . \quad . \quad . \quad . \quad . \quad [9]$$

where U = heat transmission, B.t.u. per hr. per sq. ft. per deg. F. log mean temperature

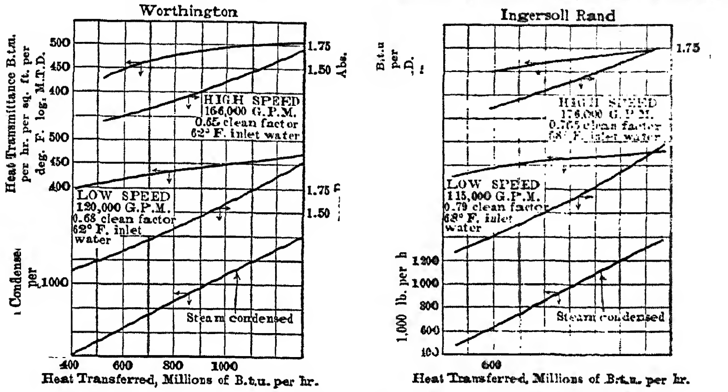


FIG. 11. Results of Tests of Two 101,000 sq. ft. Surface Condensers

difference; W = water flow, lb. per hr.; C = specific heat of water; S = surface area of tube, sq. ft.; T_a and T_b = initial and final temperature differences, respectively, deg. F. The average cleanliness factor for the entire condenser at any velocity is determined by averaging the heat transmission of all old tubes under test, and of all new tubes, and then computing the ratio of these two averages. The performance of the tubes under test should be referred to the saturated vapor temperature immediately adjacent to them, otherwise a higher cleanliness factor than the true one may be obtained.

The cleanliness factor varies inversely as some power of the velocity, that is

$$c = f(1/v)^n \quad . \quad . \quad . \quad . \quad . \quad . \quad [10]$$

where c = cleanliness factor; v = velocity in any unit; and n = velocity exponent for c. With velocities v_1 and v_2 and cleanliness factors c_1 and c_2,

$$n = \log\,(c_1/c_2) \div \log\,(v_1/v_2).$$

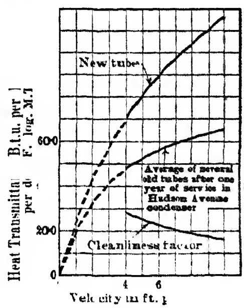


FIG. 12. Effect of Velocity on Variation of Cleanliness Factor and Heat Transmission in Old and New Tubes

See J. N. Landis and S. A. Tucker (*Trans.* A.S.M.E., FSP-56-3, 1934) for test data of two 101,000 sq. ft. condensers at the Hudson Ave. plant of the Brooklyn Edison Co. In these tests cleanliness factors were determined in accordance with the foregoing method. Results are summarized in Fig. 11. An explanation of the difference in cleanliness factors at high and low speeds in both sets of curves is given in Fig. 12, which shows that foul tubes do not respond to increases of velocity as do clean tubes. The increase of velocity reduces only the resistance to heat flow of the water film. This is a much smaller proportion of the total resistance in a foul tube than in a clean one.

Chlorination of the circulating water tends to improve the cleanliness factor, particularly if the water contains organic matter tending to promote the growth of algae. V. M. Frost and W. F. Rippe (*Trans.* A.S.M.E. FSP-53-10, 1931) describe experiments on chlorinating the water used in a condenser at the Kearney, N. J., station of the Public Service Elect. and Gas Co. Two 50,000-sq. ft. 2-pass condensers were used in the test

the water to one being with rubber plugs. T] 0.2 in. of mercury over

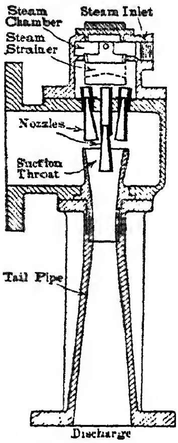


FIG. 13. Multiple Nozzle Ejector

320 to 72 lb. per day, depending on the ... Chlorine must be used with caution, for aside from its hazard as a toxic gas, it has a tendency, when present in excess, to cause corrosion of condenser tubes (see report of A.S.M.E. Research Comm. on Condenser Tubes, *Trans.* A.S.M.E., FSP-53-18, 1931). The report of the Kearny tests, however, states that no corrosion was evident after one year of operation with chlorine. The maximum residual chlorine at Kearny was found to be 0.3 part per million, at the condenser inlet.

3. AIR PUMPS

Air usually is removed from condensers by steam-jet pumps. These almost universally have replaced the mechanical pumps formerly used. Single-stage pumps are adapted to vacua under 26 in. of mercury; 2-stage pumps are used for vacua between 26 and 29 in.; 3-stage pumps for vacua above 29 in.

Multi-stage condensers usually have an inter-condenser between stages, and an after-condenser after the final stage. Condensate is used as cooling water. All of the heat in the steam jet thus is recovered, when the inter- and after-condensers are of the surface type. Fig. 13 shows a multiple nozzle ejector. Fig. 14 shows a 2-stage steam-jet air pump with surface inter- and after-condensers (Foster-Wheeler Corp.). Fig. 15 shows a 2-stage pump with jet inter-condenser.

CAPACITY OF STEAM-JET PUMPS depends on the area of jet in contact with the air. A pump with multiple nozzles will ...uch greater steam-jet surface than a single nozzle passing the same amount ... A single-stage pump will operate economically only within a compression ratio about 8 to 1. Beyond this range, high vacuum can be obtained only at the cost of

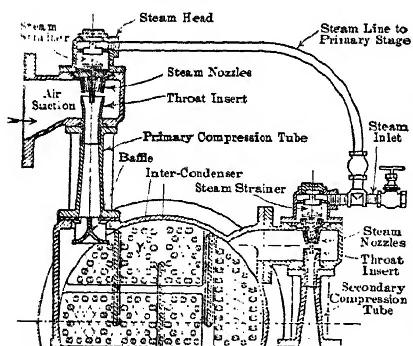


FIG. 14. Two-Stage Steam Jet Air Pump with Surface Inter- and After-Condensers

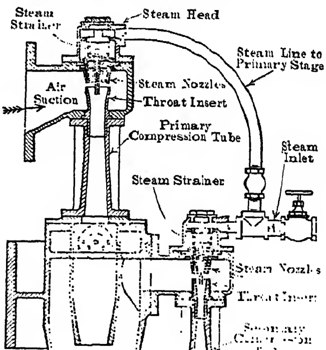


FIG. 15. Two-Stage Steam Jet Air Pump with Jet Inter-Condenser

excessive steam consumption, and a 2- or 3-stage pump is indicated. Fig. 16 shows the relative capacity of a 2-stage and a 3-stage Foster-Wheeler steam-jet pump using the ...ount of steam.

Commercial steam-jet air pumps range in capacity from about 10 lb. of dry air per hour to 100 lb. per hr. for single units. Twin units have double the capacity of single units.

The steam consumption may be taken approximately as 15 lb. per 1 lb. of dry air per hr., based on a mixture temperature of 7.5 deg. lower than the temperature of the steam at the condenser pressure.

Capacity of Air Pump Required for Surface Condensers is based generally (1935) on 29 in. vacuum and a temperature depression at the pump suction of 7.5° F. below the temperature of the steam at the pressure in the condenser. Under these conditions, the mixture of air and vapor in the condenser comprises about 2.2 lb. of vapor per 1 lb. of dry air. The air to be removed is that which leaks into the system. The quantity may cover a wide range, depending on the construction, tightness of joints and character of maintenance. Fig. 17 represents outside values that may be assumed for surface condensers, based on 29 in. vacuum and 7.5° temperature depression. In general, the actual leakage in well-maintained systems will be lower than the figures given by Fig. 17.

For conditions of vacuum and temperature other than those given above the necessary pump capacity can be estimated as follows: Let V_a = volume of atmospheric air leaking into condenser, per hr., cu. ft.; V_c = volume of air to be removed from condenser, cu. ft.; v_a = specific volume of air at temperature T_a, cu. ft. per lb. (see Table 10, page 1–08); v_s = specific volume of steam at temperature T_m, cu. ft. per lb. (see Table 3); T_a = temperature of atmosphere, deg. F.; T_m = temperature of mixture of air and vapor in condenser at pump suction, deg. F.; P_a = pressure of atmosphere; P_c = total pressure in condenser; P_s = pressure of vapor in condenser; all pressures in in. of mercury; W_a = weight of air to be removed per hr., lb.; W_s =

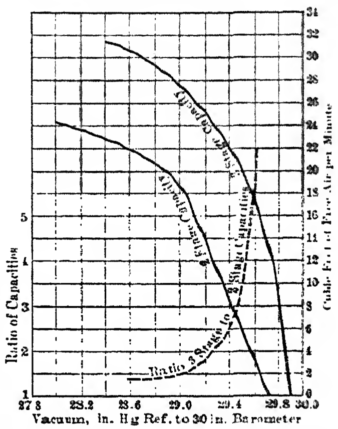


Fig. 16. Comparative Capacity Curve of 2- and 3-stage Steam-jet Pumps using same amount of Steam.

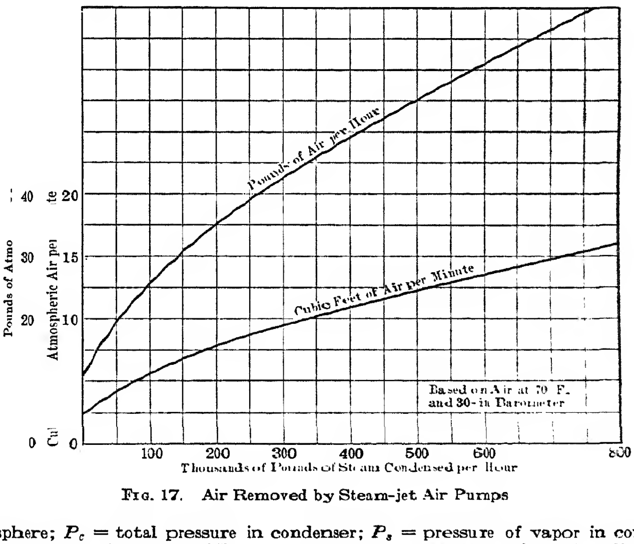


Fig. 17. Air Removed by Steam-jet Air Pumps

weight of vapor to be removed per hr., lb.; W_m = weight of mixture of vapor and air to be removed per hr., lb. Then

$$V_c = \frac{V_a P_a (T_m + 460)}{(P_c - P_s)(T_a + 460)} \quad [11]$$

$$W_a = V_a / v_a \quad [12]$$

$$W_s = V_c \times (1/v_s) \quad [13]$$

$$W_m = W_a + W_s \quad [14]$$

Values of V_c and W_s for the usual range of condenser pressures and temperatures are given in Tables 4 and 5.

Capacity of Air Pump for Jet Condensers may be estimated for preliminary calculations by assuming double the values given by Fig. 17. If the actual volume of air in the injection water is known, this figure should be used instead of the value from Fig. 17.

EXAMPLE—Required the capacity of pump to remove air from a surface condenser condensing 250,000 lb. of steam per hr. Temperature of atmosphere, 65° F.; barometer, 29.6 in.; vacuum, 28.5 in.; temperature of condensate, 87° F. *Solution.*—From Fig. 17, W_a = 39.0; the specific volume v_a of air at 65° F. is 13.22, whence V_a = 13.22 × 39.0 = 515.58 cu. ft. P_a = 29.6; P_c = 1.5; T_m = 87; T_a = 65. From Table 3, P_s = (30 − 28.708) = 1.292, or by interpolation in Table 5, $(P_c - P_s)$ = 0.208.

Then from formula [11],

$$V_c = \frac{515.58 \times 29.6 \times (87 + 460)}{0.208 \times (65 + 460)} = 76{,}445.5 \text{ cu. ft. per hr.}$$

Interpolating in Table 4, 1 lb. of dry air at 28.5 in. vacuum and 87° F. will have a volume of 1982 cu. ft. and contain 3.871 lb. of vapor. The weight of vapor to be removed, therefore, will be

$$W_s = (76{,}445.5/1982) \times 3.871 = 149.3 \text{ lb.}$$

$$W_m = 39.0 + 149.3 = 188.3 \text{ lb. per hr.}$$

COOLING EQUIPMENT

Water is cooled in a tower or pond by: *a*. Sensible heat flow from the water to the air, by radiation and conduction. *b*. Flow of latent heat to the air by evaporation of a portion of the water. The flow by radiation and conduction may range from 0 to 15% of the total flow. The evaporation of 1 lb. of water will absorb about 1000 B.t.u. from the body of the water to be cooled, thus reducing its temperature. The speed of evaporation depends on: 1. Difference in temperature of water and air. 2. Vapor tension differential between water and air. 3. Relative velocity of water and air.

A cooling tower is largely independent of climatic conditions. Ordinary air is not saturated, and when brought into contact with water will absorb sufficient moisture to saturate it. Even if saturated, and brought into contact with warmer water, the temperature of the air will be raised and a portion of the water will pass into the form of vapor and be absorbed by the air. The main body of water will be reduced in temperature by the loss of heat necessary to raise the temperature of the air and to evaporate the portion passing into vapor.

1. COOLING TOWERS

In a cooling tower, the water is broken into fine drops, or into thin films, and brought into contact with a current of air, induced by natural draft or mechanical means. For the various arrangements used to break up the water consult the catalogs of manufacturers.

TYPES OF COOLING TOWERS.—Cooling towers are classified by the method of producing the air current through them as:

1. Atmospheric Towers, depending on natural winds blowing horizontally across the tower. Louvres on the sides prevent water from being blown out of the tower. This type combines maximum thermal efficiency with minimum cost of construction, operation and maintenance. Its performance depends on the wind velocity, and therefore may vary widely.

2. Chimney Towers.—A chimney is superposed on the cooling surfaces, and draft is produced by the column of air in the chimney heated by its passage through the hot water in the tower. This type is well adapted to high temperatures and limited space.

3. Mechanical Draft Towers.—Draft is supplied by a fan, and may be either induced or direct. Since the air supply is absolutely controlled, this type is very flexible in operation, and the maximum of cooling capacity can be placed in a given area. The practical limit to the cooling capacity of a given mechanical draft tower is the horsepower required

to operate it. As the quantity of water is increased, both the volume of air required and the pressure against which the fan must operate increase. This necessitates larger fans and motors, and increased operating expense. A point is reached at which the increased cost of fans and motors, and the increased operating expense, is greater than the cost of building and operating a larger tower.

COOLING TOWER CALCULATIONS are based on the fact that the heat carried into the tower by the inflowing water and air is equal to the heat carried out by the outflowing water and air. Formulas covering the several variables involved are given below. The formulas and tabular data accompanying them should be used with caution, since the performance of a tower will vary widely with a relatively small change in any of the several variables involved. The tabular data are exact only for the conditions stated, and serve as a guide to the performance within a range of conditions. For other conditions than those stated, the performance should be calculated, and calculations should be based on the most unfavorable climatic conditions, *i.e.*, highest atmospheric temperatures and highest relative humidity. Table 1 gives the average temperature and humidity in July for a large number of places in the U. S., and will serve as a guide for conditions to be met. It should be borne in mind that the figures in Table 1 are averages and provision should be made for more unfavorable conditions. Thus, it would be wise to increase both temperature and humidity by 5%, and in certain localities by at least 10%. Otherwise, in very hot, humid weather, the tower may fail to give the desired range of cooling. Furthermore, while theoretically a tower can cool to the wet-bulb temperature, such a degree of cooling would be exceedingly costly. Cooling to within 5 deg. of wet-bulb

Table 1.—Meteorological Conditions throughout North America

30-Year Average Based on Records of U. S. Weather Bureau for July

(Copyright, 1929, by Cooling Tower Co., New York)

State	City				Wind, M per H	State	City		Wet d		ind, Miles per Hr.
Alabama...	Mobile......	80.5	73.5	79.0	6.0	Montana...	Helena......	66.9	53.4	43.0	
	Birmingham..	79.8	73.3	73.0	5.0	Nebraska..	Omaha......	76.5	66.5		7.0
Arizona....	Yuma.......	.0	71.0	36.0		New Jersey	Atlantic City.	72.5	69.0	84.0	8.0
	Phoenix.....	90.5	68.5	31.0		New York.	Albany......	72.0	65.5	71.0	7.7
Arkansas...	Little Rock..	80.	73.0	72.0	5.0		Buffalo......	70.2	64.0	71.0	10.
	Ft. Smith....	80.	72.5	68.0	5.0		New York...	73.	0	71.0	9.1
California...	San Francisco	57.3	54.3	82.0	14.0		Rochester....	70.4	63.2	67.0	7.1
	Sacramento..	72.4	60.6	51.0	9.7	N. Carolina	Charlotte....	77.8	70.3	69.0	5.0
	Los Angeles..	67.4	61.6	72.0	4.5		Wilmington..	78.7	73.		
Colorado...	Denver......	71	57.8	45.0	7.5	N. Dakota.	Bismarck....	70.	.2	61.0	
	Pueblo......	72.6	58.0	44.0	7.0	New Mex..	Santa Fe....	68.7	55.2	46.0	6.1
Conn......	New Haven..	71.9	65.9	73.	8.0	Ohio......	Cincinnati...	77.7	68.7	63.0	6.6
Dist. Col...	Washington..	76.8	69.3	69.	5.3		Cleveland....	72.5	65.8	70.0	11.7
Florida.....	Jacksonville..		75.0	75.	8.0		Columbus....	75.0	66.5	64.0	8.7
	Tampa......		75.0	79.	6.0	Oklahoma..	Okla. City...	79.0	0	64.0	9.0
Georgia....	Atlanta......		69.6	67.	8.6	Oregon....	Portland.....	66.3	58.0	60.0	7.9
	Savannah....		75.5	79.	6.4	Pa........	Erie.........	71.8	64.8	69.0	9.0
Illinois.....	Cairo........		72.4	75.	6.2		Philadelphia..	75.8	67.8	66.0	9.4
	Chicago.....		65.8	71.	15.1		Pittsburgh...	74.6	67.	68.0	5.2
	Springfield...		68.0	64.	6.6	S. Carolina	Charleston...	81.3	75.3	76.0	9.8
Indiana...	Indianapolis..	76.4	68.0	65.0	8.2	S. Dakota..	Yankton.....	74.6	66.8	66	6.5
Iowa......	Davenport...	75	67.2	65.0	7.4	Tennessee..	Chattanooga.	77.8	70.8	71.0	5.2
	Des Moines..	75	67.0	66.0	7.1		Memphis....	80.7	72.7	68.0	7.4
Kansas....	Wichita......	78	69.5			Texas.....	Galveston...	83.0	77.0	76.0	10.0
Kentucky..	Louisville....	78.6	69.6				Ft. Worth...	82.5	69.5	51	10.0
Louisiana...	New Orleans.	81.3	75.3	76.	6.5	Utah......	SaltLake City	76.2	58.2	34.0	6.3
	Shreveport...	82.	75.0	72.	5.0	Vermont...	Burlington...	68.2	63.2	76.0	
Maine.....	Eastport.....	59.	56.3	81.0		Virginia...	Norfolk......	78.4	72.6	74.0	
	Portland.....	68.	62.0	71.0			Richmond...	79.2	72.0	70.0	
Maryland..	Baltimore....	77.	69.6	70.0	6.6	Wash.....	Seattle......	63.3	56.3	64.0	5.5
Mass.......	Boston......	71.	64.8	70.0	9.3		Spokane.....	68.8	55.0	41.0	....
Michigan...	Detroit......	72.	65.1	59.0	9.0	W. Virginia	Parkersburg..	74.9	67.5	68.0	4.0
	Grand Rapids	72.6	65.4	68.0	9.0	Wisconsin..	Milwaukee...	69.7	7	77.0	9.8
Mississippi..	Vicksburg....	80.4	74.4	75.0	6.0	Wyoming..	Cheyenne....	67.4	54.6	46.0	8.0
Minnesota..	St. Paul.....	72.	64.5	67.0	7.2	Canada....	Montreal....	69.5	0	7[illegible].0	11.3
	Duluth......	66.	60.0	71.0	11.0		Toronto.....	68.5	62.3	70.5	7.9
Missouri...	Kansas City..	76	68.9	0	7.5		Winnipeg....	66.0	0	77.5	11.3
	St. Louis....		70.6	0	8.2						

temperature represents good practice. Manufacturers of cooling towers should be consulted before a final decision is reached as to the size and type of tower advisable for any specified conditions.

ATMOSPHERIC COOLING.—An empirical formula for average temperature reduction in summer weather is

where T_1 = temperature of entering water; T_2 = temperature of leaving water; t_1 = temperature of dry bulb thermometer; t_2 = temperature of wet bulb thermometer, all in deg. F. A more exact formula, including wind variation, given by the Cooling Tower Co. is

$$T_{\cdot} = \left[\left\{ \frac{1.9385}{\quad} + 1. \right\} - \quad t_2 \right] \div \left[\left\{ \left(\quad 1.9385 \quad + 1.53 \right) \frac{\quad}{100} \right. \right. \qquad [2]$$

where, in addition to above notation, R = cooling range, deg. F.; A/W = ratio of air to water by weight. Let V = wind velocity, mi. per hr.; J = a constant depending on the of the tower; H = height of tower, ft.; B = width of distributing deck, ft.;

Table 2.—Values of $f(A/W)$

A/W	Values of R, deg. F.										
		10	15	20	25	30	35	40	45	50	55	
2.667	11.23	22.46	33.69	44.92	56.15	67.38	78.61	89.84	101.07	112.30	123	134.76
2.833	11.02	22.03	33.04	44.06	56.08	66.09	77.11	88.12	99.14	110.15	121	132.18
3.000	10.83	21.65	32.48	43.30	54.13	64.95	75.78	86.60	97.43	108.25	119	129.90
3.167	10.66	21.31	31.97	42.62	53.28	63.93	74.59	85.24	95.90	106.55	117	127.86
3.333	10.51	21.01	31.52	42.02	52.53	63.03	73.54	84.04	94.55	105.05	115	126.06
3.500	10.37	20.73	31.10	41.46	51.83	62.19	72.56	82.92	93.29	103.65	114	124.38
3.667	10.24	20.48	30.72	40.96	51.20	61.44	71.68	81.92	92.16	102.40	112	122.88
3.833	10.13	20.25	30.38	40.50	50.63	60.75	70.88	81.00	91.13	101.25	111	121.50
4.000	10.02	20.04	30.06	40.08	50.10	60.12	70.14	80.16	90.18	100.20	110	120.24
4.167	9.93	19.85	29.78	39.70	49.63	59.55	69.48	79.40	89.33	99.25	109	119.10
4.333	9.83	19.66	29.49	39.32	49.15	58.98	68.81	78.64	88.47	98.30	108	117.96
4.500	9.76	19.51	29.27	39.02	48.78	58.53	68.29	78.04	87.80	97.55	107	117.06
4.667	9.68	19.35	29.03	38.70	48.38	58.05	67.73	77.40	87.08	96.75	106	116.10
4.833	9.61	19.21	28.82	38.42	48.03	57.63	67.24	76.84	86.45	96.05	105	115.26
5.000	9.54	19.08	28.62	38.16	47.70	57.24	66.78	76.32	85.86	95.40	104	114.48
5.167	9.48	18.96	28.44	37.92	47.40	56.88	66.36	75.84	85.32	94.80	104	113.76
5.333	9.42	18.84	28.26	37.68	47.10	56.52	65.94	75.36	84.78	94.20	103	113.04
5.500	9.37	18.73	28.10	37.46	46.83	56.19	65.56	74.92	84.29	93.65	103	112.38
5.667	9.32	18.63	27.95	37.26	46.58	55.89	65.21	74.52	83.84	93.15	102	111.78
5.833	9.27	18.53	27.80	37.06	46.33	55.59	64.86	74.12	83.39	92.65	101	111.18
6.000	9.22	18.44	27.66	36.88	46.10	55.32	64.54	73.76	82.98	92.20	101	110.64
6.167	9.18	18.36	27.54	36.72	45.90	55.08	64.26	73.44	82.62	91.80	100	110.16
6.333	9.14	18.27	27.41	36.54	45.68	54.81	63.95	73.08	82.22	91.35	100	109.62
6.500	9.10	18.20	27.30	36.40	45.50	54.60	63.70	72.80	81.90	91.00	100	109.20
6.667	9.06	18.12	27.18	36.24	45.30	54.36	63.42	72.48	81.54	90.60		108.72
6.833	9.03	18.05	27.08	36.10	45.13	54.15	63.18	72.20	81.23	90.25		108.30
7.000	9.00	17.99	26.99	35.98	44.98	53.97	62.97	71.96	80.96	89.95		107.94
7.333	8.93	17.86	26.79	35.72	44.65	53.58	62.51	71.44	80.37	89.30		107.16
7.667	8.88	17.75	26.63	35.50	44.38	53.25	62.13	71.00	79.88	88.75		106.50
8.000	8.83	17.65	26.48	35.30	44.13	52.95	61.78	70.60	79.43	88.25		105.90
8.333	8.78	17.55	26.33	35.10	43.88	52.65	61.43	70.20	78.98	87.75		105.30
8.667	8.73	17.46	26.19	34.92	43.65	52.38	61.11	69.84	78.57	87.30	96.03	104.76
9.000	8.69	17.38	26.07	34.76	43.45	52.14	60.83	69.52	78.21	86.90	95.59	104.28
9.333	8.66	17.31	25.97	34.62	43.28	51.93	60.59	69.24	77.90	86.55	95.21	103.86
9.667	8.62	17.24	25.86	34.48	43.10	51.72	60.34	68.96	77.58	86.20	94.82	103.44
10.00	8.59	17.17	25.76	34.34	42.93	51.51	60.10	68.68	77.27	85.85	94.44	103.02
10.50	8.54	17.08	25.62	34.16	42.70	51.24	59.78	68.32	76.86	85.40	93.94	102.48
11.00	8.50	17.00	25.50	34.00	42.50	51.00	59.50	68.00	76.50	85.00	93.50	102.00
11.50	8.46	16.92	25.38	33.84	42.30	50.76	59.22	67.68	76.14	84.60	93.	101.52
12.00	8.43	16.86	25.29	33.72	42.15	50.58	59.01	67.44	75.87	84.30	92.	101.16
12.50	8.40	16.79	25.19	33.58	41.98	50.37	58.77	67.16	75.56	83.95	92.	100.74
13.00	8.37	16.73	25.10	33.46	41.83	50.19	58.56	66.92	75.29	83.65	92.02	100.38
14.0	8.32	16.63	24.95	33.26	41.58	49.89	58.21	66.52	74.84	83.15		99.78
15.0	8.27	16.54	24.81	33.08	41.35	49.62	57.89	66.16	74.43	82.70		99.24
17.5	8.23	16.46	24.69	32.92	41.15	49.38	57.61	65.84	74.07	82.30		98.76
20.0	8.11	16.21	24.32	32.42	40.53	48.63	56.73	64.84	72.94	81.05		97.26
22.5	8.06	16.12	24.18	32.24	40.30	48.36	56.42	64.48	72.54	80.60		96.72
25.0	8.02	16.04	24.06	32.08	40.10	48.12	56.14	64.16	72.18	80.20		96.24
27.5	7.99	15.97	23.96	31.94	39.93	47.91	55.90	63.88	71.87	79.85		95.82
30.0	7.96	15.91	23.87	31.82	39.78	47.73	55.69	63.64	71.60	79.55		95.46
33.333	7.93	15.85	23.78	31.70	39.63	47.55	55.48	63.40	71.33	79.25		95.10

F = flow per sq. ft. of horizontal deck area, gal. per min. Then $A/W = VJ$; $J = 0.8H/FB$. Values of A/W range from 2.66 to 33.33; of R from 0 to 60; of t_2 from 60 to 80; of T_2 from 60 to 91.4; of F from 1.5 to 3.5. Table 2 gives values of

Table 3 gives capacities of atmospheric cooling towers for steam plants operating at various degrees of vacuum. Table 4 gives capacities and dimensions of similar towers for cooling the jacket water of Diesel engines. Table 5 gives dimensions and capacities

Table 3.—Capacities of Atmospheric Cooling Towers for Steam Power Plants

Based on 70° F. wet-bulb and wind velocity of 5 miles per hour
(Cooling Tower Co., New York, 1935)

Vacuum, in. of Mercury	Temperature of Vacuum, deg. F.	Differential between Temperature of Vacuum and Temperature of Water at Condenser Outlet, deg. F.															
		5		10		15		20		5		10		15		20	
		Loading 1 1/2 gal. per sq. ft. of active horizontal area								Loading 2 1/2 gal. per sq. ft. of active horizontal area							
		Lb. Cooling Water per lb. of Steam	Gal. per min. per 1000 lb. of Steam	Lb. of Cooling Water per lb. of Steam	Gal. per min. per 1000 lb. of Steam	Lb. of Cooling Water per lb. of Steam	Gal. per min. per 1000 lb. of Steam	Lb. of Cooling Water per lb. of Steam	Gal. per min. per 1000 lb. of Steam	Lb. of Cooling Water per lb. of Steam	Gal. per min. per 1000 lb. of Steam	Lb. of Cooling Water per lb. of Steam	Gal. per min. per 1000 lb. of Steam	Lb. of Cooling Water per lb. of Steam	Gal. per min. per 1000 lb. of Steam	Lb. of Cooling Water per lb. of Steam	Gal. per min. per 1000 lb. of Steam
28	101	56	112	69	138	94	188	138	276	60	120	88	176	119	238	179	358
27	116	34	68	39	78	46	92	55	110	39	78	46	92	55	110	68	136
26	126	26	52	30	60	34	68	39	78	30	60	34	68	39	78	46	92
25	134	22	44	25	50	27	54	31	62	25	50	28	56	31	62	36	72
24	141	20	40	21	42	24	48	25	50	22	44	24	48	26	52	30	60
23	147	18	36	19	38	21	42	23	46	19	38	21	42	23	46	26	52
22	152	16	32	18	36	19	38	21	42	18	36	19	38	21	42	23	46

Table 4.—Dimensions and Capacities of Atmospheric Cooling Towers for Diesel Engines

Based on Wind Velocity of 5 miles per hour
(Cooling Tower Co., New York, 1935)

Overall Dimensions, ft.			Shipping Weight, lb.		Open System †			Closed System, 75° Wet-bulb			
								Shell and Tube Type ‡		Bent Tube Type ††	
					Wet-bulb Temp., deg. F.						
					70	75	80				
Width	Length	Height	Tower	Basin	Diesel Horsepower			Diesel Horse-power	Gal. per min. over Tower	Diesel Horse-power	Gal. per min. over Tower
8*	8*	9*	1600	300	180	160	135	50	23	......	.....
8*	8*	12*	2400	400	240	210	180	68	30	......	.....
8*	8*	16*	3000	400	320	280	240	90	40	......	.....
14	13	16	6600	1300	540	475	400	150	68	180	54
14	13	22	8800	1300	720	630	540	200	90	240	72
14	19	16	9000	2000	1080	940	810	300	135	360	108
14	19	22	12000	2000	1440	1260	1080	400	180	480	144
14	25	16	11500	2600	1620	1420	1200	450	200	540	162
14	25	22	15200	2600	2160	1890	1620	600	270	720	216
14	31	22	18400	3200	2880	2520	2160	800	360	960	288
14	37	22	21600	3900	3600	3150	2700	1000	450	1200	360
14	43	22	24800	4500	4320	3780	3240	1200	540	1440	432
19	25	32	26900	3300	4500	3940	3380	1250	562	1500	450
19	31	32	32000	4200	6000	5250	4500	1670	750	2000	600
19	37	32	37000	5000	7500	6550	5650	2090	940	2500	750
19	43	32	42000	5900	9000	7900	6750	2500	1125	3000	900

* Built of steel; all other sizes built of redwood.
† Cooling range, 130°–100° F.; water over jacket and tower, gal. per min., = 0.2 × Hp.
‡ Jacket water, 130°–110° F.; gal. per min. = 0.3 × Hp. Raw tower water, 97°–84° F.; gal. per min. = 0.45 × Hp.
†† Jacket water, 130°–110° F.; gal. per min. = 0.3 × Hp. Raw tower water, 99°–83° F.; gal per min. = 0.3 × Hp.

at various rates of loading per sq. ft. of horizontal deck area of atmospheric cooling towers.

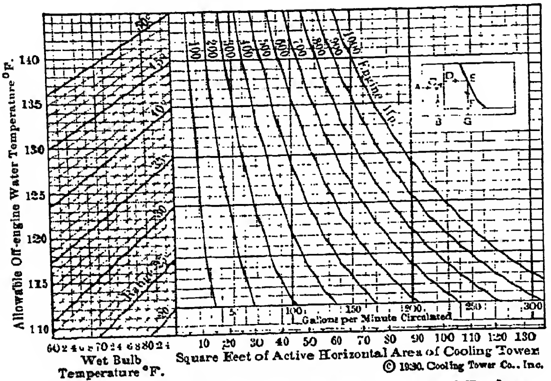


FIG. 1. Atmospheric Cooling Tower Chart for Diesel Engines

Fig. 1 is a chart for determining the size of tower for cooling the jacket water of Diesel engines for various atmospheric conditions, cooling ranges and engine horsepowers. The method of using the chart is shown by the inset. Find the intersection C of the allowable off-engine water temperature and of the wet bulb temperature B. Follow to the right to D, the diagonal range line from this intersection and then horizontally to the curve of engine horsepower E. At the bottom of the chart read the required gallons per minute F, and the square feet of active horizontal area G.

; 5.—Capacities and Dimensions of Atmospheric Cooling Towers

(Cooling Tower Co., New York, 1935)

Active Horizontal Area, sq. ft.*	Sq. Ft. per min. of Active Horizontal Area — Loading, gal. per min. 1.5	2.5	3.5	Overall Dimensions, ft. Width	Length	Height	Shipping Weight, lb. Tower	Basin	Type of Construction
16	24	40	56	8	8	9	1600	300	Steel
16	24	40	56	8	8	12	2400	400	"
16	24	40	56	8	8	17	3000	400	"
32	48	80	112	8	12	12	3200	700	"
32	48	80	112	8	12	17	4400	700	"
36	54	90	126	14	13	23	10500	700	"
36	54	90	126	14	13	23	8800	1300	Wood
48	72	120	168	8	16	17	5700	900	Steel
64	96	160	224	8	20	17	7000	1100	"
72	108	180	252	14	19	23	14300	1200	"
72	108	180	252	14	19	23	12000	2000	Wood
80	120	200	280	8	24	17	8300	1300	Steel
96	144	240	336	8	28	17	9600	1500	"
108	162	270	378	14	24	23	18000	1600	"
108	162	270	378	14	25	23	15200	2600	Wood
112	168	280	392	8	32	17	10900	1700	Steel
128	192	320	448	8	36	17	12200	1900	"
144	216	360	504	8	40	17	13500	2100	"
144	216	360	504	14	30	23	21800	2100	"
144	216	360	504	14	31	23	18400	3200	Wood
150	225	375	525	20	18	32	26200	2000	Steel
150	225	375	525	19	19	32	21800	2500	Wood
160	240	400	560	8	44	17	14800	2300	Steel
180	270	450	630	14	36	23	25600	2500	"
180	270	450	630	14	37	23	21600	3900	Wood
216	324	540	756	14	42	23	29400	3000	Steel
216	324	540	756	14	43	23	24800	4500	Wood
225	338	563	788	20	24	32	32600	2800	Steel
225	338	563	788	19	25	32	26900	3300	Wood
300	450	750	1050	20	29	32	38200	3600	Steel
300	450	750	1050	19	31	32	32000	4200	Wood
375	563	938	1313	20	35	32	33300	4400	Steel
375	563	938	1313	19	37	32	37000	5000	Wood
450	675	1125	1575	20	41	32	37600	5200	Steel
450	675	1125	1575	19	43	32	42000	5900	Wood
525	788	1313	1838	20	47	32	43100	6000	Steel
525	788	1313	1838	19	49	32	47100	6800	Wood

* Active horizontal area is the horizontal area occupied by each deck.

AIR REQUIRED IN MECHANICAL DRAFT COOLING TOWERS.—Let A = pounds of air required per pound of entering water; T_1 = temperature of entering water; T_2 = temperature of leaving water; t_1 = temperature of entering air; t_2 = temperature of leaving air, all in deg. F.; m_1 and m_2 = moisture in 1 lb. of saturated air, at temperatures t_1 and t_2 respectively (see Table 6); e_1 and e_2 = total heat, B.t.u., per lb. of water vapor, at temperatures t_1 and t_2 respectively; z = relative humidity of entering air. H_w = heat, B.t.u., entering tower in the water, per 1 lb. of water; H_a = heat, B.t u , entering tower in the air, per 1 lb. of water; h_w = heat, B.t.u., leaving tower in the water per 1 lb. of water; h_a = heat, B.t.u., leaving tower in the air, per 1 lb. of water.

$$H_w = (T_1 - 32);\ H_a = 0.2375\,A\,(t_1 - 32) + A\,(m_2 e_2);$$

$$h_w = 1 - (m_2 - m_1 z)\,(T_2 - 32);\ h_a = 0.2375\,A\,(t_2 - 32) +$$

Neglecting radiation losses, $H_w + H_a = h_w + h_a$, whence

$$A = \frac{[\text{illegible}]}{0.2375\,(t_2 - t_1) + m_2 e_2 - m_1 e_1 z - (m_2 - m_1 z)\,(T_2 - 32)} \qquad [3]$$

This formula assumes the leaving air to be saturated. If it is not saturated, the value of m_2 must be multiplied by a factor z_2 = relative humidity of the leaving air. Table 7 has been calculated from this formula. See also paper by C. S. Robinson, *Trans.* A.S.M.E., vol. xliv, p. 669, for formulas that take into account the diffusion of water vapor.

EXAMPLE.—Required the air to cool 475,000 lb. of water from 110° to 80° F. Temperature of entering air, 70° F., relative humidity 70%. Temperature of leaving air 100°; relative humidity 98%.

Solution.—T_1 = 110; T_2 = 80; t_1 = 70; t_2 = 100; z_1 = 0.70; z_2 = 0.98; from Table 6, m_1 = 0.1553; m_2 = 0.4305; from steam tables, e_1 = 1090.8; e_2 = 1104.2. Then $m_1 z_1$ = 0.01087; $m_2 z_2$ = 0.04044;

$$A = \frac{110 - 80}{0.2375(100 - 70) + (1104.2 \times 0.04044) - (1090.8 \times 0.01087) - (0.04044 - 0.01087)\,(80 - 32)}$$
= 0.7257 lb. of air per lb. of water.

From Table 10 p. 1-08, 1 lb. of dry air at 70° F. = 13.35 cu. ft. and 1 lb. of saturated air = 13.69 cu. ft. Volume of vapor = 13.69 − 13.35 = 0.34 cu. ft. Volume of 1 lb. of air at 70° F. and 70% relative humidity = 13.35 + (0.70 × 0.34) = 13.59 cu. ft. Cu. ft. of air required per lb. of water = 13.59 × 0.7257 = 9.862. Total air required = 0.7257 × 475,000 = 344,707.5 lb. = 9.862 × 475,000 = 4,684,450 cu. ft.

WATER EVAPORATED IN A COOLING TOWER per pound of air is represented by the expression $(m_2 - m_1 z)$ in formula [3] above. If A = lb. of air per lb. of water circulated (see Table 7), W = lb. of water circulated per hr. and w = amount of water evaporated per hr., $w = AW\,(m_2 - m_1 z)$. This represents the quantity of make-up water required for surface condensers. Table 8 has been calculated from the above formula.

ATMOSPHERIC COOLING TOWER CAPACITY FOR REFRIGERATION.—Cooling towers are used not only for refrigerating plants for ice-making and food cooling, but also for those used in connection with air conditioning equipment. The heat discharged per minute per ton of refrigeration, which must be eliminated by the cooling tower is:

Compression Plants.—Straight refrigeration, 250 B.t.u.; distilled water ice, 1250 B.t.u.;

Table 6.—Amount of Water Vapor in 1 Lb. of Saturated Air at Atmospheric Pressure

Deg. F.	Moisture, lb.	Deg. F.	Moisture, lb.	Deg. F.	Moisture, lb.	Deg. F.	Moisture, lb.
32	0.00377	55	0.00912	78	0.02038	101	0.04257
33	.00392	56	.00946	79	.02107	102	.04389
34	.00408	57	.00981	80	.02178	103	.04526
35	.00425	58	.01018	81	.02251	104	.04667
36	.00442	59	.01055	82	.02326	105	.04807
37	.00460	60	.01093	83	.02404	106	.04961
38	.00479	61	.01133	84	.02484	107	.05114
39	.00498	62	.01174	85	.02566	108	.05267
40	.00518	63	.01216	86	.02651	109	.05428
41	.00538	64	.01260	87	.02738	110	.05598
42	.00559	65	.01306	88	.02828	111	.05767
43	.00581	66	.01351	89	.02920	112	.05937
44	.00604	67	.01400	90	.03015	113	.06121
45	.00628	68	.01450	91	.03113	114	.06306
46	.00652	69	.01501	92	.03213	115	.06490
47	.00677	70	.01553	93	.03316	116	.06690
48	.00703	71	.01608	94	.03423	117	.06881
49	.00731	72	.01664	95	.03533	118	.07089
50	.00758	73	.01721	96	.03645	119	.07304
51	.00787	74	.01781	97	.03761	120	.07518
52	.00816	75	.01842	98	.03879	125	.08701
53	.00848	76	.01905	99	.04002	130	.1002
54	.00379	77	.01970	100	.04127	135	.1154

› system, condensing engines, 800 B.t.u.; distilled water condensers and flat coolers, 1000 B.t.u.

Absorption Plants.—Straight refrigeration, 550 B.t.u.; distilled water ice, 950 B.t.u.; distilled water condensers and flat coolers, 400 B.t.u.

The Cooling Tower Co. recommends 2 sq. ft. of active horizontal deck area per ton of

Table 7.—Pounds of Air per Pound of Circulating Water

Outflowing air saturated

T_1	T_2	t_2	$t_1 = 50°$			$t_1 = 70°$			$t_1 = 80°$		
			$z=0.5$	$z=0.7$	$z=0.9$	$z=0.5$	$z=0.7$	$z=0.9$	$z=0.5$	$z=0.7$	$z=0.9$
100	70	92	0.7470	0.7776	0.8108	0.9617	1.0743	1.2168	1.1774	1.4371	1.8440
		88	.8554	.8958	.9402	1.1493	1.3139	1.5336	1.4715	1.9008	2.6841
		84	.9860	1.0400	1.1004	1.3981	1.6494	2.0110	1.9055	2.6933	4.5922
	80	92	0.5015	0.5220	0.5443	0.6462	0.7217	0.8172	0.7915	0.9659	1.2389
		88	.5743	.6013	.6311	.7723	.8827	1.0299	.9894	1.2778	1.8034
		84	.6619	.6981	.7385	.9395	1.1081	1.3505	1.2817	1.8112	3.0862
	90	92	0.2525	0.2628	0.2740	0.3257	0.3636	0.4117	0.3991	0.4869	0.6243
		88	.2892	.3028	.3177	.3892	.4448	.5188	.4990	.6442	.9088
		84	.3333	.3514	.3717	.4736	.5584	.6802	.6466	.9135	1.5557
T_1	T_2	t_2	$t_1 = 50°$			$t_1 = 70°$			$t_1 = 90°$		
			$z=0.5$	$z=0.7$	$z=0.9$	$z=0.5$	$z=0.7$	$z=0.9$	$z=0.5$	$z=0.7$	$z=0.9$
110	70	102	0.7243	0.7456	0.7683	0.8647	0.9304	1.0070	1.1877	1.4665	1.9163
		98	.8202	.8477	.8771	1.0050	1.0949	1.2026	1.4695	1.9214	2.7749
		94	.9324	.9681	1.0066	1.1788	1.3045	1.4603	1.8735	2.6760	4.6809
	80	102	0.5472	0.5633	0.5803	0.6536	0.7032	0.7610	0.8985	1.1091	1.4489
		98	.6196	.6403	.6624	.7597	.8275	.9087	1.1118	1.4534	2.0981
		94	.7043	.7312	.7602	.8911	.9859	1.1034	1.4178	2.0247	3.5401
	90	102	0.3675	0.3782	0.3897	0.4392	0.4725	0.5112	0.6042	0.7457	0.9738
		98	.4160	.4299	.4448	.5104	.5560	.6104	.7478	.9773	1.4102
		94	.4729	.4909	.5104	.5987	.6624	.7411	.9539	1.3619	2.3800
T_1	T_2	t_2	$t_1 = 50°$			$t_1 = 70°$			$t_1 = 90°$		
			$z=0.5$	$z=0.7$	$z=0.9$	$z=0.5$	$z=0.7$	$z=0.9$	$z=0.5$	$z=0.7$	$z=0.9$
120	70	112	0.6722	0.6868	0.7021	0.7643	0.8046	0.8493	0.9464	1.0769	1.2491
		108	.7557	.7742	.7936	.8742	.9272	.9870	1.1207	1.3085	1.5719
		104	.8516	.8752	.9001	1.0052	1.0759	1.1573	1.3455	1.6255	2.0528
	80	112	0.5418	0.5535	0.5658	0.6163	0.6487	0.6847	0.7635	0.8687	1.0074
		108	.6090	.6239	.6396	.7049	.7475	.7957	.9042	1.0555	1.2677
		104	.6863	.7053	.7253	.8105	.8674	.9329	1.0856	1.3113	1.6555
	90	112	0.4094	0.4183	0.4275	0.4660	0.4904	0.5175	0.5775	0.6569	0.7617
		108	.4602	.4714	.4832	.5329	.5651	.6014	.6839	.7983	.9585
		104	.5185	.5328	.5479	.6127	.6556	.7051	.8212	.9918	1.2518

VALUES OF e_1 OR e_2

Temp., deg. F.	50	70	80	84	88	90	92
B.t.u. . .	1081.7	1090.8	1095.3	1097.1	1098.9	1099.8	1100.7
Temp., deg. F.	94	98	102	104	108	112	
B.t.u.	1101.6	1103.4	1105.1	1106.0	1107.8	1109.5	

Table 8.—Pounds of Water Evaporated per Pound of Air

Outflowing air saturated

$T_1 = 100°$	$t_1 = 50°$			$t_1 = 70°$			$t_1 = 80°$		
	$z = 0.5$	$z = 0.7$	$z = 0.9$	$z = 0.5$	$z = 0.7$	$z = 0.9$	$z = 0.5$	$z = 0.7$	$z = 0.9$
t_2									
92	0.02834	0.02682	0.02531	0.02436	0.02126	0.01815	0.02124	0.01688	0.01253
88	.02449	.02297	.02146	.02051	.01741	.01430	.01739	.01303	.00868
84	.02105	.01953	.01802	.01707	.01397	.01086	.01395	.00959	.00524
$T_1 = 110°$	$t_1 = 50°$			$t_1 = 70°$			$t_1 = 90°$		
t_2									
102	0.04010	0.03858	0.03707	0.03612	0.03302	0.02991	0.02881	0.02278	0.01675
98	.03500	.03348	.03197	.03102	.02792	.02481	.02371	.01768	.01165
94	.03044	.02892	.02741	.02646	.02336	.02025	.01915	.01312	.00709
$T_1 = 120°$	$t_1 = 50°$			$t_1 = 70°$			$t_1 = 90°$		
t_2									
112	0.05556	0.05404	0.05253	0.05158	0.04848	0.04537	0.04427	0.03824	0.03221
108	.04888	.04736	.04585	.04490	.04180	.03869	.03759	.03156	.02553
104	.04288	.04136	.03985	.03890	.03580	.03269	.03159	.02556	.01953

refrigeration for gas refrigerating machines. For steam refrigeration, it recommends 3.20 sq. ft. per ton. The air requirements are 300 cu. ft. per min. per ton of refrigeration.

Mechanical refrigeration requires 30 gal.-deg. of water per min. per ton of refrigeration; *i.e.*, if but 1 gal. of water per min. per ton of refrigeration is circulated through the tower, the cooling range will be 30° F.; if 3 gal. per min. are circulated, the cooling range will be 10° F.; if 6 gal. per min. are circulated, the cooling range will be 5° F. Steam refrigeration (see p. 10–27) requires the circulation of at least 3 times the above quantities.

ECONOMY OF COOLING TOWERS.—Gustav J. Bischof (*Power*, Sept., 1933) presents a series of curves showing the relative cost of city water, at $1.00 per 1000 cu. ft. and the cost of operating a cooling tower for ammonia condensers for refrigerating plants, for condensers for steam turbines, and for Diesel engine jacket water. These curves are combined in Fig. 2. The ammonia condenser curve is based on 3 gal. per min. per ton of refrigeration and 10° F. cooling range. The steam turbine condenser curve is based on a steam rate of 20 lb. per kw.-hr., 28-in. vacuum, 940 B.t.u. dissipated per lb. of steam condensed, and a temperature difference of 8° F. between outlet water and vacuum temperature. The city water charges are based on 63° F. temperature. The Diesel engine curve is based on a 50° F. temperature rise in the city water, a 40° F. rise in the cooling tower water and a heat absorption by the cooling water of 3400 B.t.u. per brake-Hp.

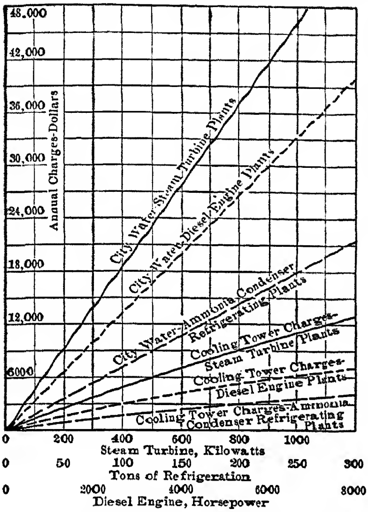


FIG. 2. Comparison of Cost of City Water with Cost of Cooling Tower Operation

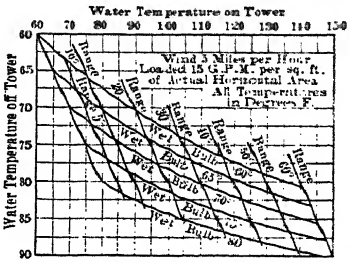


FIG. 3. Type of Guarantee Curve for Cooling Towers

COOLING TOWER GUARANTEES.—Cooling tower capacity sometimes is guaranteed on the basis of a given number of degrees of cooling with a given atmospheric temperature and relative humidity. Unless the stated temperature and humidity coincide, which may not occur over a period of years, the guarantee cannot be verified and the purchaser has no legal relief. A series of curves, similar to Fig. 3, furnished with the cooling tower, permits its performance to be checked under a wide range of conditions of temperature and humidity. The wind velocity in the case of atmospheric towers, the air velocity in mechanical draft towers, and also the loading per sq. ft. of active horizontal deck area, upon which the curves are based should be stated.

2. COOLING PONDS

EVAPORATION FROM PONDS.—In ponds, cooling is mainly by evaporation and is independent of the depth. Box (Treatise on Heat) gives for the rate of evaporation from a pond, in still air,

$$W = (240 + 3.7\,t)\,(P - p)/7000 \qquad [4]$$

where W = moisture evaporated per sq. ft. per hr., lb.; t = average temperature of water, deg. F.; P = saturation pressure of vapor at temperature t, in. of mercury; p = actual vapor pressure of air, in. of mercury. See also p. 3–48.

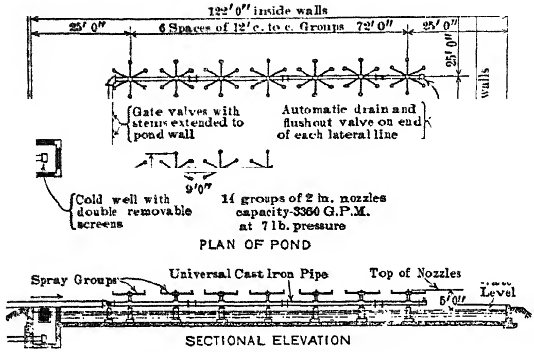
Area of Pond Required.—If the water is not sprayed into the pond, the theoretical area A sq. ft., required to cool Q lb. of water to a final temperature of t_2, deg. F., is

$$A = Q\,(t_1 - t_2)/H. \qquad [5]$$

where t_1 = initial temperature of water entering pond; t_2 = final temperature of water in

pond; H = heat dissipated per sq. ft. per hr., B.t.u., = Wh; h = latent heat of water at temperature t_1; W = weight of water evaporated as given by formula [4]. The value of A so found may be smaller than is usual in practice. Heat dissipation is modified by variations in wind velocity, air temperature and relative humidity. Actual values of H range from 4 B.t.u. per hr. per sq. ft. per deg. temperature difference in summer, to 2 B.t.u. in winter. A good practical average is 3.5 B.t.u.

SPRAY PONDS.—The heat-dissipating capacity of ponds may be increased greatly by spraying the water into them through nozzles which break the water into fine spray, increasing the evaporation and the cooling effect. Final temperatures depend on the cooling range, atmospheric conditions, arrangement of nozzles and storage capacity of pond. A lower final temperature is obtainable with moderate cooling ranges (10° to 20° F.) than with ranges of 30° to 40°. At the same relative humidity, final temperatures will be lower in warm weather than in cold, due to the greater moisture-absorbing power of warm air. Table 9 compiled from curves published by the Cooling Tower Co. gives average final temperatures that may be expected under various conditions.



G. 4. Typical Spray Cooling Pond

The quantity of water stored greatly exceeds the quantity sprayed per minute. Its average temperature is lower than that of the sprayed water, decreasing as the quantity stored increases. Fig. 4 is a layout of a typical spray pond.

Table 9.—Average Final Temperatures of Spray Cooling Ponds

Temp. of Water Entering Nozzles, deg. F.	Ammonia Condenser Service					Steam Condensing Service				
	Wet-bulb Temperature, deg. F.					Wet-bulb Temperature, deg. F.				
	60	65	70	75	80	60	65	70	75	80
	Temperature of Pond, deg. F.					Temperature of Pond, deg. F.				
60	60	......	......	......	......	60.0	......	......	......	......
70	65.	67.75	70.0	......	......	66.0	68.25	70.0	......	......
80	68.75	71.75	74.5	77.25	80.0	71.0	73.25	76.0	78.25	80
90	72.25	75.25	78.0	80.75	83.75	75.5	77.75	80.25	82.75	85.5
100	75.25	78.25	81.0	83.75	87.0	80.0	82.25	84.50	87.0	89.75
110	78.25	81.0	84.0	86.75	89.75	84.25	86.5	88.75	91.0	93.75
120	81.0	83.75	86.75	89.25	92.25	88.25	90.75	92.75	95.0	97.50
130	83.75	86.5	89.25	91.75	94.50	92.5	94.5	96.5	98.75	101.0

Nozzle pressures of 7 to 10 lb. per sq. in. are usual. Higher pressures produce a finer spray that may be blown away by the wind; lower pressures produce larger drops which retard cooling. Nozzles should be spaced 8 to 12 ft. The pond area should be 1 sq. ft. per 250 lb. of water sprayed per hour for plants of 1000 Hp. or over, and 1 sq. ft. per 150 lb. of water per hour for plants under 1000 Hp. Depth need not exceed 3 ft. Power required to circulate the water ranges from 1% to 2% of the power developed by the prime mover under high vacuum. Table 10 gives the capacities of one type of spray nozzle.

Table 10.—Capacity of Spray Cooling Nozzles

(Cooling Tower Co., New York, 1935)

Pipe Size, in.	Orifice, in.	Pressure at Nozzle, lb. per sq. in.								
		5	6	7	8	9	10	12	15	20
		Capacity, U. S. gallons per minute								
2	1 1/4	33.8	37.0	40.0	42.7	45.3	47.7	52.3	58.5	67.5
1 1/2	1	21.1	23.2	25.0	26.8	28.3	29.9	32.7	36.6	42.3
1 1/2	7/8	16.9	18.5	20.0	21.4	22.6	23.9	26.2	29.2	33.8
1	5/8	12.8	13.8	14.8	15.6	16.4	17.2	18.6	20.6	23.7

Section 10

REFRIGERATION AND ICE MAKING

By Louis A. Harding

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REFRIGERATION

REFRIGERATION AND ICE-MAKING

By Louis A. Harding

References.—An elaborate discussion of the thermodynamic theory of the various fluids used in the production of cold was published by M. Ledoux in the *Annales des Mines*, and translated in *Van Nostrand's Magazine* in 1879. This work, revised and additions made by Profs. Denton, Jacobus and Reisenberger, was reprinted in 1892 (Van Nostrand's Science Series No. 46). The work is largely mathematical. Other references are numerous papers in *Trans.* A.S.M.E. and *Trans.* A.S.R.E., and the following books: Siebel, Compend of Mechanical Refrigeration; Lorenz, Modern Refrigerating Machinery, translated by Pope; Gardner T. Voorhees, Refrigerating Machines; J. Wemyss Anderson, Refrigeration; Harding and Willard, Power Plants and Refrigeration; W. R. Woolrich, Refrigerating Engineering; H. J. McIntire, Refrigerating Engineer's Handbook; Refrigerating Data Book of A.S.R.E.; A. J. Wallis-Taylor, Refrigeration, Cold Storage and Ice-making. For Properties of Ammonia and Sulphur Dioxide, see papers by Profs. Wood and Jacobus, *Trans.* A.S.M.E., vols. ix and xii, 1888 and 1891; Univ. of Ill. Bulletin No. 66, Goodenough and Mosher; U. S. Bureau of Standards, Circular No. 142. The Data Book of A.S.R.E. gives thermal properties of a number of refrigerants.

For illustrated descriptions of refrigerating-machines, see catalogs of builders, as Frick & Co., Waynesboro, Pa.; De La Vergne Machine Co., New York; Vilter Mfg. Co., Milwaukee; York Mfg. Co., York, Pa.; Henry Vogt Machine Co., Louisville, Ky.; Carbondale Machine Co., Carbondale, Pa.; and others. See also articles in *Ice & Refrigeration*, *Power*, *Power Plant Engineering*, *Heating, Piping & Air Conditioning*.

REFRIGERATION

1. REFRIGERATION UNITS OF CAPACITY

MEASURE OF REFRIGERATING EFFECT AND STANDARD TON.—The quantity of heat abstracted or absorbed is measured in British thermal units. In the following discussion, 1 B.t.u. will be taken as equivalent to 778 ft.-lb. = 0.2930 watt-hr.

The commercial unit of refrigeration or *standard ton of refrigeration* as adopted by A.S.R.E. is the heat required to *melt* 1 ton (2000 lb.) of pure solid ice, *i.e.*, the heat absorbed by 2000 lb. of pure ice melting to water at 32° F. The 144 B.t.u. absorbed is the latent heat of ice. One ton of refrigeration, therefore, equals $(144 \times 2000) = 288{,}000$ B.t.u. per 24 hr. = 12,000 B.t.u. per hr. = 200 B.t.u. per min. All refrigeration calculations are made on a 24-hr. or day basis.

RATING OF REFRIGERATING MACHINES.—Refrigerating machines are rated in tons of refrigeration per 24 hours with a statement of the temperature and pressure range through which they operate while performing this duty.

A refrigerating machine using liquefiable gas or vapor is rated by the number of *standard tons* it is capable of producing under pressures which correspond to a saturated temperature of 5° F. for inlet or suction pressure, and 86° F. for condenser pressure, with 9° F. subcooling of the liquid entering the throttling or regulating valve, and 9° F. superheating of vapor entering the compressor. These pressures and temperatures are to be measured outside of and within 10 ft. of the machine.

The Ice-making Capacity of a machine in tons of ice is assumed as approximately one-half its refrigerating capacity. The production of 1 ton of ice necessitates reducing the temperature of the water to 32° F., the extraction of its latent heat, and reducing the temperature of the ice to the mean temperature of the brine tank (approximately (11° to 16° F.). The average temperature of the manufactured ice harvested is $(16 + 32)/2 = 24°$ F. Heat extracted to freeze 1 lb. of ice (initial temperature of water, 75°) is:

To lower temperature of water, $1 \times (75 - 32) \times 1$ (specific heat of water) =	43 B.t.u.
To freeze 1 lb. of water, 1×144 (latent heat of ice) .... =	144 "
To reduce temperature of ice, $1 \times (32 - 16) \times 0.5$ (specific heat of ice) ... =	8 "
Total..	195 B.t.u.
20% of 195 (heat transmission losses of brine tank, ice storage room, heat introduced by warm ice cans, etc.)	39 "
Total heat required to manufacture 1 lb. of ice.......................	234 B.t.u.

The heat required to manufacture 1 ton of ice from water at 75° F. is $2000 \times 234 = 468{,}000$

B.t.u. The refrigerating capacity of the machine required for 1 ton of ice, working through the above temperature range, is 468,000/288,000 = 1.625 tons of refrigeration.

REFRIGERATING LOAD.—The refrigerating load of a cold storage or other refrigerating plant consists of heat abstracted for the following purposes: 1. To cool the goods

Table 1.—Composition and Specific Heat of Food Products and Storage Temperatures

Product	Temp. Carried, deg. F.	Composition		Specific Heat above 32°	Specific Heat below 32°	Latent Heat of Freezing, B.t.u.
		Water, percent	Solids, percent			
Lean beef	30	72.00	28.00	0.77	0.41	104
Fat pork	30	39.00	61.00	.51	.30	56
Eggs	30	70.00	30.00	.76	.40	101
Potatoes	34	74.00	26.00	.80	.42	107
Cabbage	33	91.00	9.00	.93	.48	131
Carrots	33	83.00	17.00	.87	.45	120
Cream	33	59.25	30.75	.68	.38	85
Milk	35	87.50	12.50	.90	.47	126
Oysters	35	80.38	19.62	.84	.44	116
Whitefish	15	78.00	22.00	.82	.43	117
Chickens	28	73.70	26.30	.80	.42	106
Ice	28	100.00	0.00	1.00	.504	144

Table 2.—Space Required for Refrigerated Goods

(Meat rails placed approximately 30 inches on centers)

Material	Average Weight, pounds	Floor Space, sq. ft.	Space Occupied, cu. ft.	Clear Height of Room
1 barrel apples or potatoes	160	2.5	5.9	......
1 tub butter	60	2.5	2.5	......
1 cheese	60	2.0	2.0	......
1 case eggs (30 doz.)	70	.....	3.0	......
1 beef	700	9.0	108.0	12′ 0″
1 sheep	75	2.0	16.0	8′ 0″
1 hog	250	.....	.....	......
1 calf	90	.....	.....	......

Table 3.—Approximate Cold-storage Temperatures

Article	Temp., deg F.	Article	Temp., deg. F.	Article	Temp., deg. F.
Apples	33	Fish, fresh water, frozen	18	Oleomargarine	20
Asparagus	33	Fish, salt water, not frozen	15	Onions	32
Bananas	55	Fish, to freeze	5	Oranges, short carry	50
Beans, fresh	32	Fish, dried	40	Oranges, long carry	34
Beans, dried	45	Fruits, canned	40	Oxtails	30
Beef, fresh, short carry	35	Fruits, dried	40	Oysters in shell	43
Beef, fresh, long carry	30	Furs	28	Oysters in tubs	35
Beef, dried	40	Game, short carry	28	Parsnips	32
Berries, fresh short carry	40	Game, after frozen	10	Peaches, short carry	50
Buckwheat flour	42	Game, to freeze	0	Pears	33
Butter	14	Grapes	36	Peas, dried	45
Butterine	20	Hams, not brined	20	Plums	32
Cabbage	33	Hogs	30	Potatoes	34
Cantaloups, short carry	40	Honey	45	Poultry, dressed, iced	30
Cantaloups, long carry	33	Hops	32	Poultry, short carry	28
Carrots	33	Huckleberries, frozen	20	Poultry, after frozen	10
Celery	32	Ice	28	Poultry to freeze	0
Cheese, long carry	35	Ice cream, short carry	15	Raisins	55
Chestnuts	34	Lemons, short carry	50	Salt meat curing room	32
Cigars	42	Lemons, long carry	38	Sardines, canned	40
Corn, dried	45	Lambs	32	Scallops, after frozen	16
Cornmeal	42	Lard	40	Sheep	32
Cranberries	33	Maple syrup and sugar	45	Sugar	45
Cream, short carry	33	Meats, canned	40	Syrup	45
Cucumbers	38	Meats, salt, after curing	43	Tenderloins	33
Currants, short carry	32	Milk, short carry	35	Tobacco	42
Dates	55	Nuts in shell	40	Tomatoes, ripe	42
Eggs	30	Oatmeal	42	Watermelons, short carry	40
Figs	55	Oils	45	Wheat flour	42
Fish, not frozen, short carry	28			Wines	50
				Woolens	28

stored, depending on initial and final temperatures of the goods, their weights and specific heats. 2. To offset heat transmission of the cold storage room walls, depending on the difference in temperature between the inside and outside of the room and the character of wall construction. 3. For ventilation including heat to be extracted from air passing into the room for ventilating purposes, and that required for lowering the temperature of and precipitating moisture contained in the entering air, depending upon initial and final temperatures of the air and its relative humidity. 4. To offset heat inside the room, including that generated by men working, by motors, fans, etc., and lights in the room.

1. Heat abstracted to cool the goods stored = $sw(t_1 - t)$, where s = specific heat of goods; w = weight, lb.; t_1 = outside temperature, deg. F.; t = inside temperature, deg. F. If the goods are to be frozen, *i.e.*, carried at a temperature below 32° F., the B.t.u. required = $(t_1 - t_2)\ sw + (Pw \times 144)$, where P = percentage of water contained in the goods. Tables 1, 2, 3, 4 and 5 give data for estimating the above quantities.

2. Heat transmission of walls may be estimated from Tables 6 and 7.

3. If a fan is used to circulate cold air in the room or through the cold room, the heat generated by the fan will be introduced into the circulation and extra refrigeration must be provided on the basis of 1 Hp. = 2546 B.t.u. per hr. The brake-Hp. required for a fan will be B.Hp. = $5.2\,pQ/(E \times 33{,}000)$, where p = total pressure of the air, in. of water, at the fan outlet; Q = cu. ft. of air circulated per min.; E = mechanical efficiency of fan = 0.50 for a steel plate ventilating fan, and 0.60 for a multiblade fan (approximate). In a well-proportioned system of ducts and cooling chamber, p ordinarily should not exceed 1 in. of water, equivalent to a pressure of 5.2 lb. per sq. ft.

4. The heat abstracted to offset the heat generated inside the cold storage room may be estimated from the following data: Heat given off per workman per hr., 500 B.t.u.; heat given off by one gas light per hr., 3600 B.t.u.; heat given off by incandescent lights, 3.41 B.t.u. per watt.

Example.—Required the refrigeration to cool 50,000 lb. of meat from 95° to 35° F. per 24 hr.; outside temperature 85° F. and 70% humidity. Size of room 40 × 60 × 12 ft. Ventilation, 10 changes of air per 24 hr., based on inside temperature. Walls, 13 in. brick; roof, 1 1/2 in. wood with large air space and 1 in. ceiling. Floor construction, 6 in. concrete on cinder fill. Insulation on all walls, floor and ceiling, 3-in. corkboard.

1. *Refrigeration Load.*—To cool the goods stored,

$$sw(t_1 - t) = 0.8 \times 50{,}000 \times (95 - 35) = 2{,}400{,}000 \text{ B.t.u. per 24 hr.}$$

2. *Heat Transmission.*—B.t.u. = area × transmission coefficient × temperature difference.

12-in brick walls + 3-in. corkboard, 2400 sq. ft. × 1.75 × 50	210,000 B.t.u.
6-in. concrete floor + 3-in. corkboard, 2400 sq. ft. × 1.97 × 15	71,000 "
1-in. wood ceiling + 3-in. corkboard, 2400 sq. ft. × 1.99 × 50	238,800 "
	519,800 B.t.u.

3. *Ventilation.*—Cu. ft. of air introduced per 24 hr. = 40 × 60 × 12 × 10 = 288,000 cu. ft. Weight of moisture contained in air at 85° F. and 70% humidity (0.026 × 0.70) = 0.0182 lb. of vapor per lb. of air. The air will be reduced to a saturated state of 35° F. on entering the room and will then carry 0.00427 lb. of moisture per lb. of air introduced. Latent heat of the vapor at 35° F. = 1074.4 B.t.u. Specific heat of vapor = 0.46. Weight of air per 24 hr. = 23,040 lb.

To lower temperature of air, 23,040 × 0.24 (85 − 35)	276,480 B.t.u.
To lower temperature of vapor, 23,040 × 0.018 × 0.46 (85 − 35)	9,539 "
To condense vapor, 23,040 × (0.0182 − 0.00427) × 1074.4	346,558 "
	632,577 B.t.u.

The precipitated moisture deposited on the cooling coil will be frozen. The coil temperature is approximately 10° lower than room temperature; in this case it is 25°. The heat abstracted per lb. of moisture precipitated is as follows:

Reducing temperature from 35° to 32°	3 B.t.u.
Freezing	144 "
Reducing temperature of ice from 32 to 25 = 0.5 (32 − 25)	3.5 "
	150.5 B.t.u.

Total heat thus abstracted, 150.5 × 0.014 × 23,040 = 48,545 B.t.u.
Total heat abstracted by ventilation = 632,577 + 48,545 = 681,122 B.t.u.

Total Refrigeration Required.

(1) To cool the goods stored	2,400,000 B.t.u.
(2) Heat transmission	519,800 "
(3) Ventilation	681,122 "
	3,600,922 B.t.u.

1 ton of refrigeration = 288,000 B.t.u.; refrigeration required is 3,600,922/288,000 = 12.5 tons

HEAT TRANSMISSION OF BUILDING CONSTRUCTION.—The method, including conductivities of the materials and surface coefficients, used in calculating heat transmission of building construction is given in Section 11, p. 11–03. Data in Table 6 were calculated as therein described. The heat transmission of a compound wall for any combina-

Table 4.—Liquids and Solids in Foods and Humidity at which They Should Be Stored

Product	Temp., deg. F.	Composition, percent Water	Solids	Rel. Humidity, %	Product	Temp., deg. F.	Composition, percent Water	Solids	Rel. Humidity, %
Apples (Storage)	32–36	63.5	36.5	....	Lemons (Storage)	36–38	......	......	85
Bananas:					Beef (Storage)...	34–38	68	32	76
(Retarding)...	60	75.5	24.5	80	Poultry (Storage)	28–30	73.7	26.3	....
(Ripening)....	64	75.5	24.5	84	Potatoes (Storage)	36–40	74	26	....
(Forcing).....	68	75.5	24.5	88	Eggs (Storage)...	31	70	30	....

Table 5.—Operating Conditions for Meat Storage Rooms

	Precooling Rooms	Cold Store Room	Frozen Meat Room
Temperature, deg. F.	41–46.4	35.6–39.2	21.2–15.8
Relative humidity, percent	70–80	70–80	70–85
Meat stored, lb. per sq. ft.	41	31	31
Initial meat temp., deg. F.	82.4	59	32*
Final meat temp., deg. F.	59	37.4	21.2–17.6
Hours of cooling; freezing and cooling	20	30	72
Water evaporated, percent of weight of meat:			
First day	0.65		
Second and third day		0.35	
In three days			0.345

* Meat already cooled to 32° F.

Table 6.—Heat Transmission Coefficients for Insulated Walls
(B.t.u. transmitted per sq. ft. per 24 hr. per deg. difference in air temperature)

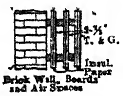
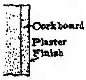
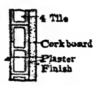
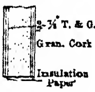
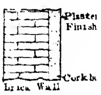
Construction	Wall Thickness, in.	Thickness of Cork Board Insulation, in. — Heat Transmission Coefficient			
	8	3.77	2.40	2.06	1.46
	12	3.36	2.29	1.75	1.39
	16	2.52	1.85	1.47	1.22
	6	4.34	2.71	1.97	1.56
	8	4.18	2.64	1.92	1.51
	10	3.98	2.56	1.90	1.49
	6	3.62	2.40	1.80	1.44
	8	3.53	2.37	1.78	1.43
	10	3.46	2.34	1.76	1.42
		Thickness of Granulated Cork, in.			
			8	10	12
		Heat Transmission Coefficient			
	1.47	1.09	0.86	0.72	0.61
	Brick Wall, in.	Number of Air Spaces — Heat Transmission Coefficient			
	8	4.30	2.78	2.06	1.62
	12	3.74	2.54	1.92	1.56
	16	3.36	2.37	1.30	1.47

Table 7.—Recommended Practice for Designing Cold-storage Walls
(Outside temperature assumed, 85° to 95° F.)

Inside Temperature, deg. F.	B.t.u. Transmitted per sq. ft. per deg. Temp. Diff. per 24 hr.	Tons Refrigeration per 24 hr. per 1000 sq. ft.
– 10° to + 5°	1.00	0.32 to 0.28
5° to 20°	1.25	.35 to .28
20° to 32°	1.50	.34 to .27
32° to 45°	2.00	.37 to .27
45° and above	3.00	.41

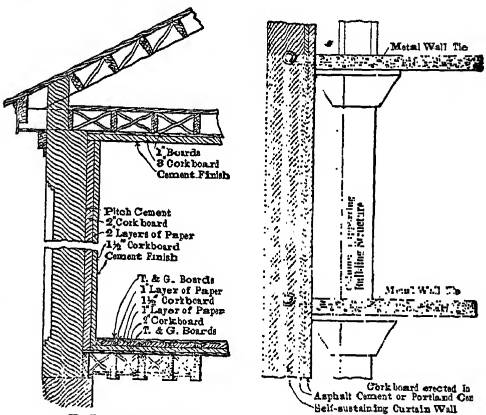
tion of materials may be found when the conductivities and surface coefficients of the materials are known.

The following conductivities C were used in the calculations for data given in Table 6; Brick, $C = 5$; concrete, $C = 8$; tile, 6-in., $C = 0.54$; 8-in., $C = 0.49$; 10-in., $C = 0.46$; cork board, $C = 0.30$; plaster, $C = 2.32$; $^7/_8$-in. boards, $C = 1.0$; one air space, $C = 1.1$. Outside wall surface coefficient, $K = 1.34$; inside wall surface coefficient, $K = 4.02$. The value of u as determined by the heat transmission formula was multiplied by 24 to obtain heat transmission per 24 hr.

2. COLD STORAGE

Relative Humidity for Cold Storage Rooms.—If air in a cold storage room is recirculated only, it soon is devoid of moisture, which is condensed on cold coil surfaces. The ultimate result is evaporation of moisture from stored product, with a loss of weight and value, which is to be avoided if possible. The proper relative humidity for each product stored has not been definitely ascertained. In general, it is safe to maintain a percent relative humidity equal to the moisture percentage in the stored product. Table 4 (H. Vetter) gives recommended data for several products.

C. E. Baker gives the optimum temperature for apple storage as 31° to 33° F. with relative humidity of 80 to 85%. Approximate control of relative humidity in rooms cooled by the direct method may be obtained by relatively large coil surfaces proportioned on a basis of 10° to 12° temperature difference.



Built-up Insulation Continuous Insulation

FIG. 1. Methods of Insulating Walls

While the method of approximately controlling relative humidity usually is satisfactory for small or medium size rooms, large rooms are best served by a central refrigerating and air conditioning system, rooms being refrigerated by cold air fan circulation. The circulated air is cooled and saturated by a spray type dehumidifier. Water for sprays is cooled by refrigeration. See Section 11, p. 11–55. Table 5, giving operating conditions for meat storage rooms cooled by a central cooling and air conditioning plant is from Sulzer *Tech. Rev.*, No. 3, 1927, translated by J. D. Blake in *Aerologist*, June, 1928.

Provision must be made to heat saturated cooled air leaving dehumidifier or cooling chamber by passing it through a heater to obtain complete and satisfactory control of relative humidity in rooms being cooled.

CORK PIPE-COVERING is a molded covering, manufactured in three types, *viz.*, ice water thickness, brine thickness, and heavy brine thickness. Ice water thickness, used for temperatures above 25° F., is $1^1/_2$ to 2 in. thick. Brine thickness, used for temperatures from 0° F. to 25° F., is 2 to 3 in. thick. Heavy brine thickness, used for temperatures below 0° F., is 3 to 4 in. thick.

THE BRINE SPRAY BUNKER system of room cooling is used principally in hog or beef chill rooms of meat packing plants. It consists of spraying cold brine in spray bunkers or lofts near the ceiling. The brine spray induces rapid circulation of air and maintains a fairly high relative humidity, which rapidly chills meat without "case hardening" and without appreciable loss of weight or shrinkage. Pressure of 8 to 20 lb. per sq. in. is maintained at the spray nozzles, spaced 1 to 5 ft. centers. At each end of the bunker is a large gravity air circulation duct. Air circulation is in direction of the spray. This system is cheaper than a cold air fan circulating system.

THE FORCED AIR CIRCULATION SYSTEM uses a fan to recirculate air over cooling coils located in a small room called a cooler. The air is piped from the cooler to

the cold storage rooms, and returned from them to the fan by a return duct. A coil with fan (unit cooler) is becoming a popular method of room cooling. The advantages of the system are: Centralizing of brine or refrigerant piping; moisture precipitation all occurs in the cooler; cold storage rooms are dry and free from cooling coil drip.

If air is cooled by a spray type dehumidifier, relative humidity in the rooms also may be controlled. It must be possible to heat the air leaving the dehumidifier to obtain complete control of relative humidity in the cold storage rooms.

Example.—Assume a forced air circulation system to be used in the previous example, the air leaving the cooler at 20° F., and returning at 35°. The fan must handle

$$3{,}600{,}922/(0.24 \times 15 \times 60 \times 24) = 694 \text{ lb. of air} = 8710 \text{ cu. ft. of air per min.}$$

With an air velocity of 500 ft. per min., through the face area of cooling coils, the total pressure rating of the fan should not exceed 1 in. of water. This will require a steel plate fan, 42 in. wheel, 475 r.p.m. and 4.58 brake-Hp., or a multiblade fan, 36 in. diam., 300 r.p.m. and 3.3 brake-Hp. Using the multiblade fan, the heat equivalent of the power required to move the air, which is intro- nto the circulation is 3.3 × 2545 × 24 = 201,564 B.t.u. per 24 hr., equivalent to [illegible] = 0.70 ton of refrigeration, which must be added to the amount previously calcu- The total refrigeration required then is 12.5 + 0.70 = 13.2 tons.

TESTS OF SMALL REFRIGERATORS.—Table 8 gives results of tests made by the author to determine the heat transmission of the walls of three well-known makes of small ice-cooled refrigerators. The figures are an average of several tests run in still air with the boxes empty and doors made tight with felt and kept closed during the tests. The boxes were cooled to constant temperature conditions before the tests were started.

REFRIGERATION REQUIRED FOR SMALL BOXES AND ROOMS.—In large rooms the heat losses may be analyzed with some degree of certainty when conditions of operation are known. For small refrigerators as in hotels, kitchens and private homes, etc., the following data are recommended as giving better results than a more elaborate analysis. Heat loss, B.t.u. per cu. ft. per 24 hr.: Pantry refrigerator, 300; kitchen refrigerator, 600 to 900; butchers' display refrigerators, 200 to 250; long storage, 150 to 200. An allowance of from 200 to 225 B.t.u. is made per lb. of ice. In applying the data, as- a refrigerator temperature of approximately 45° F. and an average summer tem-

Table 8.—Results of Tests of Small Refrigerators

	Box A	Box B	Box C
External area, square feet	32.3	29.6	28.4
Length of test, hours	12.15	12.35	12.20
Average inside temperature, deg. F	54.0	55.0	55.7
Average outside temperature, deg. F	80.2	80.7	80.5
Pounds of ice melted	10.8	10.5	11.5
B.t.u. transmitted per sq. ft. per deg. difference in outside and inside temperature in 24 hours	3.67	3.86	4.61

Table 9.—Approximate Refrigeration Required for Large Boxes

(Based on 24-hr. continuous operation)

Cu. Ft. Space in Box or Room	Temperature = 20 deg. F.							Temperature = 10 deg. F.						
	Cu. Ft. per Ton of Refrigeration	Cu. Ft. per 1 ft. of Pipe						Cu. Ft. per Ton of Refrigeration	Cu. Ft. per 1 ft. of Pipe					
		Pipe Size, in., Direct Expansion			Pipe Size, in., Brine				Pipe Size, in., Direct Expansion			Pipe Size, in., Brine		
		1	1 1/4	2	1	1 1/4	2		1	1 1/4	2	1	1 1/4	2
12	113	1.6	2.2	.....	1.4	2.0	.....	93	1.0	1.2	.....	0.6	1.1	....
20	137	1.7	2.3	.....	1.4	2.0	.....	112	1.0	1.2	.....	0.6	1.1	....
50	160	1.8	2.4	.....	1.5	2.1	.....	130	1.1	1.2	.....	0.6	1.1	....
100	205	2.0	2.6	.....	1.6	2.2	.....	168	1.2	1.3	.....	0.6	1.2	....
250	348	....	2.8	.....	....	2.4	.....	280	....	1.4	.....	....	1.3	....
1,000	580	....	3.2	.....	....	2.7	.....	470	....	1.6	2.5	....	1.5	....
3,000	820	....	3.8	5.5	....	3.0	4.0	650	....	2.2	3.0	....	1.7	2.3
5,000	1100	....	4.5	6.5	....	3.4	4.5	840	....	2.5	3.6	....	1.9	2.6
10,000	1600	....	6.0	8.0	....	4.0	5.5	1140	....	3.2	4.6	....	2.2	3.3
20,000	2100	....	7.0	10.0	....	4.7	6.5	1600	....	4.0	5.7	....	2.6	4.0
40,000	2600	....	8.0	12.0	....	5.5	7.5	2100	....	4.8	6.8	....	3.0	4.7
70,000	3200	....	9.0	14.0	....	6.5	8.5	2600	....	5.5	8.0	....	3.5	5.5
100,000	4000	....	11.0	17.0	....	7.5	10.0	3100	....	6.5	10.0	....	4.2	6.7
150,000	4900	....	14.0	20.0	....	9.0	12.0	3800	....	8.0	12.0	....	5.0	8.0
	Mean Temperature, Ammonia Expansion........ 0° F. Mean Temperature, Brine in Coils. 10° F.							Mean Temperature, Ammonia Expansion........ 0° F. Mean Temperature, Brine in Coils. 5° F.						

perature of 72° F. For pantry and kitchen refrigerators, outside dimensions are used in figuring volumes. Each lineal foot of insulated brine pipe requires approximately the same refrigeration as 1 cu. ft. of box. The approximate refrigeration required for large boxes and rooms, together with the pipe or coil surface necessary, is given in Table 9.

EXAMPLE.—Required the amount of refrigeration and brine pipe surface for a pantry refrigerator, $4 \times 2 \times 6$ ft.; average inside temperature 45° F.; average brine temperature in cooling pipes, 20° F. Allow for 50 ft. of insulated brine pipe. Total cu. ft. $= (4 \times 2 \times 6) + 50 = 98$. Tons of refrigeration per 24 hr. $= (300 \times 98)/288{,}000 = 0.102$, equivalent to $0.102 \times 12{,}000 = 1224$ B.t.u. per hr. Taking the coefficient of heat transmission $K = 2$, the amount of pipe surface required will be $S = (300 \times 48)/24 \times 2\ (45 - 20) = 12$ sq. ft. or 34 lineal feet of 1-in. pipe. See Heat Transfer in Room Piping, p. 10–10.

APPROXIMATE REFRIGERATION IN TONS REQUIRED FOR VARIOUS PURPOSES.—The following data are from a bulletin issued by Am. Carbonic Machy. Co. These data will guide preliminary estimates and check more refined calculations after operating conditions are known.

General Cold Storage Buildings

Desired temp., deg. F.	0	5	10	20	32	36
Cu. ft. per ton, rooms up to 1000 cu. ft.	500	1000	2000	3000	4000	5000
" " " over " "	1000	2000	3000	5000	7000	8000

One ton of refrigerating machine capacity is required for the quantities given in each class in the following services:

Apartment Buildings.—16–20 apartment refrigerators; 1000 cu. ft. of restaurant refrigerator.

Bakeries.—800 cu. ft. of air circulated for air-conditioned dough mixing room*; 1000 cu. ft. of air through high speed 3-bbl. dough mixer*; 1000 gal. water cooled per day.*

Candy Factories.—2000 cu. ft.*; 1500 cu. ft. of fruit and nut storage*; 2 chocolate enrobers.

Creameries.—3000 gal. of milk cooled per day; 1000 cu. ft. of butter and milk storage.

Drinking Water.—1000 gal. per day cooled from 75° to 40° F.*; 200 persons in office building*; 150 persons in factories.*

Fur Storage Vaults.—1500 cu. ft. for vault space.*

Hotels.—1000 cu. ft. of kitchen service and storage refrigerators; 5–10 ice-cream cans in serving counters; drinking water cooling, 60 rooms*; ice-making, 125 rooms; air conditioning and cooling dining-rooms, 10–12 seats*; ball rooms, 500 cu. ft. of air circulated.*

Ice Cream Factories.—120 to 150 gal. of ice cream made and hardened per day.

Ice Plants.—1000 lb. of ice per day.

Meat Markets.—1000 cu. ft. storage refrigerator space; 50 lineal ft. display counter.

Pipe Line Losses.—1000 lineal ft. drinking water piping; 750 lineal ft. brine mains.

Theater Cooling.—10–15 seats* (see Sect. 11, p. 11–60).

Ice Skating Rinks.—The following averages are based on 9 rinks in the U. S.: Sq. ft. of ice surface per ton of machine capacity, 169.3; lineal feet of 1 1/4-in. pipe per sq. ft. of ice surface, 2.85, 4-in. pipe centers, pipes across the arena. For good skating conditions, about 0.40 B.t.u. per min. per sq. ft. must be absorbed. Brine temperature, approximately 18° F. for bare pipe; 10° to 12° F. for embedded pipe. Ice surfaces removed one day and replaced the next require nearly twice as much refrigeration as a surface that remains frozen. Brine storage tanks always are used, but tank capacity is relatively small.

3. HEAT TRANSMISSION

HEAT TRANSMISSION OF PIPING IN REFRIGERATING PRACTICE.—The coefficient of heat transmission K = B.t.u. transmitted per sq. ft. per deg. F. difference in temperature per hr. It varies with the velocity of the gas or liquid in contact with the surfaces. The following formulas are given (*Trans.* A.S.R.E., 1907) for the heat transmission of double-pipe ammonia condensers: $K = 130\sqrt{w_w}$; $Q = KS \times \Delta t$, where Q = heat transmitted, B.t.u. per hr. $= K \times S \times \Delta t$; S = sq. ft. of surface; t_1, t_2 = initial and final temperatures of cooling water or brine, respectively, deg. F.; Δt = mean temperature difference; w_w = velocity of water over the pipe surface, ft. per sec.

For double-pipe brine coolers $K = 84\sqrt{w_b}$, where w_b = velocity of brine through the pipe, ft. per sec., $Q = K \times S \times \Delta t$, and $\Delta t = \{(t_1 + t_2)/2\} - t_0$, where t_0 = temperature

* Required capacity of compressor for daily operating periods of 24 hr. or less; allowances not marked (*) are based on 24-hr. operation of compressor. For less than 24-hr. operation (brine circulation), size of plant is $(24 \times Q)/h$, where Q = size as determined from above allowances and h = hours operated.

of gas maintained in evaporating coil. The York Mfg. Co., York, Pa., gives the heat transmission obtained with circulating water and brine at different velocities as follows:

For 1 1/4-in. and 2-in. double-pipe ammonia condensers:

Velocity of water in coil, ft. per min	100	150	200	250	300	400
B.t.u. per hr. per sq. ft. per deg. mean temperature difference	150	198	240	260	300	338

For 2-in. and 3-in. double-pipe brine coolers:

Velocity of brine in coil, ft. per min	100	150	200	250	300	400	500	600	700	800
B.t.u. per hr. per sq. ft. per deg. mean temperature difference	95	112	130	145	158	177	191	205	215	220

The above figures were obtained in tests. For ordinary practice the heat transmission is taken as 30 to 40% less.

The York Mfg. Co. gives for the surface required in ammonia condensers

$$S = [HA + \{AC(t - t_1)\}] \div \{K(t_1 - t_2)\} \quad . \quad . \quad . \quad . \quad . \quad . \quad [1]$$

where S = surface, sq. ft.; H = heat of vaporization at condensing pressure, B.t.u. A = ammonia circulated per min. per ton of refrigeration, lb.; C = specific heat of vapor; t = temperature of gas entering condenser, deg. F.; t_1 = temperature due to condensing pressure; t_2 = mean temperature of water on and off condenser; K = B.t.u. transmitted per hr. per sq. ft. per deg. mean temperature difference.

Various tests made on shell-tube type condensers indicate the following values for K for various water velocities through tubes:

Velocity, ft. per min	100	150	200	250
K	170	190	210	230

HEAT TRANSMITTED FROM LIQUID TO LIQUID.—Hausbrand gives for the quantity of heat transmitted from liquid to liquid, the formula

$$K = 60/[\{1/(1 + 3.33\sqrt{w_1})\} + \{1/(1 + 3.33\sqrt{w_2})\}] \quad . \quad . \quad . \quad . \quad . \quad [2]$$

where w_1 and w_2 = velocities on opposite sides of the pipe surface. If $w_1 = w_2$, then $K = 30(1 + 3.33\sqrt{w})$, and $Q = KS \times \Delta t$, where Δt = mean temperature difference between the two liquids. This formula is applicable to the "heat exchanger" used in the absorption system.

MEAN TEMPERATURE DIFFERENCE.—The mean temperature difference Δt between two fluids which alter their temperatures during an exchange of heat, may be determined by the constant in Table 10. If D_e and D_a = smallest and greatest temperature difference, respectively, and D_e/D_a = ratio of smallest to greatest difference (see column 1) and M = a constant (see column 2), then $\Delta t = M \times D_a$ is the mean temperature difference.

HEAT TRANSMITTED FROM STEAM TO BOILING WATER.—Prof. Greene gives from the experiments of Jelinck, the formula

$$K = 953/\sqrt{ld} \quad . \quad . \quad . \quad . \quad . \quad . \quad . \quad . \quad . \quad . \quad . \quad [3]$$

d = diam of pipe or tube, ft ; l = total length of pipe, ft.

HEAT TRANSFER IN ROOM PIPING.—The rate of heat transfer per deg. difference in temperature between ammonia or brine and air, circulated by gravity in cold-storage rooms as interpolated from a diagram by C. H. Herter (*Refrig. Wld.*, Oct., 1915) are:

D =	6	8	10	12	14	16	18	20
K =	1.20	1 50	1.75	2.00	2.17	2.30	2.40	2.48

D = temperature difference between room and refrigerating medium; K = heat transfer,

Table 10.—Mean Temperature Difference Δt. (Hausbrand)

1	2	1	2	1	2	1	2
$\frac{D_e}{D_a}$	Δt. $D_a = 1$	$\frac{D_e}{D_a}$	Δt. $D_a = 1$	$\frac{D_e}{D_a}$	Δt. $D_a = 1$	$\frac{D_e}{D_a}$	Δt. $D_a = 1$
0.0025	0.166	0.10	0.391	0.21	0.509	0.55	0.756
.005	.189	.11	.405	.22	.518	.60	.786
.01	.215	.12	.418	.23	.526	.65	.815
.02	.251	.13	.430	.24	.535	.70	.843
.03	.277	.14	.440	.25	.544	.75	.872
.04	.298	.15	.451	.30	.583	.80	.897
.05	.317	.16	.461	.35	.624	.85	.921
.06	.335	.17	.466	.40	.658	.90	.953
.07	.352	.18	.478	.45	.693	.95	.982
.08	.368	.19	.489	.50	.724	1.00	1.000
.09	.378	.20	.500				

B.t.u. per hr. per sq. ft. per deg. temp. diff. These values are to be applied to room piping fairly free from frost. Table 11, condensed from one by O. Gueth in the Refrigerating Engrs. Manual, presumably represents practice of various makers of refrigerating machines. See also series of papers, Basic Laws and Data of Heat Transmission, by W. J. King, *Mech. Engg.*, Mar.-Aug., 1932, for a comprehensive survey of this subject.

The method used in the following example, to ascertain coil surface required for a given amount of work is due to C. D. Fehl.

EXAMPLE.—5000 lb. of beef are stored per day of 24 hr. in a room 31 × 12 × 15 ft. high, exposed on all sides to outside temperature of 90° F. Beef enters the room at 90° F. and is cooled to room temperature of 33° F. Using 4-in. corkboard on all parts of room, allows a heat leakage of 1.5 B.t.u. per sq. ft. per 24 hr. per deg. temperature difference. Specific heat of beef assumed as 0.77. Total exposed surface = 1980 sq. ft.

Heat transmitted through wall, $\{1980 \times 1.5 \times (90 - 33)\}/24$	=	7,054 B.t.u.
Heat abstracted from beef, $\{5000 \times 0.77 \times (90 - 33)\}/24$	=	9,144 "
Two workmen at 500 B.t.u., each	=	1,000 "
Three 16-cp. incandescent lights, at 254 B.t.u., each	=	762 "
Total B.t.u. per hr.	=	17,960 B.t.u.
Add 20% for opening doors, etc.	=	3,592 B.t.u.
Net heat to be abstracted per hr	=	21,552 B.t.u.

The surface required in the room is based on the assumption that the brine enters the room coils at 15° and leaves at 20° F. The least difference in temperature is $(33 - 20) = 13$, and the greatest is $(90 - 15) = 75$. Using Hausbrand's method (see above) $13/78 = 0.173$, and from Table 10, by interpolation, the coefficient corresponding to 0.173 is $M = 0.47$; whence $0.47 \times 75 = 35.25$ mean temperature difference. If the arithmetical mean temperature difference is used, it is

$$\{(90 + 33)/2\} - \{(20 + 15)/2\} = 44° \text{ F.}$$

The surface required is $21{,}552/(2 \times 35.25) = 305.7$ sq. ft., assuming that the pipe is frosted and transmits 2 B.t.u. per hr. per sq. ft. per deg. of temperature difference. Lineal feet of 1 1/4-in. pipe required is $305.7 \times 2.301 = 703.4$.

The above method may be used to calculate all coil surfaces, where velocities of liquids or gases as well as the heat transmission at the different velocities are known. It is advisable to add 20% to the calculated surface, since in practice the brine may rise to a higher temperature than assumed.

4. METHODS OF PRODUCING ARTIFICIAL REFRIGERATION

Low temperatures may be produced by a comparatively rapid absorption of heat by certain substances during either a chemical or a physical change of state. Chemical changes require, and are accompanied by, a heat transfer. The tendency of certain salts in combination with water, acid or ice, to pass into a liquid state is so great that the heat energy required for the change cannot all be supplied from outside the mixture. The deficiency is supplied by the heat of the mixture itself, with a consequent lowering of the temperature. The salts used in so-called freezing mixtures are those alkalies which possess the property of solubility at comparatively low temperature. Refrigeration by a chemical change is not commercially possible, as the energy expended to change the state of the resultant mixture back to its original constituents is excessive. The high cost of the materials used render imperative a continuous cycle of operation. The most common form of freezing mixture is ice and common salt Other mixtures use snow and calcium chloride, snow and sulphuric acid, snow and hydrochloric acid, sulphate of sodium and hydrochloric acid or sulphuric acid, etc. See p. 3–23 for freezing mixtures.

PHYSICAL CHANGE OF STATE.—Artificial refrigeration is produced commercially by apparatus causing a physical change of state only by mechanical process. The fol-

Table 11.—Coefficient of Heat Transmission *K* for Wrought-iron or Steel Pipe

(B.t.u. per hr. per sq. ft. per deg. temp. diff.)

Conditions	*K*
Ammonia gas inside, water outside. Submerged condenser	50
Ammonia gas inside, running water outside; atmospheric condenser	60
Ammonia gas inside, brine outside. Brine tank	25
Ammonia gas inside, air outside. Direct expansion piping	3.5
Cold brine inside, water outside. Water cooler	80
Ammonia liquor inside, water outside. Absorber	60
Ammonia liquor inside and outside. Heat exchanger	50
Steam inside, water outside. Counter current steam condenser	500
Steam inside, water or ammonia liquor outside. Ammonia generator or still	300
Steam inside, air outside	2
Brine inside, air outside. Brine piping, black pipe	
Brine inside, air outside. Brine piping, frosted pipe	

lowing systems are at present (1935) in use: 1. Cold air machines, using air as the refrigerating medium. 2. Vapor compression machines, using a volatile liquid, as ammonia (NH_3), sulphur dioxide (SO_2), carbon dioxide (CO_2), Dielene ($CH_3 Cl_2$), Carrene ($CH_2 Cl_2$), etc., as the refrigerant. 3. Steam jet vacuum machines, using water vapor as the medium. 4. Absorption machines, using ammonia as the refrigerant.

Cold-Air Machines

The operation of the cold-air machines is based on the first law of thermodynamics. If a compressed gas or air, cut off from the source of supply, expands in a cylinder, moving a piston and performing external work, heat is abstracted from the working substance, since heat and work are mutually convertible. If the expansion occurs in a non-conducting cylinder, it is adiabatic and the temperature of the air expanded in the cylinder falls. The cold expanded air is circulated through the space to be cooled, absorbing heat and producing the refrigerating effect desired. The excessive size of compressor cylinders and their relatively low efficiency, as compared with machines using saturated vapor for a refrigerating medium, has practically limited cold-air installations to shipboard. The low specific heat of air requires the circulation of large volumes and the use of bulky apparatus. The compressor cylinder capacity is approximately 16 times that required for equal duty with ammonia used as a refrigerant.

Cold-air systems should be worked on a closed cycle, the air being recirculated to keep the working temperature range low and also to avoid the loss and operating difficulty caused by freezing of moisture in outside air taken into the system. If air introduced to make up leakage loss is chilled below its dew point before introduction, its water vapor will be precipitated and the system will operate practically with dry air.

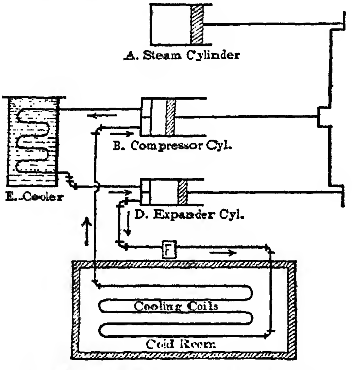


Fig. 2. Diagram of Cold Air Machine

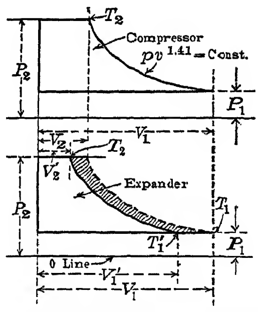


Fig. 3. Indicator Cards, Cold Air Machine

THE ALLEN DENSE AIR MACHINE.—The operating cycle above indicated is accomplished in the Allen dense air machine, Fig. 2. Air is drawn into the compressor B from the refrigerator coil at a gage pressure of from 60 to 70 lb. per sq. in. It is compressed (single stage) to from 210 to 240 lb. per sq. in., gage, and passes through a cooling coil E surrounded by water, which removes the heat of compression. The air leaves the coil at about 10° F. higher than the final temperature of the cooling water. The reduced volume of cooled air expands in the cylinder D which has an adjustable cut-off gear The air, in expanding, assists in driving the compressor, thereby recovering a large portion of the external work of compression. The expansion of the air causes its temperature to fall, and it then passes to the coils of the room to be refrigerated, returning thence to the compressor cylinder. An oil extractor F in the discharge line permits the frozen oil to be collected and melted. Make-up air required to keep the system fully charged is compressed in a small cylinder and discharged into cooler E, where its temperature is lowered and moisture precipitated before it is introduced into the system.

The operation of the cold-air system is explained by formulas below. Compression of the air and its re-expansion are assumed to take place adiabatically (Fig. 3).

Compression.—Let v_1 = volume of air drawn into compressor cylinder per min.; T_1 = absolute initial temperature of this air and p_1 = its initial pressure. T_1 is about

Table 12.—Tests on Cold-air Machines

(Linde, *Trans.* A.S.M.E., xiv, p. 1416)

	System		
	Bell-Coleman	Lightfoot	
Air pressure in receiver, lb. per sq. in., absolute	61.0	65.0	64.0
Temp. of air entering compression cylinders, deg. F	65.5	62.0	
Temp. of air after expansion, deg. F	−52.6	−82.0	−85.0
I.Hp. in compression cylinder	124.5	43.1	346.4
I.Hp. in expansion cylinder	58.5	28.0	176.2
I.Hp. in steam cylinder	84.4	24.6	332.7
B.t.u. abstracted per hour per I.Hp. of steam cylinder, at 20° F	668.0	1554.0	954.0

10° below the temperature to be maintained in the refrigerator. Let v_2, p_2 and T_2 be the volume, pressure and absolute temperature, respectively, of the air after compression All pressures are lb. per sq. in., absolute. Then

$$T_2 = T_1 (p_2/p_1)^{0.29};\ v_2^{1.41} = p_1 v_1^{1.41}/p_2 \quad [4]$$

Work of compression, ft.-lb. per min., is $w = 3.45 (p_2 v_2 - p_1 v_1) \times 144$ . . . [5]

Cooling.—Volume v_2 is reduced to volume v'_2 in the cooler at constant pressure.

$$v'_2 = v_2(T'_2/T_2) \quad [6]$$

The B.t.u. imparted to the cooling water per lb. of air circulated is

$$H = C_{pa}(T_2 - T'_2) \quad [7]$$

where $C_{pa} = 0.24$ = specific heat of air at constant pressure.

Expansion Cylinder.—The reduced volume v'_2 expands adiabatically to v'_1 and pressure p'_1. Then

$$(v'_2)^{1.41} = p_1(v'_1)^{1.41}/p'_2 \quad [8]$$

Work recovered by expansion, ft.-lb., is $w' = 3.45(p_2 v'_2 - p_1 v'_1) \times 144$, . . [9]

Final absolute temperature is $= T'_1 = T'_2(p_1/p_2)^{0.29}$. . . . . . [10]

Expansion in Refrigerator Coils.—Volume v'_1 expands in the refrigerator coil at constant pressure p_1, returning to original volume v_1 and temperature T_1. $v_1 = v'_1(T_1/T'_1)$ . . [11] The shaded portion of Fig. 3 gives the refrigerating effect (heat removed = H) per cu. ft. of piston displacement.

$$H = C_{pa}\, d(T_1 - T'_1)\ \text{B.t.u.} \quad [12]$$

where d = density, lb. per cu. ft. of air at temperature T_1 and pressure p_1. The density for any pressure is given by the characteristic equation of gases $PV = MRT$, where P = absolute pressure, lb. per sq. ft., T = absolute temperature, and R = 53.35 for air. If M is 1 lb. and v is the volume, cu. ft., $Pv = Rt$ or $1/v = P/RT$. But $1/v = d = P/RT$ which gives the density in terms P, R and T.

Compressor Displacement Required per Ton of Refrigeration.—Let D = displacement of compressor per 24 hr. per ton of refrigeration, cu. ft., E = volumetric efficiency of compressor (80% approximately). Then

$$D = 288{,}000 \div \{d \times C_{pa}(T_1 - T'_1) \times E\} \quad [13]$$

and the net horsepower required is

$$33{,}000 = \frac{\{3.45(p_2 v_2 - p_1 v_1 - p_2 v'_2 + p_1 v'_1) \times 144\}}{33{,}000} \quad [14]$$

Vapor Compression Machines

The media generally used in compression machines, ammonia, sulphur dioxide, and carbon dioxide, etc., exist only as a gas or vapor at atmospheric pressures and ordinary temperatures, but they are liquefied when compressed to a sufficiently high pressure and cooled. The heat absorbed in re-evaporating the liquid at a reduced pressure constitutes the refrigerating effect. To periodically return the refrigerating medium to its original liquid state, the system must comprise the following parts: 1. Evaporating coils, wherein the liquid is evaporated, absorbing heat from its surroundings and producing the refrigerating effect. 2. The compressor, in which vapor from the evaporating coils is compressed into the condenser at a terminal pressure corresponding to the temperature of the saturated vapor obtainable with the cooling water available. 3. The condenser, in which the latent heat and heat of compression is removed and the vapor liquefied by cooling water circulated through or over the condenser pipes or tubes.

MEDIA.—The choice of vapor depends on: 1. Pressure range corresponding to temperature to be maintained in condenser and evaporator. 2. Volume of medium to be

Table 13.—Physical Properties of Various Refrigerants

	Ammonia, NH_3	Carbon Dioxide, CO_2	Sulphur Dioxide, SO_2	Propane, C_3H_8	Methyl Chloride, CH_3Cl	Ethyl Chloride, C_2H_5Cl	Dichloro-ethylene, $C_2H_2Cl_2$ (Dielene)	Trichloro-ethylene, C_2HCl_3	Dichlor-methane, CH_2Cl_2 (Carrene)	Dichlorodi-fluro-methane, CCl_2F_2 (Freon)
At 86 deg. F.										
Absolute pressure, lb. per sq. in.	169.2	1039.0	66.45	158.0	95.53	27.10	6.9	1.72	10.6	107.9
Volume of vapor, cu. ft. per lb.	1.772	0.0474	1.185	0.681	1.075	3.29	8.5	25.2	6.68	0.389
Density of liquid, lb. per cu. ft.	37.16	37.41	84.44	30.37	55.8	54.88	...	...	...	80.6
Heat, B.t.u. per lb.										
Of liquid	138.9*	45.45‡	42.12*	51.0†	59.34*	23.1‡	23.2†	20.1†	...	27.7*
Latent	492.6	27.00	142.80	144.0	162.09	162.6	113.0	109.5	143	59.7
Of vapor	631.5	72.46	184.92	195.0	222.25	185.7	156.2	129.6	...	87.4
Specific heat, vapor, C_p	0.79	0.2025	0.154	0.365	0.24	0.273	0.1625	0.12	0.154	...
Specific heat, vapor, C_v	0.4011	0.1558	0.123	0.316	0.20	...	0.1425	0 105	0.128	...
At 5 deg. F.										
Absolute pressure, lb. per sq. in.	34.27	334.4	11.81	45.2	20.89	4.65	0.82	0.16	1.17	26.5
Volume of vapor, cu. ft. per lb.	8.150	0.2673	6.421	2 34	4.530	17.06	63.0	240.0	49.9	1.49
Density of liquid, lb. per cu. ft.	41.11	61.22	92.0	34.33	61.0	59.00	79.0	91.6	...	90.0
Heat, B.t.u. per lb.										
Of liquid	48.35	−13.16	14.11	3.0	21.19	−11.6	1.35	1.17	...	9.3
Latent	565.0	115.30	169.38	169.5	178 50	177.0	136.0	112.5	162	69.5
Of vapor	613.35	102.14	183.49	172.0	199.76	165.4	37.25	113.67	...	78.8
Specific heat of vapor, C_p	0.91	0.2025	0.154	0.365	0.24	0.273	0.1625	0.12	0.154	...
Specific heat of vapor, C_v	0.4011	0.1558	0.123	0.316	0.20	...	0.1425	0.105	0.128	...
$n = C_p/C_v$	1.3	1.31	1.25	1.16	1.20	...	1.14	1.14	1.21	...
Critical pressure, lb. per sq. in.	...	1071.0	1141.5	661.5	969.2	784.0	800	...	6.40	...
Critical temperature, deg. F	...	87.8	314.8	204.1	289.6	361.0	470	...	4.21	...
Molecular weight	131.35	44.0	64.065	49.062	50.481	64.497	96.9	131.35	84.9	...
Boiling point (1 atm.), deg. F.	187	−108.4	14.00	−48.1	−10.66	53.96	122	187	105	...

* Above 40° F. † Above 0° F. ‡ Above 32° F.

drawn into compressor to produce a given amount of refrigeration. This determines the displacement of the compressor used.

Table 13 (condensed from Data Book, A.S.R.E., 1933 ed.) gives data on commonly used refrigerants. The Data Book contains tables of thermal properties of isobutane, butane, methyl chloride, ethane and dichlorodifluoromethane (freon, F-12).

Ammonia (NH_3) is used most generally in cold storage and ice-making plants; for the temperature range in practice between evaporator and condenser it does not require the compressor to handle large volumes of gas, and condensing pressure is approximately 180 lb. per sq. in. for usual condensing water temperatures. Ammonia gas from a safety standpoint, in case the gas escapes, is not favored for cooling auditoriums, installations on ships, etc., where an odorless refrigerating equipment may be required; carbon dioxide (CO_2) often is preferred.

Carbon Dioxide operating pressures are relatively high, about 1000 lb. per sq. in. for usual condensing water temperatures; steel compressor cylinders are required. CO_2 machines are used in chemical processes requiring low temperatures, which NH_3 machines cannot meet without going below atmospheric pressure on suction (evaporation) side of machine, allowing air and its contained vapor to enter. Air, being non-condensible at temperatures and condenser pressures available, would lower efficiency and capacity.

Compressed CO_2, so-called *dry ice*, is used as a refrigerant in shipment of ice cream, fish and meats It completely disappears when vaporized and has high refrigerating effect for its weight.

Sulphur Dioxide (SO_2) is not used for evaporator temperatures below about 20° F. Another objection is that SO_2 forms highly corrosive sulphurous acid on contact with moisture in in-leaking air. For usual temperature ranges, however, SO_2 is very satisfac-

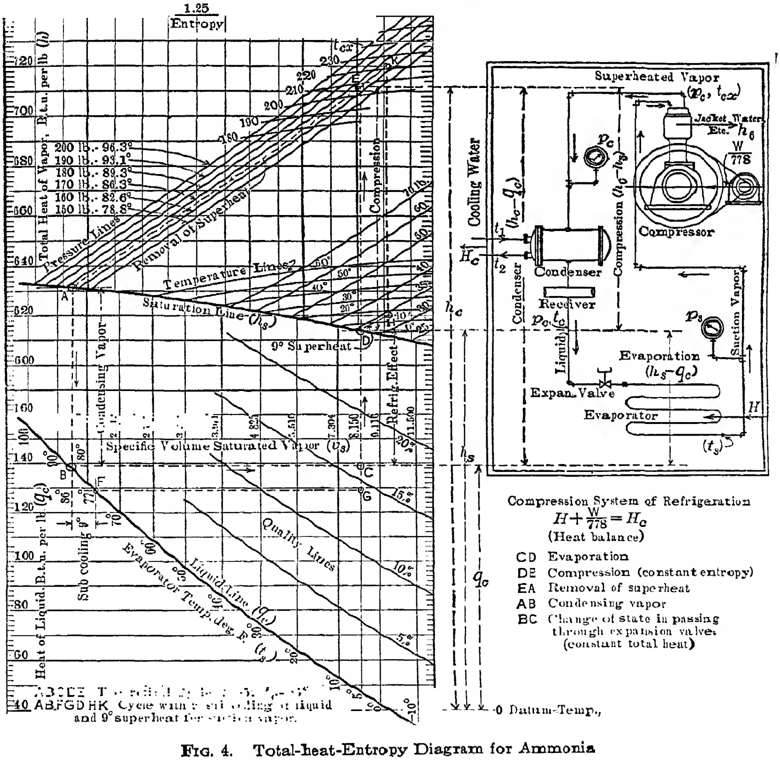


Fig. 4. Total-heat-Entropy Diagram for Ammonia

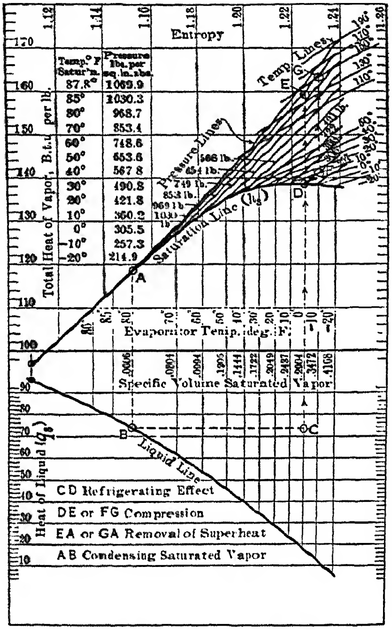


Fig. 5. Total-heat-Entropy Diagram for Carbon Dioxide

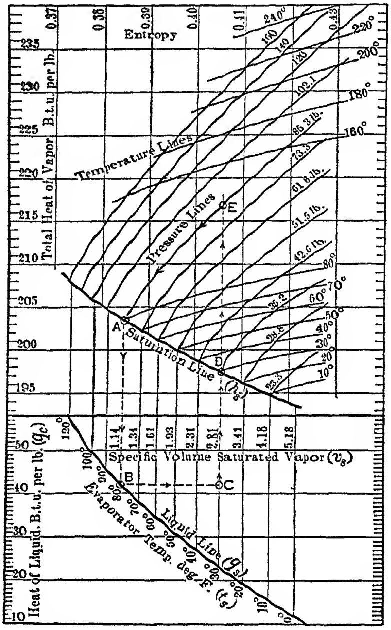


Fig. 6. Total-heat-Entropy Diagram for Methyl Chloride

Table 14.—Thermal Properties of Liquid and Saturated Ammonia

Figures in bold face are for *standard ton* conditions. This table and Table 15 were condensed from U. S. Bureau of Standards Tables for Ammonia. See Fig. 4

Temp., deg. F. (t)	Pressure, lb. per sq. in., abs. (p)	Specific Volume of Vapor (v_s)	Heat Content Above − 40° F., B.t.u. per lb. Liquid (q)	Latent (r)	Total (h_s)	Entropy of Vapor (s)
− 60	5.55	44.73	− 21.2	610.8	589.6	1.4769
− 50	7.67	33.08	− 10.6	604.3	593.7	1.4497
− 40	10.41	24.86	0.0	597.6	597.6	1.4242
− 30	13.90	18.97	10.7	590.7	601.4	1.4001
− 28	14.71	18.00	12.8	589.3	602.1	1.3955
− 20	18.30	14.68	21.4	583.6	605.0	1.3774
− 10	23.74	11.50	32.1	576.4	608.5	1.3558
0	30.42	9.116	42.9	568.9	611.8	1.3552
5	**34.27**	**8.150**	**48.3**	**565.0**	**613.3**	**1.3253**
10	38.51	7.304	53.8	561.1	614.9	1.3157
15	43.14	6.562	59.2	557.1	616.3	1.3062
20	48.21	5.910	64.7	553.1	617.8	1.2969
25	53.73	5.334	70.2	548.9	619.1	1.2879
30	59.74	4.825	75.7	544.8	620.5	1.2790
40	73.32	3.971	86.8	536.2	623.0	1.2618
50	89.19	3.294	97.9	527.3	625.2	1.2453
60	107.6	2.751	109.2	518.1	627.3	1.2294
70	128.8	2.312	120.5	508.6	629.1	1.2140
77	145.4	2.055	128.5	501.7	630.2	1.2035
80	153.0	1.955	132.0	498.7	630.7	1.1991
86	**169.2**	**1.772**	**138.9**	**492.6**	**631.5**	**1.1904**
90	180.6	1.661	143.5	488.5	632.0	1.1846
95	195.8	1.534	149.4	483.2	632.6	1.1775
100	211.9	1.419	155.2	477.8	633.0	1.1705

tory and is used in small electric refrigerators for homes, etc. The condenser pressure for SO_2 is less than 80 lb. per sq. in. for usual condenser water temperatures.

Carrene (CH_2Cl_2) and **Dielene** ($C_2H_2Cl_2$), used in turbo-compressors, are odorless, water-white liquids at normal atmospheric conditions, and may be handled in open con-

Table 15.—Thermal Properties of Superheated Ammonia

v = specific volume, cu. ft. per lb.; h = heat content, B.t.u. per lb.; s = entropy superheated vapor

Temp., deg. F.	Pressures, lb. per sq. in.: Absolute, **170**, Gage **155.3** Sat. Temp., 86.29° F.			Pressures, lb. per sq. in.: Absolute, **180**, Gage **165.3** Sat. Temp., 89.78° F.			Pressures, lb. per sq. in.: Absolute, **190**, Gage **175.3** Sat. Temp., 93.13° F.		
t	v	h	s	v	h	s	v	h	s
sat.	*1.764*	*631.6*	*1.1900*	*1.667*	*632.0*	*1.1850*	*1.581*	*632.4*	*1.1802*
100	1.837	641.9	1.2087	1.720	639.9	1.1992	1.615	637.8	1.1899
110	1.889	649.1	1.2215	1.770	647.3	1.2123	1.663	645.4	1.2034
120	1.939	656.1	1.2336	1.818	654.4	1.2247	1.710	652.6	1.2160
130	1.988	662.8	1.2452	1.865	661.3	1.2364	1.755	659.7	1.2281
140	2.035	669.4	1.2563	1.910	668.0	1.2477	1.799	666.5	1.2396
150	2.081	675.9	1.2669	1.955	674.6	1.2586	1.842	673.2	1.2506
160	2.127	682.3	1.2773	1.999	681.0	1.2691	1.884	679.7	1.2612
170	2.172	688.5	1.2873	2.042	687.3	1.2792	1.925	686.1	1.2715
180	2.216	694.7	1.2971	2.084	693.6	1.2891	1.966	692.5	1.2815
190	2.260	700.8	1.3066	2.126	699.8	1.2987	2.005	698.7	1.2912
200	2.303	706.9	1.3159	2.167	705.9	1.3081	2.045	704.9	1.3007
210	2.346	713.0	1.3249	2.208	712.0	1.3172	2.084	711.1	1.3099
220	2.389	719.0	1.3338	2.248	718.1	1.3262	2.123	717.2	1.3189
230	2.431	724.9	1.3426	2.288	724.1	1.3350	2.161	723.2	1.3278
240	2.473	730.9	1.3512	2.328	730.1	1.3436	2.199	729.3	1.3365
250	2.514	736.8	1.3596	2.367	736.1	1.3521	2.236	735.3	1.3450
260	2.555	742.8	1.3679	2.407	742.0	1.3605	2.274	741.3	1.3534
270	2.596	748.7	1.3761	2.446	748.0	1.3687	2.311	747.3	1.3617
280	2.637	754.6	1.3841	2.484	753.9	1.3768	2.348	753.2	1.3698
290	2.678	760.5	1.3921	2.523	759.9	1.3847	2.384	759.2	1.3778
300	2.718	766.4	1.3999	2.561	765.8	1.3926	2.421	765.2	1.3857
310	2.758	772.3	1.4076	2.599	771.7	1.4004	2.457	771.1	1.3935
320	2.798	778.3	1.4153	2.637	777.7	1.4081	2.493	777.1	1.4012
330	2.838	784.2	1.4228	2.675	783.6	1.4156	2.529	783.1	1.4088
340	2.878	790.1	1.4303	2.713	789.6	1.4231	2.565	789.0	1.4163
350	2.918	796.2	1.4377	2.750	795.6	1.4305	2.601	795.1	1.4238
360	2.957	802.0	1.4450	2.788	801.5	1.4379	2.637	801.0	1.4311
370	2.997	808.0	1.4522	2.825	807.5	1.4451	2.672	807.0	1.4384
380	3.036	814.0	1.4594	2.863	813.5	1.4523	2.707	813.0	1.4456
390	3.075	820.0	1.4665	2.900	819.5	1.4594	2.743	819.0	1.4527
400	3.114	826.0	1.4735	2.937	825.5	1.4665	2.778	825.1	1.4598

Table 16.—Thermal Properties of Liquid and Saturated Carbon Dioxide

Boldface figures are for *standard ton* conditions. Condensed from A.S.R.E. Data Book. S

Temp., deg. F. (t)	Absolute Pressure, lb. per sq. in. (p)	Specific Volume of Vapor (v_g)	Heat Content Above −40° F., B.t.u. per lb. Liquid (q)	Latent (r)	Total (h_g)
−60	94.7	0.9270	−9.2	145.8	136.6
−50	118.2	.7492	−4.7	141.9	137.2
−40	145.8	.6113	0.0	137.8	137.8
−30	177.8	.5029	4.5	133.7	138.2
−20	214.9	.4168	9.1	129.4	138.5
−10	257.3	.3472	13.9	124.8	138.7
0	305.5	.2904	18.8	120.1	138.9
5	**331.9**	**.2660**	**21.3**	**117.5**	**138.8**
10	360.2	.2437	24.0	114.7	138.7
14	383.9	.2274	26.1	112.5	138.6
20	421.8	.2049	29.4	108.9	138.3
26	462.2	.1846	32.9	105.1	138.0
30	490.8	.1722	35.4	102.4	137.8
40	567.8	.1444	41.7	95.0	136.7
50	653.6	.1205	48.4	86.6	135.0
60	748.6	.0994	55.5	76.6	132.1
70	853.4	.08040	63.7	63.8	127.5
77	933.1	.06674	70.5	51.4	121.9
80	968.7	.06064	73.9	44.8	118.7
86	**1043.0**	**.04789**	**83.3**	**27.1**	**110.4**
87.8	1066.2	.03454	97.0	0.0	97.0

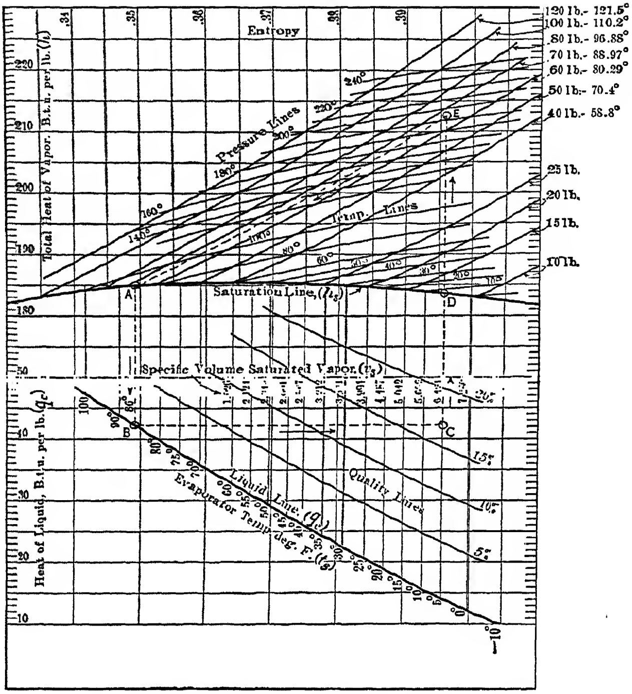


FIG. 7. Total-heat-Entropy Diagram for Sulphur Dioxide

Table 17.—Thermal Properties of Liquid and Saturated Sulphur Dioxide

Bold face figures are for *standard ton* conditions. The data in this table were first published in *Ice and Cold Storage*, London. See Fig. 7

Temp., deg. F. (t)	Absolute Pressure, lb. per sq. in. (p)	Specific Volume of Vapor (v_s)	Heat Content Above −40° F., B.t.u. per lb. Liquid (q)	Latent (r)	Total (h_s)	Entropy of Vapor (s)
−40	3.136	22.42	0.00	178.61	178.61	0.42562
−30	4.331	16.56	2.93	176.97	179.90	.41864
−20	5.883	12.42	5.98	175.09	181.07	.41192
−10	7.863	9.44	9.16	172.97	182.13	.40544
0	10.35	7.28	12.44	170.63	183.07	.39917
5	**11.81**	**6.421**	**14.11**	**169.38**	**183.49**	**.39609**
10	13.42	5.682	15.80	168.07	183.87	.39306
15	15.21	5.042	17.49	166.72	184.21	.39005
20	17.18	4.487	19.20	165.32	184.52	.38707
25	19.34	3.994	20.92	163.87	184.79	.38412
30	21.70	3.581	22.64	162.38	185.02	.38119
40	27.10	3.212	26.12	159.25	185.37	.37541
50	33.45	2.348	29.61	155.95	185.56	.36969
60	40.93	1.926	33.10	152.49	185.59	.36405
70	49.62	1.590	36.58	148.88	185.46	.35846
77	56.55	1.371	39.00	146.82	185.27	.35458
80	59.68	1.321	40.05	145.12	185.17	.35291
86	**66.45**	**1.185**	**42.12**	**142.80**	**184.92**	**.34954**
90	71.25	1.104	43.50	141.22	184.72	.34731
100	84.52	0.9262	46.90	137.20	184.10	.34173

Table 18.—Thermal Properties of Superheated Sulphur Dioxide

(From A.S.R.E. Data Book)

v = specific volume, cu. ft. per lb.; h = heat content, B.t.u. per lb.; s = entropy of superheated vapor

Temp., deg. F.	Absolute Pressure, **40** lb. per sq. in. Sat. Temp., 58.83° F.			Absolute Pressure, **50** lb. per sq. in. Sat. Temp., 70.40° F.			Absolute Pressure, **60** lb. per sq. in. Sat. Temp., 80.29° F.			Absolute Pressure, **70** lb. per sq. in. Sat. Temp., 88.97° F.		
t	v	h	s	v	h	s	v	h	s	v	h	s
sat.	*1.970*	*185.60*	*0.36470*	*1.577*	*185.45*	*0.35826*	*1.3144*	*185.16*	*0.35272*	*1.125*	*184.77*	*0.34789*
100	2.246	196.1	0.38415	1.775	193.9	0.37369	1.288	191.4	0.36403	1.181	187.6	0.35443
110	2.304	198.3	.38810	1.825	196.4	.37815	1.346	194.3	.36906	1.228	191.6	.36020
120	2.360	200.4	.39183	1.872	198.8	.38234	1.403	197.0	.37375	1.272	194.8	.36545
130	2.413	202.5	.39541	1.917	201.1	.38627	1.459	199.5	.37810	1.313	197.6	.37028
140	2.465	204.6	.39881	1.961	203.3	.38998	1.514	201.9	.38217	1.352	200.3	.37478
150	2.515	206.5	.40209	2.003	205.4	.39353	1.563	204.2	.38603	1.389	202.9	.37897
160	2.565	208.5	.40525	2.044	207.5	.39691	1.608	206.5	.38963	1.424	205.3	.38291
170	2.614	210.4	.40831	2.084	209.6	.40015	1.650	208.6	.39310	1.457	207.6	.38662
180	2.662	212.3	.41127	2.123	211.6	.40327	1.689	210.7	.39639	1.489	209.9	.39014
190	2.709	214.2	.41416	2.161	213.4	.40628	1.726	212.8	.39956	1.521	212.0	.39348
200	2.755	216.0	.41694	2.199	215.4	.40919	1.751	214.8	.40260	1.551	214.1	.39670
210	2.800	217.9	.41966	2.237	217.3	.41200	1.785	216.8	.40554	1.580	216.1	.39978
220	2.845	219.7	.42233	2.274	219.2	.41477	1.819	218.7	.40839	1.608	218.1	.40275
230	2.889	221.5	.42494	2.311	221.1	.41748	1.853	220.7	.41118	1.636	220.1	.40564
240	2.933	223.3	.42751	2.347	223.0	.42015	1.885	222.6	.41391	1.664	222.1	.40845
250	2.977	225.1	.43007	2.383	224.9	.42275	1.917	224.5	.41657	1.691	224.1	.41120
260	3.021	227.0	.43262	2.418	226.7	.42535	1.948	226.4	.41917	1.718	226.0	.41389

Temp., deg. F.	Absolute Pressure, **80** lb. per sq. in. Sat. Temp., 96.88° F.			Absolute Pressure, **100** lb. per sq. in. Sat. Temp., 110.15° F.			Absolute Pressure, **120** lb. per sq. in. Sat. Temp., 121.52° F.			Absolute Pressure, **140** lb. per sq. in. Sat. Temp., 131.64° F.		
t	v	h	s	v	h	s	v	h	s	v	h	s
sat.	*0.9809*	*184.33*	*0.34357*	*0.7786*	*183.30*	*0.33603*	*0.6430*	*182.190*	*0.33954*	*0.5451*	*181.04*	*0.32388*
140	1.163	198.6	0.36819	0.8928	194.6	0.35528	0.7085	190.1	0.34264	0.5734	185.1	0.33089
150	1.199	201.3	.37270	.9255	197.9	.36061	.7403	193.9	.34904	.6055	189.7	.33777
160	1.232	203.9	.37692	.9561	200.9	.36558	.7700	197.4	.35484	.6345	193.6	.34442
170	1.263	206.4	.38093	.9848	203.7	.37009	.7972	200.6	.36012	.6613	196.3	.35041
180	1.292	208.7	.38461	1.012	206.4	.37431	.8228	203.7	.36494	.6861	200.8	.35588
190	1.320	211.0	.38813	1.038	209.0	.37829	.8470	206.7	.36936	.7092	204.0	.36088
200	1.347	213.3	.39150	1.062	211.5	.38203	.8699	209.4	.37348	.7309	207.1	.36548
210	1.374	215.5	.39471	1.086	213.8	.38556	.8916	212.0	.37737	.7513	210.0	.36976
220	1.400	217.5	.39780	1.109	216.1	.38892	.9124	214.5	.38104	.7707	212.7	.37379
230	1.426	219.6	.40079	1.131	218.4	.39214	.9324	217.0	.38451	.7892	215.4	.37758
240	1.451	221.6	.40369	1.152	220.5	.39524	.9515	219.3	.38785	.8070	217.9	.38118
250	1.476	223.6	.40651	1.173	222.6	.39824	.9700	221.5	.39106	.8241	220.3	.38461
300	1.593	233.4	.41974	1.268	232.8	.41207	1.056	233.2	.40558	.9017	231.5	.39985

tainers. Dielene is slightly combustible; its flame generally is self-extinguishing. Carrene, which may be interchangeably used in the same system, is non-combustible. The relatively large volume of gas necessary to produce refrigerating effect makes their use in reciprocating machines impractical. These refrigerants always operate below atmospheric pressure, and the apparatus requires more or less continuous removal of air, which carries with it some refrigerant. The loss is not serious, and replacement cost probably does not exceed that of other refrigerants which operate above atmospheric pressure. Freon, F-12 (CCl_2F_2) is used in connection with surface-type, direct expansion coolers, used for air conditioning. Ordinarily, condensing pressure does not exceed 120 lb. per sq. in. with usual available condenser cooling water temperatures.

Water-vapor, used with a turbo-compressor or a steam-jet ejector-type thermo-compressor, is used to cool water for industrial uses and air conditioning. Its application is limited to cooling to approximately 40° F. minimum when steam is available. As a large volume of water-vapor is drawn from the evaporator, a reciprocating compressor is impractical.

TOTAL-HEAT-ENTROPY DIAGRAMS FOR AMMONIA, CARBON DIOXIDE, SULPHUR DIOXIDE, METHYL CHLORIDE AND FREON (Figs. 4 to 8, by the author, *Heating, Piping & Air Conditioning Magazine*, Aug., 1935) may be used to approximate refrigerating effect R and the value of M_m or M_r, heat equivalent of the work of compression $(h_c - h_s)$ or $(h_c - h_1)$, and horsepower per ton of refrigeration. To compress the diagrams to fit the page, the heat content scale has been shortened. Upper scale is for vapor and lower scale is for liquid. On the NH_3 diagram, standard conditions (5° evaporator temp. and 86° condenser temp.) for the theoretical Rankine cycle are shown; also standard machine rating conditions with 9° subcooling of liquid and 9° superheat for suction vapor.

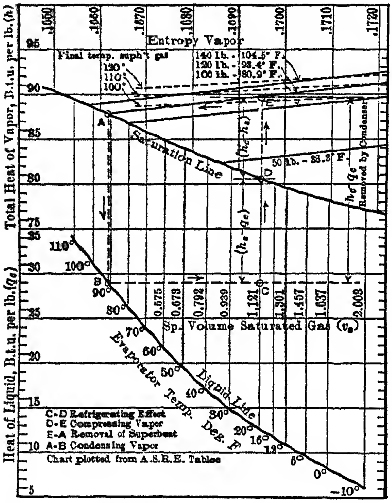


FIG. 8. Total-heat-Entropy Diagram for Freon

For NH_3, and theoretical Rankine cycle, evaporator pressure for 5° is 34.27 lb. per sq. in., abs.; condenser pressure for 86° is 169.2 lb. per sq. in., abs. $q_c = 138.9$ B.t.u.; $h_s = 613.3$ B.t.u. $v_s = 8.15$ cu. ft.; $h_c = 711.3$ B.t.u. $M_r = 200/(613.3 - 138.9) = 0.422$ lb. per ton per min.

Hp. $= 0.422 \times (711.3 - 613.3)/42.5 = 0.973$ per ton. Volume of saturated vapor to be pumped per min. per ton is $V = Mv_s = 0.422 \times 8.15 = 3.44$ cu. ft.

For SO_2 and theoretical Rankine cycle, evaporator pressure for 5° is 11.81 lb. per sq. in., abs.; condenser pressure for 86° is 66.45 lb. per sq. in., abs.; $q_c = 42.12$; $h_s = 183.49$; $v_s = 6.421$; $h = 213$; $M_r = 200/(183.49 - 42.12) = 1.414$ lb. per ton per min.; Hp. $= 1.414\ (213 - 183.5)/42.5 = 0.98$ per ton; $V = 1.414 \times 6.421 = 9.08$ cu. ft. of saturated vapor to be pumped per min. per ton.

For CO_2, and theoretical Rankine cycle, evaporator pressure for 5° is 332 lb. per sq. in., abs.; condenser pressure for 86° is 1043 lb. per sq. in., abs.; $q_c = 83.3$; $h_s = 138.75$; $v_s = 0.266$; $h = 159$; $M_r = 200/(138.75 - 83.3) = 3.60$ lb. per ton per min.; Hp. $= 3.60\ (159 - 138.75)/42.5 = 1.70$ per ton; $V = 3.60 \times 0.266 = 0.958$ cu. ft. of saturated vapor to be pumped per ton per min.

The lines joining points $ABCDE$ in Figs. 4 to 8 represent various states of the refrigerant as it passes through the apparatus for the theoretical Rankine cycle with no subcooling or superheat in the suction gas. The lines $ABFGDHK$ are for the same cycle with 9° F. subcooling of liquid and 9° superheat in suction gas. The following description refers to cycle without subcooling or suction superheat $(ABCDE)$:

Liquid at 86° F. and corresponding condenser pressure p_c, point B, is throttled to the lower pressure p_e, point C, corresponding to evaporator temperature (5° F.), as it passes the expansion valve. Heat-content during throttling remains constant. The remaining liquid evaporates to dry saturated vapor in the expansion coils and absorbs the difference

between heat content at saturation for 5°, point C, and heat content of mixture at point D, which is the refrigerating effect R per lb. of refrigerant circulated. Compression begins at D; it is complete when condenser pressure p_c is reached at E. Adiabatic compression is assumed, and the heat equivalent of the work done in compressing 1 lb. of vapor is difference between heat content at D and E. Entropy remains constant at an adiabatic change of state. The compressed vapor condenses at constant condenser temperature 86° and corresponding pressure p_c due to removal of heat of compression and its latent heat by the condenser cooling water. Heat to be removed per pound of refrigerant is difference between heat content of superheated vapor at E and heat content of liquid at B.

Using the heat-content-entropy diagrams, the values of Table 19 were obtained for standard machine rating conditions.

VAPORIZATION OF A LIQUID AND "REFRIGERATING EFFECT" R.—The liquid to be used as a refrigerating medium should evaporate at a relatively low temperature (low boiling point) in order to extract heat from its surroundings during the evaporating period. Let t_c = temperature of condensed liquid in the receiver (Fig. 9); t_s = temperature of saturated vapor in the evaporating coils; v_s specific volume of saturated vapor at temperature t_s; q_c = heat of the liquid for temperature t_c; h_s = heat content at temperature t_s. Then the heat R_1 in B.t.u. per lb., which the medium is capable of extracting from the surroundings will be the refrigerating effect, and $R_1 = (h_s - q_c)$, the initial state being liquid and the assumed final state dry saturated vapor. The volume, in cubic feet, of dry saturated vapor leaving the evaporating coil per ton of refrigeration per minute is $V = (200 \times v_s)/R_1$. If the vapor is allowed to superheat in the evaporator, heat content becomes h_1, and the refrigerating effect is $(h_1 - q_c)$ B.t.u. per lb. Superheating increases the refrigerating effect.

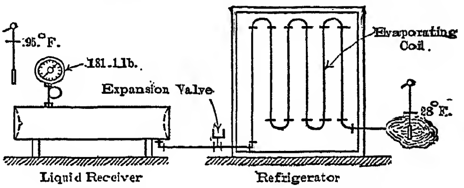


FIG. 9. Diagram of Refrigerating Effect

EXAMPLE.—If 1 lb. of liquid ammonia at the temperature of the outside air, 95° F., flows through a needle valve from a drum into the evaporating coil, Fig. 9, one end of which is open to the atmosphere, the liquid will evaporate into dry saturated vapor at atmospheric pressure. The pressure existing in the drum (Table 14) will be 181.1 lb. per sq. in., gage, and the heat of the liquid q_c at this temperature will be 149.4. From Table 14, the boiling point of the liquid is −28° F. and the heat of the saturated vapor is 602.1 B.t.u. per lb. = h_s. The refrigerating effect

$$R_1 = (602.1 - 149.4) = 452.7 \text{ B.t.u.}$$

Refrigerating effect of 1 cu. ft. of vapor is $R_2 = R_1/v_s = 452.7/18 = 25.2$ B.t.u.

THE WEIGHT OF REFRIGERANT M_r to be circulated per minute per ton of refrigeration per 24 hours is $M_r = 288{,}000/R_1 \times 24 \times 60 = 200/(h_s - q_c)$ lb. The volume of saturated vapor to be pumped per ton per min. is $G = (M \times v_s)$ cu. ft. Actually the volume may be somewhat greater, as the vapor usually enters the machine with some superheat. If the liquid is subcooled below the temperature of the saturated vapor corresponding to the condenser pressure, the weight to be pumped is less, due to greater refrigerating effect of the vapor.

VAPOR COMPRESSION CYCLE.—The four essential parts of a vapor refrigerating system are shown in Fig. 10. The complete system comprises: 1. An expansion or regulating valve through which liquid refrigerant is throttled from condenser pressure to evaporator pressure. 2. The evaporator, pressure in which corresponds to that tempera-

Table 19.—Standard Machine Rating Conditions

Evaporator temp., 5° F., condenser temp., 86° F; 9° F. subcooling of liquid; 9° F. superheat in suction gas; temp. of liquid entering expansion valves, 77° F.; of vapor at beginning of compression, 14° F.

	NH_3	SO_2	CO_2
p_c, lb. per sq in., absolute	169.2	66.45	1043
p_e, lb. per sq. in., absolute	34.27	11.81	332
q, B.t.u.	128.5	39.01	70.5
h_1, B.t.u.	619.5	185.3	140.7
h_c, B.t.u.	721.0	215.2	162.0
M_m, lb. per min. per ton	0.407	1.37	2.80
Theoretical Hp. per ton	0.97	0.96	1.42*

*Carbon dioxide may be subcooled, in a double-pipe condenser, to a temperature below that corresponding to the condenser pressures, resulting in a reduction in the Hp. per ton.

ture of saturated vapor as may be required for refrigeration. 3. The compressor, where vapor from evaporator is compressed to pressure required for liquefaction. This corresponds to temperature of the condenser cooling medium, usually water, although air often is used for small house refrigerators with certain refrigerants. 4. The condenser, where superheat due to temperature rise of the vapor during compression is removed, and the vapor condensed. Sometimes a liquid cooler is added to further cool (subcool) liquid below saturation temperature corresponding to condenser pressure.

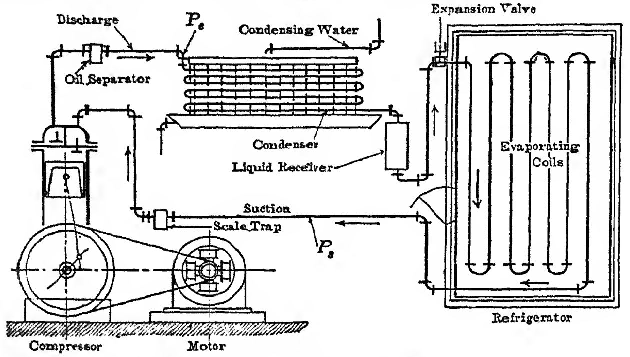


FIG. 10. Diagram of Vapor Compression System

The theoretical Rankine cycle, Fig. 11, assumes vapor to enter the compressor dry and saturated, corresponding to temperature t_s and pressure p_s of evaporator. The compression assumed is adiabatic (constant entropy) to pressure p_c corresponding to temperature of liquid leaving the condenser t_c. Let h_s = heat content (enthalpy) of the saturated vapor corresponding to t_s and p_s, B.t.u. per lb.; h_c = heat content of superheated vapor at end of compression, B.t.u. per lb.; heat equivalent Q of external work of compression is $(h_c - h_s)$ per lb. Theoretical horsepower of the compressor, based on the Rankine cycle is, Hp. = $M_r\ (h_c - h_s)/42.5$, where M_r = weight of refrigerant circulated per min.; 42.5 (or 33,000/778) is the B.t.u. heat equivalent per min. of 1 Hp. The expected B.Hp. of the compressor may be assumed as approximately 30% more than the theoretical. The values of h_s and h_c are taken from the saturated and superheated vapor tables or from a Mollier diagram for the refrigerant used. The value of h_c may be approximated by $\{h + C_p(t - t_c)\}$, where C_p mean specific heat of vapor at constant pressure for the superheated vapor between temperature t_s and final temperature t at the end of compression. h = heat content of saturated vapor corresponding to p_c. The temperature t at end of compression may be approximated by the equation

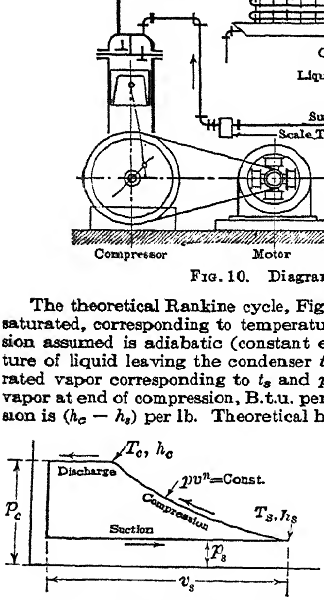


FIG. 11. Theoretical Rankine Cycle

$$t + 460 = (t_s + 460)\ (p_c/p_s)^{(n-1)/n} \quad . \ . \ . \ . \ . \ . \ . \ . \ [15]$$

See Table 13 for value of n for various refrigerants, and Fig. 12 for mean specific heat of superheated ammonia gas.

The heat equivalent of the work of compression per lb. of refrigerant circulated may be approximated by

$$Q = h_c - h_s = (n/n - 1)\ p_s v_s[1 - (p_c/p_s)^{(n-1)/n}] \times (144/778) \text{ B.t.u. per lb.} \quad . \ [16]$$

where p_s and v_s are respectively the pressure, lb. per sq. in., and volume, cu. ft. per lb., of saturated vapor entering compressor. Table 13 gives values of n for various refrigerants. See standard works on thermodynamics for derivation of equations for t and Q.

HEAT BALANCE FOR VAPOR COMPRESSION CYCLE (Fig. 4).—The heat H_c absorbed by cooling water used in the condenser (and liquid cooler, if used) must equal heat absorbed by the evaporator H plus heat equivalent of the work of compression $W/778$ neglecting any gain or loss of heat by radiation or convection to or from the apparatus. $H_c = H + (W/778)$ per lb. of refrigerant circulated. W = work of compression, ft.-lb. Heat absorbed by evaporator per lb. of refrigerant circulated equals difference

between the heat content (enthalpy) of liquid leaving condenser (or liquid cooler, if used) and heat content of vapor at beginning of compression, h_1. If the liquid is not subcooled, q equals the heat content of the liquid q_c corresponding to condenser pressure p_c for saturated vapor. If the liquid is subcooled, q equals heat content corresponding to final temperature of liquid leaving the liquid cooler. For machine rating conditions, the liquid is assumed to be subcooled 9° F.

In the Rankine cycle, vapor at beginning of compression is assumed saturated, or $h_1 = h_s$. In standard rating of refrigerating machines, vapor at beginning of compression is assumed with 9° superheat or $h_1 = h_s + C_p\ (t_s + 9)$; C_p = mean specific heat of vapor corresponding to the temperature range. $H = (h_s - q_c)$ for the theoretical Rankine cycle, $= (h_1 - q)$ for machine rating conditions.

The weight of refrigerant to be circulated per min. per ton of refrigeration is: $M_r = 200/(h_s - q_c)$ for the theoretical Rankine cycle; and $M_m = 200/(h_1 - q)$ for machine rating conditions. Subcooling reduces the weight of refrigerant to be circulated and the power requirement for compression.

The heat equivalent of the work of compression is $W/778 = (h - h_s)$ for the theoretical Rankine cycle, and $W/778 = (h - h_1)$ for machine rating conditions with superheat for the vapor entering compressor from the evaporator. For the theoretical Rankine cycle compression horsepower per ton is Hp. $= M\ (h_c - h_s)/42.5$ $(42.5 = 33{,}000/778)$. The complete refrigerating cycle is best illustrated by indicating it on the heat content (enthalpy)-entropy diagram. See Figs. 4 to 8.

LOSSES WITH VAPOR REFRIGERATING CYCLE.—One inherent loss is due to heat of the liquid. It equals the difference between heat of liquid q_c leaving condenser and heat of liquid q_e corresponding to evaporator temperature. The refrigerating effect of a small portion of the evaporating gas cools, from condenser to evaporator temperature, the liquid passing through the expansion valve.

Another loss is due to heat added (cylinder heating) by compression, which superheats the gas to a temperature well above saturation temperature corresponding to condenser pressure. If it were possible to remove this heat as rapidly as formed, the compression would be isothermal instead of adiabatic as assumed.

The sum of these losses, *i.e.*, liquid expansion plus superheat, frequently is used as the criterion in comparing the percentage of power losses for various refrigerating media.

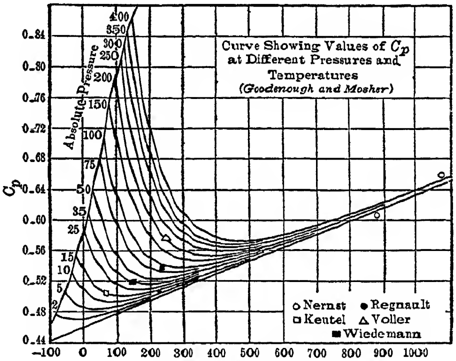


Fig. 12. Mean Specific Heat of Superheated Ammonia

COMPOUND OR STAGE VAPOR COMPRESSORS in refrigerating machines are analogous to stage air compressors with intercoolers. In refrigerating machines the intercooler between stages must be cooled by the refrigerant, due to the comparatively low temperature of the vapor entering the cylinder on the suction stroke. Some saving in power input may result in plants using more than one suction pressure and temperature.

COMBINED OR BINARY VAPOR REFRIGERATION SYSTEMS.—Ammonia is not well adapted as a refrigerant for the low temperatures used in oil refineries, quick food freezing, etc. (−50° to −75° F.), as the suction side of the system must work below atmosphere. CO_2 is well adapted to low temperatures. The power required per ton of refrigeration is somewhat greater with CO_2 than with NH_3. A combination of a CO_2 and an NH_3 system is satisfactory and economical of power input. The low temperature evaporating coil is part of the CO_2 system, the condenser being cooled by the evaporating coil of the NH_3 system. The NH_3 condenser is cooled by condensing water.

VOLUME CONTROL FOR MOTOR-DRIVEN COMPRESSORS.—A varying refrigerating load requires pumping a varying weight and volume of gas. Steam-driven compressors do this by varying engine speed. With compressors driven by constant-speed motors, another means of varying the weight of vapor pumped is required. Throttling the suction decreases capacity, but at the expense of increased power.

One method varies and reduces compressor capacity by clearance pockets around the

compressor cylinder, which communicate through auxiliary valves with the clearance space. The pockets increase the clearance volume by fixed percentages of cylinder volume, and vary the weight of vapor in the clearance space, and consequently the piston displacement volume available for drawing in vapor from the evaporator on the suction stroke. Some machines obtain 100%, 50% and zero capacity. The clearance pockets provide means of varying compressor capacity without increasing power requirement.

TURBO OR CENTRIFUGAL VAPOR COMPRESSORS.—Refrigerating media such as Dielene, Carrene and water vapor, are well adapted to turbo-compressors, the relative volume of the first two media to be handled being approximately 8 times that of ammonia for corresponding evaporator temperatures.

With standard ton conditions (5° F. evaporator and 86° F. condenser temperature), absolute pressures with Dielene are 1.7 in. Hg and 14 in. Hg; for Carrene, 2.39 in. Hg and 20.5 in. Hg; the complete system always operates below atmospheric pressure. The compressor is motor-driven at about 3300 r.p.m. A 5-stage machine operating at this speed is used in a number of plants for an evaporator temperature of approximately 39° F., with corresponding evaporator pressure of 24.8 in. Hg. The construction of the machines is similar to standard makes of multi-stage turbo air compressors. Calculations involved in design are similar, bearing in mind the different value of ratio n for the compression curve.

VOLUMETRIC EFFICIENCY OF RECIPROCATING COMPRESSORS is here defined as ratio of actual weight of vapor handled by a compressor in unit time, to weight of vapor computed from piston displacement, based on temperature and pressure conditions in suction line at compressor. The actual weight of vapor handled will be somewhat less than the theoretical, due to re-expansion of vapor in clearance space of the compressor, and superheating of vapor in passing the warmer intake valves and cylinder walls. Also the higher the discharge pressure, the greater is the re-expansion of vapor in the clearance from the previous stroke, with lower volumetric efficiency.

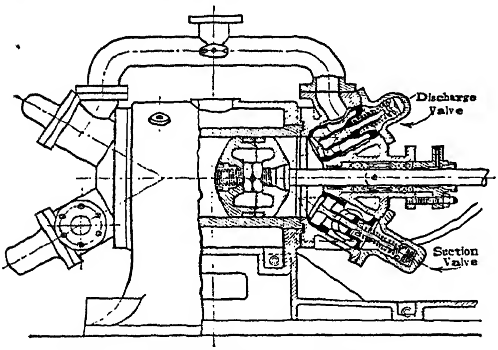


Fig. 13. Double-acting Compressor Cylinder

Clearance volume varies somewhat with different ratios of stroke to cylinder diameter; the actual condition or state of vapor at the end of the suction stroke and beginning of the compression is impossible to determine accurately in tests. This precludes giving very accurate data. Table 20 gives test results obtained by Reed and Ambrosius in 1931 at the Univ. of Ill. on a single-acting ammonia compressor with a clearance volume of 5.56% of the piston displacement volume.

SIZE OF COMPRESSOR CYLINDER.—To compensate for reduced compressor capacity, due to re-expansion of vapor on the suction stroke, and an increase in volume of saturated vapor, due to superheating while passing through hot suction ports and passages, volume G of saturated gas, as previously calculated, is taken as $G/0.75$ to $G/0.80$, the result being compressor displacement D, cu. ft. required per min. per ton of refrigeration per 24 hr. For purposes of calculation, assume $D = G/0.77$. Let N = number of working strokes per min. = 2 × r.p.m. for 1 single-, double-acting compressor or 2 single-

Table 20.—Volumetric Efficiency of Single-acting Ammonia Compressor

Suction Pressure, lb. per sq. in., abs.	Discharge Pressure, lb. per sq. in., abs.			
	100	150	200	250
	Volumetric Efficiency (E)			
10	0.826	0.780	0.735	0.677
20	.882	.840	.790	.743
30	.925	.875	.823	.776
40	.970	.910	.865	.817
50	.990	.945	.892	.848

acting compressors; l = length of stroke, in.; d = diameter of compressor cylinder, in.; M.E.P. = mean effective pressure in compressor cylinder, lb. per sq. in.

Approximate piston displacement, cu. in. per stroke $= {}^1/_4\, \pi d^2 l = 1728D/N$.

$$\text{M.E.P.} = p_s\{n/(n-1)\}\{(p_c/p_s)^{(n-1)/n} - 1\}, \quad . \quad . \quad . \quad . \quad . \quad [17]$$

where n = 1.31 for NH_3; = 1.25 for SO_2; = 1.31 for CO_2.

$$\text{I. Hp.} = \text{M.E.P.} \times M \times v_s/33{,}000. \quad . \quad . \quad . \quad . \quad . \quad . \quad [18]$$

Table 21 gives approximate mean effective pressures for various suction and condenser pressures. The brake Hp. of the compressor will be 1.5 × theoretical I.Hp. (approx.) for large compressors, and 2 × theoretical Hp. for small ones.

The heat-content (enthalpy)-entropy diagram generally is used to determine heat equivalent of work of compression per lb., $(h_c - h_s)$ B.t.u. See Figs. 4–8. Theoretical Hp. per ton of refrigeration $= 778\,(h - h_s)M/33{,}000 = (h_c - h_s)M/42.5$........ [19]

Example.—Required the size of NH_3 compressor cylinders for a 2-cylinder, single-acting machine and compressor, and also the I.Hp. to produce 40 tons of refrigeration per 24 hr., operating dry compression. Condensing pressure, 181.1 lb. per sq. in., gage, corresponding to a temperature of 95° F. for liquid entering the evaporating coil; suction pressure, 15.7 lb. per sq. in., gage, corresponding to a temperature of 0° F. for saturated gas leaving the evaporator coils. No subcooling of liquid, or superheat in suction vapor. Assumed volumetric efficiency of compressor = 80%.

Solution.—Refrigerating effect of 1 lb. of $NH_3 = R_1 = (h_s - q_c)$ = (611.8 − 149.4) = 462.4 B.t.u. per lb. NH_3 to be circulated per min. per ton of refrigeration = 200/462.4 = 0.43 lb.

Compressor Displacement Required.—Specific volume of NH_3 leaving the evaporator coils = 9.12 cu. ft. Volume of saturated vapor to be pumped per min. per ton of refrigeration per 24 hr. = 0.43 × 9.12 = 3.93 cu. ft. Piston displacement required per ton = 3.93/0.80 = 4.9 cu. ft. = 196 cu. ft. per min. for 40 tons. With an assumed stroke of 18 in. and a speed of 76 r.p.m., the area of each compression cylinder will be (196 × 144) ÷ (2 × 76 × 1.5) = 123.8 sq. in. or 12.55 in. diam. From Table 23, the nearest standard single-acting compressor is 12 1/2 × 18 in.

Compressor I.Hp.—The work to be performed per lb. of $NH_3 = (h - h_s)$ or, from Fig. 4, (733 − 611). Expected I.Hp. will be approximately 20% more than theoretical. Theoretical I.Hp. per ton = {0.43 (733 − 611)} ÷ 42.5 = 1.23. The expected I.Hp. per ton is 1.22 × 1.2 = 1.48. Expected I.Hp. of compressor = 40 × 1.23 × 1.20 = 59.0. Combined mechanical efficiency of engine and compressor averaging about 85%, the engine I.Hp. required is 58.6/0.85 = 69.4 Hp. Compare the piston displacement and the compressor I.Hp. with the data of Table 25.

BACK PRESSURES.—Table 22 gives the average temperature and corresponding back pressures of the expanding ammonia in the evaporating coils to maintain given room temperatures or brine tank temperatures. It is assumed that the room or tank has sufficient coil surface.

COMPARISON OF SINGLE- AND DOUBLE-ACTING AMMONIA MACHINES OPERATING DRY COMPRESSION.—Table 25 compares results obtained in tests by the York Mfg. Co. of single- and double-acting machines with a compressor cylinder 12 1/2 × 18 in. The better results (I.Hp. per ton of refrigeration) obtained with a single-acting machine were ascribed to the different design of suction valve in the two machines. In the double-acting machine it was small and had a more contracted area as compared with the large valve in the piston of the single-acting machine, giving an increased superheating effect of the gas during the suction stroke of the double-acting machine. The

Table 21.—Indicated Mean Effective Gage Pressures in Ammonia Compressors

(De La Vergne Machine Co., New York)

			Condenser								
	P_s	P_c =	103	115	127	139	153	168	184	200	218
		T_c	65°	70°	75°	80°	85°	90°	95°	100°	105°
Evaporator	4	− 20° F.	41.46	43.91	46.34	48.77	51.23	53.68	56.11	58.54	60.99
	6	− 15°	42.72	45.38	47.90	50.74	53.40	56.08	58.86	61.40	64.08
	9	− 10°	44.40	47.38	50.33	53.29	56.25	59.20	62.16	65.14	68.09
	13	− 5°	45.86	49.15	52.42	55.70	58.97	62.25	65.53	68.81	72.08
	16	0°	46.94	50.56	54.16	57.78	61.40	65.00	68.62	72.22	75.84
	20	5°	47.74	51.73	55.70	59.68	63.67	67.66	71.62	75.61	79.81
	24	10°	48.04	52.40	56.77	61.13	65.51	69.86	74.24	78.59	82.97
	28	15°	47.88	52.67	57.44	62.23	67.02	71.81	76.60	81.39	86.18
	33	20°	47.08	52.30	57.53	62.75	67.98	73.23	78.46	83.68	88.91
	39	25°	45.06	51.34	57.05	62.75	68.46	74.17	79.88	85.58	91.29
	45	30°	43.16	49.71	55.92	62.14	68.35	74.56	80.77	86.98	93.19
	51	35°	40.52	47.26	54.02	60.76	67.52	74.28	81.02	87.78	94.52

Table 22.—Pressures and Temperatures in Ammonia Evaporating Coils (F. E. Matthews)

Room temperature, deg. F.	5	10	15	20	28	32	36	40	50	60
Back pressure, lb. per sq. in., gage..	6.9	8.6	11.8	15.3	21.6	25.1	27	30	35.2	40
Temperature ammonia, deg. F.	− 13	− 10	− 5	0	8	12	14	17	22	26

Table 23.—Sizes and Capacities of Vertical Single-acting Ammonia Compressors
(York Mfg. Co., York, Pa.)

Compressor Bore, in.	Compressor Stroke, in.	Engine Bore, in.	Engine Stroke, in.	Tons Refrigeration	Revolutions per Minute	Horsepower of Engine
7 1/2	10	11 1/2	10	10	95	18
9	12	13 1/2	12	20	110	35
11 1/2	15	16	15	35	94	60
12 1/2	18	18	18	40	76	69
14	21	20	21	65	84	111
16	24	24	24	90	78	154
18	28	26	28	125	74	214
21	32	28 1/2	32	175	66	300
24	36	34	36	250	64	427
27	42	36	42	350	61	598
30	48	44	48	500	62	855

Table 24.—Capacities of Two-cylinder, Single-acting Ammonia Compressors

Size	Condensing Pressure, lb. per sq. in., gage	Suction Temperature (Evaporator), deg. F. −8.4 Tons Refrig.	−8.4 B.Hp.	0 Tons Refrig.	0 B.Hp.	21.4 Tons Refrig.	21.4 B.Hp.	35 Tons Refrig.	35 B.Hp.
3 × 3 in.	165	1.42	3.22	1.79	3.41	3.03	3.76	3.98	4.0
400 r.p.m.	185	1.37	3.30	1.70	3.46	2.94	3.91	3.86	4.1
4 × 4 in.	165	3.19	6.89	4.01	7.31	6.81	8.10	8.40	8.22
375 r.p.m.	185	3.09	7.10	3.88	7.54	6.61	8.50	8.69	8.86
5 × 5 in.	165	6.06	12.90	7.60	13.80	12.90	15.40	17.00	16.00
360 r.p.m.	185	5.86	13.40	7.40	14.30	12.60	16.30	16.40	17.00
6 × 6 in.	165	10.60	20.90	13.30	22.40	22.60	25.20	29.50	26.50
360 r.p.m.	185	10.20	21.50	12.90	23.30	21.90	26.60	28.60	27.90
7 × 7 in.	165	17.10	32.50	21.20	37.30	37.20	40.10	49.50	43.10
360 r.p.m.	185	16.70	34.20	20.60	38.90	36.40	43.00	48.30	44.70
8 × 8 in.	165	26.30	48.40	32.90	52.00	56.00	58.40	73.20	62.00
360 r.p.m.	185	25.40	50.20	31.90	54.00	54.40	62.00	68.50	64.40
9 × 9 in.	165	32.20	56.40	40.30	60.50	68.60	68.20	90.00	72.30
300 r.p.m.	185	31.10	58.60	39.00	63.10	66.60	72.60	86.60	73.20
10 × 10 in.	165	45.50	76.00	57.10	82.00	97.00	92.50	127.00	98.50
300 r.p.m.	185	44.00	79.50	55.20	86.00	94.30	98.50	122.60	103.00

V-belt drives generally are used for small motor-driven compressors.

Table 25.—Displacement and Horsepower per Ton of Refrigeration of Single-acting (S.A.) and Double-acting (D.A.) Ammonia Compressors, Dry Compression
(York Mfg. Co., York, Pa.)

Condenser Gage Pressure and Corresponding Temperature. Temperature of Liquid at Expansion Valve: Lb. per sq. in.	Deg. F.	Suction Gage Pressure and Corresponding Temperature: 5 lb. = −17.5° F. Volumetric Efficiency, % of Displacement	5 lb. Cu. in. Displacement per min. per Ton of Refrigeration	5 lb. I.Hp. per Ton (Compressor)	10 lb. = −8.5° F. Volumetric Efficiency	10 lb. Cu. in. Displacement	10 lb. I.Hp. per Ton (Compressor)	15.67 lb. = 0° F. Volumetric Efficiency	15.67 lb. Cu. in. Displacement	15.67 lb. I.Hp. per Ton (Compressor)	20 lb. = 5.7° F. Volumetric Efficiency	20 lb. Cu. in. Displacement	20 lb. I.Hp. per Ton (Compressor)	25 lb. = 11.5° F. Volumetric Efficiency	25 lb. Cu. in. Displacement	25 lb. I.Hp. per Ton (Compressor)
S.A. 145	82	79.	12,608	1.65	81.2	9,811	1.4	83.	7829	1.20	84.2	6765	1.07	85.5	5836	0.943
D.A. 145	82	68.	14,465	1.93	80.5	11,300	1.61	73.	8901	1.36	74.7	7625	1.2	76.5	6522	1.054
S.A. 165	89	77.5	13,045	1.83	79.7	10,148	1.56	81.5	8092	1.34	82.7	6990	1.20	84.	6027	1.071
D.A. 165	89	66.5	15,203	2.14	69.	11,720	1.80	71.5	9224	1.53	73.2	7898	1.36	75.	6751	1.2
S.A. 185	95.5	76.	13,491	2.01	78.2	10,487	1.72	80.	8362	1.49	81.2	7219	1.34	82.5	6223	1.197
D.A. 185	95.5	65.	15,774	2.35	67.5	12,150	1.99	70.	9555	1.7	71.7	8176	1.51	73.5	6985	1.344
S.A. 205	101.4	74.5	13,947	2.19	76.7	10,834	1.88	78.5	8630	1.63	79.2	7450	1.47	81.	6420	1.323
D.A. 205	101.4	63.5	16,362	2.57	66.	12,590	2.18	68.5	9890	1.87	70.7	8459	1.67	72.	7222	1.488

NOTE.—The above efficiencies and displacements apply when the piston clearance does not exceed 1/32 in. Unless clearance is excessive no addition to the horsepower will be necessary. Where liquid is cooled lower than temperature corresponding to condensing pressure, there will be a reduction in horsepower and displacement proportional to the increase of work done by each pound of liquid handled.

initial volume to be compressed thus was increased, resulting in increased power consumption. For engine horsepower add 17% to the compressor horsepower up to 20 tons capacity, and 15% for larger machines.

WET COMPRESSION.—In the wet compression system, sufficient liquid is by-passed from the liquid line to the suction line of the compressor, so that as the end of compression the vapor will be dry and saturated ($x = 1$). Let x_2 = vaporized portion of 1 lb. of the mixture at the beginning of compression, r_s = latent heat at temperature t_s. Then

$$W = 778 \times \{H_c - (q_s + x_2 r_s)\}, \quad \ldots \quad [20]$$

where W = ft.-lb. of work per lb. of medium circulated. The I.Hp. per ton of refrigeration per 24 hr. = $MW/33{,}000$. Entropy being constant for an adiabatic change, the value of x_2 may be obtained from the entropy tables for the vapor under consideration. Weight per pound of vapor circulated that reaches evaporator coils and produces useful refrigeration is $(1 - x_2)$. Let n_s = entropy of the liquid corresponding to t_s; R_s/T_s = entropy of vaporization corresponding to t_s; N_c = entropy of the vapor corresponding to t_c. Then

$$n_s + (x_2 r_s/T_s) = N_c, \quad \text{and} \quad x_2 = (T_s/r_s)(N_c - n_s) \quad \ldots \quad [21]$$

Table 26 compares the results of tests made by the York Mfg. Co. obtained when operating with wet and dry compression. The tests were made on a double-acting machine at a compression pressure of 185 lb. per sq. in., gage, and 15.67 lb. per sq. in., gage, suction pressure. The figures in the table are the average results of 6 wet and 6 dry compression runs of 6 hr. duration each, wet and dry runs alternating.

Allowable Velocities in Vapor and Liquid Piping.—Table 27 gives suitable velocities in piping for liquid and vapor refrigerating media.

Steam-jet-Vacuum Refrigeration

The refrigerating effect in steam-jet-vacuum refrigeration is produced by evaporating part of the liquid circulated, in this case water, in a partial vacuum. Final temperature of the water corresponds to temperature of evaporation for the partial vacuum. See Fig. 14. The relatively large volumes of water vapor to be removed and compressed, for refrigerating effect, have made reciprocating vapor compressors in this connection impracticable. Water is sprayed into the evaporator to present a surface from which evaporation may occur in a reasonable space. The unit is compact, simple and has few parts; it is used to cool water for industrial purposes, and with air conditioning apparatus when steam is available. Usual final temperature t_e of the chilled water is 40° to 60° F. Considerably more condensing water is required than for electrically-driven NH_3, CO_2,

Table 26.—Comparison of Results of Wet and Dry Ammonia Compressors

	Wet Compression	Dry Compression
Tonnage refrigeration (brine cooling).....	20.94	20.47
Total I.Hp. of compressor..............	42.88	32.83
Total I.Hp. of engine..................	49.51	36.16
Friction in percent of engine horsepower..	13.39	9.19
Compressor I.Hp. per ton of refrigeration	2.066	1.603
Engine I.Hp. per ton of refrigeration.....	2.368	1.766

Table 27.—Velocities, ft. per min., of Refrigerating Media (R. C. Doremus)

Fluid	Suction Side	Discharge Side	Fluid	Suction Side	Discharge Side
Ammonia..............	4000–5000	5000–6000	Sulphur dioxide.......	4000–5000*	5000–6000*
Aqua ammonia........	30–50	100–250	Sulphur dioxide.......	1000–2000 †	2000–2500 †
Carbon dioxide........	1000–1200	1000–1200	Brine................	60–150	100–200
Methyl chloride........	3000–4000*	4000–5000*	Water................	60–200	100–250
Methyl chloride........	1000–2000 †	2000–2500 †			

* In large pipe sizes (manufacturing plants). † In small tubing (electric refrigerators).

Table 28.—Approximate Pressure Loss, lb. per sq. in., in Ammonia Vapor Piping*

Pipe Size, in.	Velocity, ft. per min.		Pipe Size, in.	Velocity, ft. per min.	
	4000	6000		4000	6000
3/4	7.5	16.7	3	1.95	4.2
1 1/4	4.3	9.5	4	1.4	3.2
2	2.9	6.2	6	0.95	2.1

* Allowances for fittings and valves to be added to measured length of piping: 90° ell, 40 ft.; tee, 60 ft.; globe valve, 60 ft.

SO_2, etc. machines, as the evaporated vapor producing refrigerating effect must be condensed along with steam supplied to the jet.

Fig. 15 is a diagram of the apparatus. Steam expands in a single or multiple set of steam nozzles to a pressure corresponding to temperature t_e maintained in the evaporator. The velocity, ft. per sec., of steam leaving nozzle exit, for adiabatic expansion is $w = 223.7\sqrt{h_1 - h_e}$, where h_1 and h_e = heat content of steam supplied to nozzle, and after expansion to evaporator pressure, respectively. For an initial pressure p_1 = 125 lb. per sq. in., abs., w = 4000 ft. per sec. (approx.) for usual chilled water temperatures.

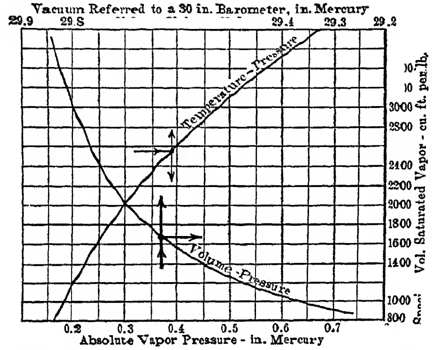


FIG. 14. Pressure-Temperature-Volume Curve of Water Vapor

The high velocity steam mixes in the injector chamber with saturated vapor from the evaporator, and passes through the converging portion of the machine. Mixture is compressed adiabatically to pressure corresponding to the partial vacuum in condenser.

Part of the kinetic energy of the nozzle steam S lb. is used to accelerate the evaporated vapor W lb. to the velocity at which the mixture enters the compressor section. Part of the kinetic energy of mixture $(S + W)$ is used to compress the mixture from evaporator pressure p_e to condenser pressure p_c. Condenser temperature t_c is approximately 5 to 10°F. higher than initial temperature of condensing water. The usual partial vacuum for the condenser is 27.5 to 28.5 in. Hg. The vapor mixture enters the diverging portion of the unit, where it is compressed and its velocity reduced to approximately 500 ft. per sec., at which velocity it passes into the condenser. The compression is from p_e to p_c.

Let M = water to be cooled, lb. per hr.; t, t_e = initial and final temperature of water to be cooled, deg. F.; 12,000 = heat to be extracted from water per hr. per ton of refrigeration, B.t.u.; W = lb. of water to be evaporated at evaporator temperature t_e, deg. F. per hr. per ton of refrigeration. Then $M\,(t - t_e)/12{,}000$ = tons of refrigeration required.

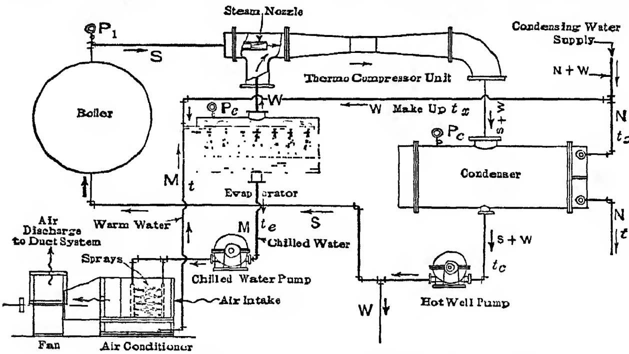


FIG. 15. Diagram of Steam-jet Refrigerating Apparatus

WEIGHT OF WATER, W, lb., to be evaporated per hr. per ton of refrigeration is, $W = 12{,}000/(h_e - q_x)$, (approx. 12 lb.). It may be assumed for approximation that the quality of the vapor leaving the evaporator is 90% at pressure p_e. h_e = total heat of vapor leaving evaporator, B.t.u. per lb.; q_x = heat of liquid make-up water, B.t.u. per lb.

WEIGHT OF STEAM S, lb. per hr. per ton of refrigeration, to be supplied unit may be approximated as follows, using the total-heat-entropy diagram or chart, page 5–19: Let h_1 = total heat of steam supplied to nozzles, B.t.u. per lb.; h_e = total heat of steam

expanded adiabatically to pressure p_e, B.t.u. per lb.; E = combined efficiency of expanding nozzles, entrainment and compression (approximately 0.40).

Assume the vapor entering condenser to be dry and saturated, with total heat h_c at pressure p_e (actually it will contain some moisture). Compression is assumed to be adiabatic. From the chart determine the total heat h per lb. of the mixture $(S + W)$ at beginning of compression. The heat available for work of compression is $S\,E\,(h_1 - h_e)$, and the heat equivalent to the work of compression is $(S + W)\,(h_c - h)$ B.t.u. per hr. per ton of refrigeration. Then

$$0.42\,S\,(h_1 - h_e) = (S + 12)\,(h_c - h) \quad . \; . \; . \; . \; . \; . \; . \; [22]$$

from which the value of S is obtained. The steam used by the steam-jet condenser air pumps is not included in the value of S.

Example.—Assume chilled water temperature required to be 40° F.; condenser vacuum, 28 in.; initial steam pressure 125 lb., gage, dry and saturated.

Referring to heat-content-entropy chart, $h_1 = 1193$; $h_e = 792$; $h_c = 1105$; $h = 990$.

$0.42\,S \times 401 = (S + 12)\,115$. $\therefore S = 30.6$ lb. of steam per hr. per ton of refrigeration.

The overall thermal efficiency of the complete system sometimes is stated as the ratio of heat extracted from the evaporator to heat input to boiler, including the steam used by steam-jet condenser air pumps, as well as the heat equivalent of the work of boiler-feed pump. A detailed analysis and discussion of the steam ejector cycle by P. Kalusian, will be found in *Trans.* A.S.R.E., 1934.

Fig. 16 gives results obtained with a steam-jet refrigerating unit for various operating conditions with an initial steam pressure of 125 lb., gage, dry and saturated. The condensing water REQUIRED is approximately

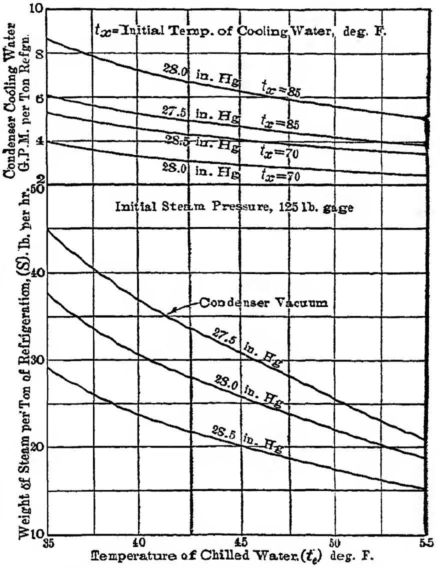


Fig. 16. Performance Curve, Steam-jet Refrigerating Apparatus

$$N = (S + W)\,r_c/(t_y - t_x) \quad . \; . \; . \; . \; . \; . \; . \; . \; . \; [23]$$

where N = lb. per ton per hr.; t_y may be assumed 5 to 10° F. below t_c; t_x = initial temperature of condensing water, deg. F.; r_c = latent heat corresponding to t_c, B.t.u. per lb. When a cooling tower is used assume t_x = 80° F. for preliminary estimates.

A number of American manufacturers build steam jet refrigerating machines; the majority use multiple nozzles in the ejector chamber. A surface condenser ordinarily is used, although barometric condensers may be employed, either type equipped with a two-stage steam-jet air eliminator having inter- and after-condensers. Maximum efficiency is obtained only when the machine operates at the designed rate of flow of steam nozzles and other portions of the apparatus. It is customary to provide two or more separate steam-jet units to obtain the best overall efficiency under variable loads. The units all are connected to the same evaporator and condenser.

5. THE AMMONIA ABSORPTION MACHINE

Fig. 17 is a diagram of the ammonia absorption machine. Superheated ammonia, gas and steam are driven off from a solution of ammonia and water in the generator, heated by steam coils. The total pressure existing in the generator consists of the partial vapor pressures of ammonia, gas and water. Vapors leaving the generator are condensed in the rectifier (dehydrator) in which the temperature is maintained at a sufficiently low point to condense practically all water vapor but not the ammonia. The condensed water re-absorbs a portion of the ammonia and is returned to the generator as rich liquor. The practically dry ammonia gas then is liquefied in the ammonia condenser and ex-

Table 29.—Boiling Points of Solutions of Ammonia and Water

(Figures are temperatures, deg F., corresponding to concentration of NH_3 solution)

Percent NH_3	Absolute Pressure								
	148	150	152	154	156	158	160	162	164
10	306.3	307.4	308.4	309.4	310.5	311.6	312.6	313.6	314.7
12	295.8	297.0	298.0	299.0	300.2	301.2	302.2	303.2	304.1
14	286.1	287.3	288.3	289.3	290.4	291.4	292.4	293.3	294.2
16	276.5	277.5	278.6	279.6	280.7	281.7	282.8	283.7	284.6
18	266.9	268.0	269.0	270.0	271.0	272.0	273.0	273.8	274.8
20	257.6	258.6	259.7	260.7	261.7	262.6	263.6	264.5	265.5
22	248.9	250.0	250.9	251.9	252.9	253.9	254.9	255.8	256.6
24	240.8	241.9	242.8	243.7	244.8	245.7	246.7	247.6	248.5
26	232.8	233.7	234.7	235.6	236.6	237.4	238.4	239.3	240.3
28	224.5	225.5	226.5	227.4	228.4	229.3	230.3	231.2	232.0
30	216.3	217.3	218.3	219.2	220.2	221.1	222.0	223.0	224.0
32	208.5	209.4	210.3	211.1	212.1	213.1	213.8	214.6	215.4
34	200.8	201.7	202.6	203.5	204.4	205.3	206.1	206.9	207.7
36	193.6	194.5	195.5	196.4	197.4	198.3	199.1	199.9	200.6
38	186.3	187.3	188.1	189.1	190.0	190.8	191.7	192.6	193.4
40	179.1	180.1	181.0	182.0	182.9	183.8	184.6	185.5	186.4
42	172.4	173.3	174.2	175.1	175.9	176.9	177.7	178.4	179.2
44	165.4	166.3	167.2	168.0	168.7	169.8	170.8	171.4	172.2
46	158.6	159.4	160.3	161.3	162.1	162.0	163.8	164.5	165.3
48	152.4	153.3	154.1	155.0	155.8	156.7	157.5	158.3	159.0
50	145.5	146.4	147.3	148.0	149.0	149.8	150.7	151.5	152.3

Percent NH_3	Absolute Pressure								
	166	168	170	172	174	176	178	180	182
10	315.6	316.6	317.6	318.7	319.7	320.8	321.7	322.7	323.6
12	305.1	306.1	307.1	308.1	309.1	310.1	311.0	312.0	313.0
14	295.2	296.2	297.1	298.1	299.0	300.0	301.0	302.0	302.9
16	285.5	286.4	287.3	288.3	289.2	290.1	291.1	292.0	293.0
18	275.7	276.7	277.6	278.5	279.5	280.5	281.4	282.4	283.3
20	266.5	267.5	268.4	269.3	270.3	271.3	272.3	273.3	274.1
22	257.5	258.4	259.3	260.2	261.1	262.1	263.0	263.9	264.7
24	249.4	250.3	251.0	252.0	253.0	253.8	254.6	255.5	256.5
26	241.1	242.0	243.0	243.8	244.6	245.5	246.3	247.2	248.0
28	232.9	233.7	234.5	235.4	236.3	237.1	238.0	238.9	239.8
30	224.9	225.8	226.7	227.5	228.4	229.3	230.1	231.1	231.8
32	216.3	217.0	217.7	218.6	219.5	220.4	221.2	222.0	223.0
34	208.4	209.2	210.0	210.9	211.7	212.5	213.4	214.2	215.0
36	201.4	202.2	202.9	203.7	204.4	205.2	206.0	206.8	207.5
38	194.1	194.9	195.6	196.3	197.0	197.8	198.6	199.3	200.0
40	187.2	188.0	188.9	189.8	190.5	191.2	192.0	192.8	193.6
42	180.0	180.6	181.4	182.2	183.0	183.7	184.5	185.3	186.0
44	173.9	173.6	174.3	175.0	175.8	176.8	177.3	178.2	178.9
46	166.0	166.8	167.5	168.3	169.0	169.7	170.5	171.3	172.0
48	159.1	160.5	161.2	161.9	162.7	163.3	164.1	164.9	165.8
50	153.8	154.1	154.9	155.5	156.3	157.2	158.1	158.7	159.5

Percent NH_3	Absolute Pressure								
	184	186	188	190	192	194	196	198	200
10	324.5	325.5	326.5	327.0	328.2	329.0	329.9	330.7	331.6
12	313.9	314.7	315.7	316.5	317.3	318.3	318.1	320.0	320.7
14	303.7	304.6	305.6	306.5	307.4	308.4	309.3	310.3	311.1
16	293.9	294.9	295.7	296.6	297.6	298.5	299.5	300.5	301.3
18	284.2	285.1	286.0	287.0	287.9	288.9	289.9	290.8	291.7
20	275.0	276.0	277.0	278.0	278.9	279.9	280.8	281.7	282.7
22	265.6	266.5	267.5	268.5	269.3	270.3	271.2	272.1	273.0
24	257.4	258.2	259.1	260.0	260.9	261.8	262.7	263.6	264.5
26	248.9	249.7	250.6	251.5	252.3	253.0	253.9	254.8	255.8
28	240.6	241.5	242.4	243.2	244.0	244.9	245.8	246.7	247.5
30	232.6	233.5	234.4	235.1	236.0	236.8	237.7	238.7	239.3
32	223.8	224.7	225.5	226.4	227.3	228.2	229.1	230.0	230.8
34	215.9	216.7	217.5	218.3	219.2	220.0	220.8	221.7	222.5
36	208.3	209.0	209.8	210.6	211.3	212.0	212.9	213.7	214.4
38	200.7	201.6	202.3	203.0	203.8	204.5	205.2	206.0	206.8
40	194.3	195.0	195.7	196.5	197.3	198.1	198.7	199.4	200.1
42	186.9	187.6	188.3	189.2	190.0	190.7	191.5	192.3	193.0
44	179.7	180.5	181.3	182.0	182.8	183.5	184.3	185.1	185.0
46	172.8	173.6	174.3	175.2	175.9	176.7	177.4	178.3	179.9
48	166.4	167.1	167.9	168.6	169.3	170.1	170.9	171.7	172.5
50	160.3	161.0	161.8	162.4	163.2	163.9	164.7	165.3	166.0

panded in evaporating coils, producing a refrigerating effect as in the compression system. The expanded ammonia gas from the evaporating coil is re-absorbed in the absorber by weak liquor drawn from the bottom of the generator. The rich liquor produced by absorption is returned to the generator, flowing over a series of trays in the analyzer where it reduces the superheat in the gases given off by the generator. This reduces the amount of heat to be removed in the rectifier and condenser. The analyzer sometimes is omitted to reduce first cost of the apparatus, but this saving is effected at the expense of operating economy. Pressure existing in generator and rectifier depends on temperature maintained in the condenser, which, in turn, is governed by quantity and temperature of cooling water available.

Calculations for the design of an absorption machine involve: 1. Dalton's law of partial vapor pressure, modified. 2. Properties of saturated and superheated steam and saturated and superheated ammonia vapor. 3. Boiling point of ammonia solutions of various strengths corresponding to various pressures. 4. Weight of ammonia absorbed by water. 5. Heat developed by chemical reaction occurring when ammonia is absorbed by water.

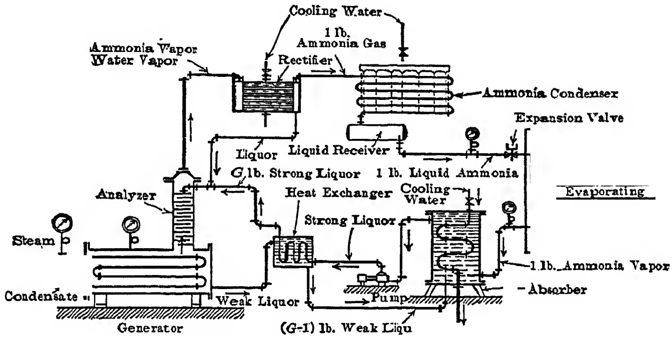


Fig. 17. Arrangement of Ammonia Absorption Plant

BOILING POINT OF AMMONIA SOLUTIONS.—Table 29 gives boiling points at various pressures of solutions of water and ammonia. It is condensed from a table calculated by H. J. MacIntire, based on the experiments of Mollier, using a formula $T_s = T_a/(0.00466x + 0.656)$, where T_α = absolute temperature of saturated ammonia corresponding to the observed pressure; T_s = absolute temperature of the solution; x = percent of ammonia in solution, by weight. Between concentrations of 15% and 35% this formula checks with the experimental work of both Starr and Mollier.

HEAT OF ABSORPTION.—Let x = mean concentration or weight of ammonia in 1 lb. of solution; x_1 = weight of ammonia in 1 lb. of weak solution; x_2 = weight of am-

Table 30.—Heat of Absorption h (B.t.u.) and Weight G (lb.) of Strong Liquor Circulated per Pound of Ammonia

Concentration of Weak Solution		Concentration of Strong Solution								
		0.20	0.22	0.24	0.26	0.28	0.30	0.32	0.35	0.40
0.10	h =	825	821	816	811	806	801	796	788	774
	G =	9.0	7.5	6.43	5.63	5.0	4.5	4.09	3.6	3.0
0.12	h =	821	816	811	806	801	796	791	783	769
	G =	11.0	8.8	7.33	6.29	5.5	4.9	4.4	3.83	3.14
0.14	h =	816	811	806	801	796	791	786	777	763
	G =	14.3	10.75	8.6	7.17	6.14	5.19	4.8	4.1	3.31
0.15	h =	814	809	804	799	793	788	783	774	761
	G =	17.0	12.14	9.44	7.73	6.54	5.67	5.0	4.25	3.4
0.16	h =	811	806	801	796	791	786	780	772	758
	G =	21.0	14.0	10.5	8.4	7.0	6.0	5.25	4.32	3.5
0.18	h =	806	801	796	791	786	780	775	766	752
	G =	....	20.5	13.67	10.25	8.2	6.83	5.7	4.82	3.73
0.20	h =	....	796	791	786	780	775	769	761	746
	G =	....	40.0	20.0	13.3	10.0	8.0	6.7	5.3	4.0

monia in 1 lb. of strong solution; n = pounds of water to 1 lb. of ammonia in solution; m = pounds of ammonia gas added to the solution; then

$$x = \tfrac{1}{2}(x_1 + x_2);\ x_1 = 1/(1+n);\ x_2 = (1+m)/(1+n+m).$$

The heat h, in B.t.u., developed when 1 lb. of ammonia gas is added to a solution of ammonia of concentration x_1 may be determined by Mollier formula $h = 887 - 350x - 400x^2$. See Table 30. The formula is good for values of x up to about 0.60 at which value and over h is constant at 540 B.t.u. The heat in B.t.u. developed when 1 lb. of ammonia is added to water so that the final concentration is x, is $h = 893x - \{142.5x^2/(1-x)\}$.

AMOUNT OF LIQUOR TO BE CIRCULATED.—For each pound of anhydrous ammonia passing through the condenser and evaporating coils, G lb. of strong solution must be circulated, and $(G - 1)$ lb. of weak liquor enters the absorber. Weight G depends on the degree of concentration of the strong and weak solution. Using the notation of the preceding paragraph $G = (1 - x_1)/(x_2 - x_1)$. Table 30 gives values of G for various concentrations of weak and strong solutions.

RELATIVE WEIGHTS OF SUPERHEATED AMMONIA AND WATER VAPOR.—Dalton's law for mixtures of perfect gases is only approximately true for mixtures of superheated ammonia and water vapor. *Jour.* Am. Chem. Soc., vol. 83, shows that for low pressures and low percentages of ammonia solutions, the partial water-vapor pressure is directly proportional to the number of molecules in solution. Assuming this to be true for higher pressures, vapor pressure existing in the analyzer may be approximated by Spangler's formula

$$p_3 = p_1\{(1700 - 17x)/(1700 + x)\};\ p_2 = p - p_3 \quad . \quad . \quad . \quad . \quad . \quad [24]$$

where p = total pressure in generator, rectifier and condenser (gage pressure + 14.7), being fixed by the ammonia condenser temperature and pressure; p_1 = pressure of saturated water-vapor corresponding to its temperature; p_2 = pressure of ammonia gas; p_3 = partial water-vapor pressure; all pressures are in lb. per sq. in., absolute; x = percent of ammonia by weight in 1 lb. of solution. One cu. ft. of mixture leaving the analyzer and entering the rectifier consists of 1 cu. ft. of superheated ammonia gas and 1 cu. ft. of steam. If d_2 = lb. per cu. ft. of ammonia gas at pressure p_2, and d_3 = lb. per cu. ft. of steam at pressure p_3, each cu. ft. of vapor contains d_2 lb. of ammonia gas and d_3 lb. of water vapor. For each pound of ammonia gas entering rectifier there will be d_3/d_2 lb. of water vapor. If v_2 = specific volume of 1 cu. ft. of ammonia gas at pressure p_2, and v_3 = specific volume of 1 cu. ft. of vapor at pressure p_3, $d_2 = 1/v_2$ and $d_3 = 1/v_3$.

HEAT SUPPLIED TO GENERATOR.—The amount of ammonia to be circulated per ton of refrigeration depends on the condensing pressure and temperature desired in the evaporating coil. Total heat taken into the system must equal that removed, radiation being neglected. Calculations can be made most conveniently on the basis of 1 lb. of active ammonia passing through the evaporating coil. Let h_g = B.t.u. supplied to generator per lb. of active ammonia passing through evaporating coils; h_r = B.t.u. to be removed from rectifier; h_c = B.t.u. to be removed by condenser; h_e = B.t.u. taken in by evaporator coils; h_a = B.t.u. removed from absorber; h_i = B.t.u. lost in heat exchanger. Then

$$h_g + h_e = h_a + h_c + h_r + h_i, \quad \text{and} \quad h_g = h_a + h_c + h_r - h_e + h_i.$$

The weight of saturated steam required in the generator coils per lb. of active ammonia is $W = (h_g/L) + 15\%$ to 20% for radiation losses. L is the latent heat of steam corresponding to pressure in the coil. Steam temperature in coil should not be less than boiling temperature of the weak solution. All quantities are per lb. of ammonia.

Heat to be supplied the generator may be approximated by the following method: Heat developed when 1 lb. ammonia liquid is absorbed in a solution of strength x_1, thereby raising the strength to x_2, is given by the Mollier formula $h = 345 - 345x - 400x^2$, where $x = (x_1 + x_2)/2$. This is practically the same as the previous equation for heat developed when ammonia gas is absorbed by a solution, if 542 be added. It is equivalent to assuming that 542 B.t.u. must be removed in every case to change gas to a liquid. Heat to be supplied to generator per lb. of NH_3 is sum of items $a + b + c$, below:

a. Heat required to change concentration of solution from x_1 to x_2, per lb. of NH_3 driven off; $h = 345 - 345\,x - 400\,x^2$.

b. Difference between heat content of 1 lb. superheated NH_3 gas (at pressure carried by generator and temperature of strong solution returned to generator) and heat content of 1 lb. liquid NH_3 at temperature of weak solution leaving generator.

c. Weight of weak solution circulated per lb. of NH_3, $(G - 1)$ × temperature difference between strong and weak solution entering and leaving generator × specific heat of solution (approx. 1.05).

The small amount of heat required to evaporate and superheat the water vapor driven off with the ammonia here is neglected. Temperature in generator must be sufficient to boil the weak solution.

HEAT TO BE REMOVED FROM THE RECTIFIER.—Temperature t_3 of gas leaving the rectifier need be only approximately 10° higher than condenser temperature t_c, to condense approximately 95% of the water vapor, leaving the ammonia gas practically dry and uncondensed. The condensing temperature of the water vapor corresponding to its pressure p_3 is higher than the condensing temperature of the ammonia corresponding to its pressure p_2. In practice it is sufficiently accurate to assume all water vapor to be condensed from the mixture of gas and vapor in the rectifier. As shown above, each pound of ammonia gas entering rectifier is mixed with d_3/d_2 lb. of water vapor. To determine the amount of ammonia which this amount of vapor will re-absorb, the maximum concentration x_m possible for the absolute temperature of the solution T_s, and the absolute temperature T_a corresponding to the partial pressure p_2 must be determined.

$$T_s = T_a \div (0.00466x_m + 0.656) \quad . \quad . \quad . \quad . \quad . \quad . \quad . \quad [25]$$

whence the value of x_m may be found. If the initial concentration of the water vapor be that of the strong solution $= x_2$, the weight of ammonia absorbed by d_3/d_2 lb. of water condensed is $w = (d_3/d_2) \times (x_m/x_2)$ per lb. of ammonia entering rectifier. Then for 1 lb. of active ammonia passed to condenser and evaporator coils, $w_2 = 1 + \{(d_3/d_2) \times (x_m/x_2)\}$ lb. of ammonia vapor, and $w_3 = (d_3/d_2)\ \{1 + (d_3/d_2) \times (x_m/x_2)\}$ lb. of water vapor entering rectifier.

Let t_x = temperature of vapors entering rectifier (approximately 30° to 40° lower than boiling temperature of the strong solution of concentration x_1); t_3 = temperature leaving rectifier (approximately 10° higher than the ammonia condensing temperature t_c corresponding to pressure p_3); C_{ps} = mean specific heat of water vapor = 0.44 (approx.); C_{pa} = mean specific heat of ammonia vapor for pressure p_3 and temperature range t_3 to t_x = 0.66 (approx.) for usual conditions of operations; L_3 = latent heat of water vapor for temperature t_3; x = mean concentration of final solution of ammonia gas absorbed by the water vapor condensed $= {}^1/_2\,(x_m + x_2)$; A = heat in B.t.u. removed from rectifier to lower temperature of the vapors from t_x to t_3; B = heat, B.t.u., removed from rectifier to condense the water vapor; h = heat, B.t.u., developed by the absorption of w lb. of ammonia in the water condensed; h_r = total heat, B.t.u., removed by cooling water. Then

$$A = (C_{pa}w_2 + C_{ps}\,w_3)\,(t_x - t_3);\ B = r_3\,w_3. \quad . \quad . \quad . \quad . \quad . \quad . \quad [26]$$

$$h = 893\,x - \{142.5\,x_2/(1 - x)\};\ h_r = A + B + h. \quad . \quad . \quad . \quad . \quad [27]$$

HEAT REMOVED BY CONDENSER.—If the water vapor has been removed by the rectifier, the heat, B.t.u., to be removed by the condenser is $h_c = r_c + C_{pa}\,(t_3 - t_c)$, where t_3 = temperature of superheated gas arriving at the condenser (approximately $t_c + 10$); t_c = temperature of saturated ammonia gas corresponding to condenser pressure; r_c = latent heat of condenser pressure, and C_{pa} = mean specific heat at constant pressure of superheated ammonia vapor for condensing pressure p_3. See Fig. 12.

HEAT TAKEN INTO THE SYSTEM BY THE EVAPORATING COILS is $h_e = (h_s - hq_c)$, as in the compression system, where h_s = heat content of saturated vapor corresponding to evaporator temperature t_s, and q_c = heat of the liquid at temperature t_c.

HEAT TO BE ABSTRACTED FROM THE ABSORBER AND THE WEAK LIQUOR COOLER.—The temperature of the strong liquor leaving absorber ordinarily should be below 125° F., and 110° F. often is specified. With low steam and suction pressures, the temperature must be still further reduced. The temperature limit of absorber is fixed by the pressure in the evaporating coil, which is approximately the pressure in the absorber. Maximum absolute temperature of solution is determined by

$$T_s = T_a/(0.00466x_2 + 0.656) \quad . \quad . \quad . \quad . \quad . \quad . \quad . \quad [28]$$

where T_a = absolute temperature of saturated ammonia gas in evaporating coil and x_2 = concentration of strong solution.

The cool strong liquor from the absorber at temperature t_a absorbs heat in the heat exchanger from the hot weak liquor and enters the analyzer at a temperature t_g approximately 35° less than that of the outgoing weak liquor. For approximate calculations, temperature of the weak solution may be assumed as $(t_a + 10)$, being cooled to absorber temperature in an auxiliary cooler.

The heat, B.t.u., developed by the addition of ammonia gas to a weak solution of ammonia is

$$h = 887 - 350x - 400x^2 \quad . \quad . \quad . \quad . \quad . \quad . \quad . \quad . \quad [29]$$

The total heat h_a to be abstracted from the weak liquor cooler and absorber by the cooling water per pound of active ammonia passing through evaporating coil, comprises: *a*. Heat developed by absorption = h_1. *b*. Difference between heat contained in the weak liquor at the temperature of the generator t_g, and temperature t_a carried in the absorber, $= (G - 1) \times (t_g - t_a)$, where G = lb. of strong solution circulated per lb. of anhydrous ammonia passing through evaporating coil. Specific heat of solution is taken as 1. *c*. The negative quantity of heat introduced by the ammonia in raising its tem-

perature from the evaporating temperature t_s, to the temperature of the absorber t_a, or $\{-C'_{pa}(t_a - t_s)\}$, where C'_{pa} = specific heat of ammonia gas for the pressure existing in the evaporator coils. Then

$$h_a = h_1 + (G - 1)(t_g - t_a) - C'_{pa}(t_a - t_s) \quad . \quad . \quad . \quad . \quad . \quad [30]$$

LIQUOR PUMP.—The power required for the pump depends on the quantity of liquor to be pumped, cu. ft. per min., and the effective pressure per sq. in. on the plunger. The effective pressure is the difference between the pressure existing in the generator and evaporator coils plus the friction pressure loss. A non-condensing, fly-wheel type pump with cut-off requires approximately 35 to 40 lb. of steam per I.Hp.-hr. A direct-acting pump requires 100 to 150 lb. per I.Hp.-hr.

NOTES ON THE DESIGN OF ABSORPTION MACHINERY.—The concentration of weak liquor in high-pressure plants may be assumed as approximately 20% ammonia, and of strong liquor as 28%, by weight, for ordinary conditions. Corresponding figures for low-pressure plants are 30% and 38% respectively. Temperature of steam in the generator coils should not be less than that required to boil the weak solution.

Temperature t_x of vapors entering the rectifier may be assumed as 30° to 40° lower than temperature t_g of the weak hot liquor. Temperature of ammonia gas leaving the rectifier may be assumed as approximately 5° to 10° higher than its condensing temperature corresponding to condensing pressure p_3 and temperature t_c. Temperature of the strong liquor leaving the absorber may be assumed as 110° to 130° F. A heat exchanger of sufficient surface will increase the temperature of the strong liquor approximately 73% of the number of degrees temperature drop of the weak liquor. Absorption plants are usually operated with exhaust steam at 5 to 10 lb. per sq. in. back pressure.

STEAM REQUIRED PER HOUR PER TON OF REFRIGERATION.—Table 31 gives test data based on various condensing gage pressures and evaporator pressures of 15.67 lb. gage, on actual steam consumption per ton of refrigeration. The steam consumptions given are without analyzer. If an analyzer is used, deduct 3 lb. per ton of refrigeration as an approximation. Actual difference depends on amount of aqua ammonia circulated.

PERFORMANCE OF ABSORPTION MACHINES.—From an elaborate review by Gardner T. Voorhees of the action of an absorption machine under certain stated conditions, showing the quantity of ammonia circulated per hour per ton of refrigeration, its temperature, etc., at the several stages of the operation, and its course through the several parts of the apparatus, the following condensed statement is obtained:

Generator.—30.9 lb. dry steam, 38 lb. gage pressure, condensed, evaporates 32.2% strong liquor to 22.3% weak liquor.

Exchanger.—3.01 lb. weak liquor at 264° F. cools to 111° F.

Absorber.—Adds 0.43 lb. vapor from the brine cooler, making 3.44 lb. strong liquor at 111° F. to go to the pump.

Exchanger.—3.44 lb. heated to 224° F.; some of it is now gas, and the rest liquor of a little less than 32% NH_3.

Analyzer.—Delivers strong liquor to the generator, while the vapor, 91% NH_3, 0.4982 lb., goes to the rectifier.

Rectifier.—Cools the gas to 110° F., separating water vapor as 0.0682 lb. drip liquor which returns through a trap to the generator.

Condenser.—0.43 lb. NH_3 gas at 110° F. cooled and condensed to liquid at 90° F. by 2 gal. of water per min. heated from 73° F. to 86° F.

Table 31.—Steam Consumption, Lb. per Ton of Refrigeration for Various Condensing Pressures

(Evaporator pressure = 15.67 lb. per sq. in., gage)

Steam Pressure, lb. per sq. in., gage	Condensing Pressure						
	125	135	145	155	165	175	185
1	34.3	35.5	....	....	....	....	....
3	34.1	35.3	36.7	....	....	....	....
5	33.9	35.1	36.4	37.8	39.0	....	....
10	33.3	34.5	35.9	37.2	38.4	40.1	41.3
15	32.8	34.0	35.3	36.7	37.9	39.6	40.7
20	32.2	33.4	34.8	36.1	37.3	39.0	40.2
25	31.7	32.9	34.2	35.6	36.8	38.5	39.6
30	31.1	32.3	33.7	35.0	36.2	37.9	39.0
35	30.6	31.8	33.1	34.5	35.7	37.3	38.5
40	30.0	31.2	32.5	33.9	35.1	36.8	37.9
45	29.5	30.7	32.0	33.4	34.6	36.2	37.4
50	29.0	30.1	31.4	32.8	34.0	35.7	36.8

Expansion Valve and Cooler.—Reduces liquid to 0° F. and boils it at 0° F., cooling 3 gal. of brine per min. from 12° F. to 3° F. Gas passes to absorber and cycle repeats.

Of the 2 gal. per min. of cooling water flowing from the condenser, 0.2 gal. goes to the rectifier, where it is heated to 142° F., and 1.8 gal. through the absorber, where it is heated to 110° F.

Heat Balance.—Absorbed in the generator, 496 B.t.u.; in the brine cooler, 200 B.t.u.; total, 696 B.t.u. Rejected: condenser, 220 B.t.u.; absorber, 383 B.t.u.; rectifier, 93 B.t.u.; total, 696 B.t.u.

Table 32 shows the strength of liquors and quantity of steam required per hour per ton of refrigeration under the conditions stated.

Table 33 gives steam consumption, lb. per hour per ton of refrigeration, for engine-driven compressors and for absorption machines with liquor pump not exhausting into the generator at suction and condenser pressures (gage) given.

The economy of the absorption machine is much better for all conditions than that of a simple non-condensing engine-driven compressor. At suction gage pressures above 8 to 10 lb. the economy of the compound condensing engine-driven compressor exceeds that of the absorption machine, the absorption machine giving the superior economy at suction pressures below 8 to 10 lb.

Relative Performance of Ammonia Compression and Absorption Machines, Assuming No Water to be Entrained with the Ammonia-gas in the Condenser. (Denton and Jacobus, *Trans.* A.S.M.E., xiii.)—See Table 34. It is assumed in the calculation for both machines that 1 lb. of coal imparts 10,000 B.t.u. to the boiler. The condensed steam from the generator of the absorption machine is assumed to be returned to the boiler at temperature of steam entering the generator. The engine of the compression machine is assumed to exhaust through a feedwater heater that heats the feedwater to 212° F. and to consume 26 1/4 lb. of water per Hp.-hr. Figures for the compression machine include the effect of friction, which is taken at 15% of the net work of compression.

For discussion of the efficiency of the absorption system, see Ledoux's work; paper by Prof. Linde, and discussion on the same by Dr. Jacobus, *Trans.* A.S.M.E., xiv, 1416, 1436; and papers by Denton and Jacobus, *Trans.* A.S.M.E., x, 792, xiii, 507.

SILICA-GEL SYSTEM.—The following was abstracted from A.S.R.E. Data Book: Silicon dioxide SiO_2, or silica gel, is a hard glassy material resembling quartz sand. It is chemically inert toward most substances, and particularly those used as refrigerating media except ammonia. It is obtained as a gelatinous mass, and after drying becomes a sponge which is granulated to from 8–20 mesh. It has the property of absorbing large quantities of vapors or liquids, SO_2 being used with it as a refrigerant. The internal volume of silica gel is about 50% of its total volume. In refrigeration 1 lb. SiO_2 will absorb 0.25 to 0.35 of SO_2. During adsorption, the latent heat of adsorption, about equal to the heat of vaporization, is liberated and must be removed. The adsorbed SO_2 is given up by activating the silica by heat, heat of desorption being about twice the latent heat. Silica gel is used also to dry air in air conditioning. Here, heat of desorption is more favor-

Table 32.—Strength of Liquor and Steam per Hour per Ton of Refrigeration

	Condenser Pressures, lb. per sq. in., gage								
	140			170			200		
	Suction Pressures, lb. per sq. in., gage								
	0	15	30	0	15	30	0	15	30
Strong liquor, percent..	24	35	42	22	32	38	18	28	36
Weak liquor, percent...	13.13	25.75	33.70	10.85	22.3	29.15	6.28	17.7	26.9
Lb. steam per hr. per ton of refrigeration:									
Generator..........	30.1	27.9	22.9	41.3	30.9	26.2	48.7	34.1	27.9
Liquor pump.......	1.7	1.6	1.4	2.1	1.9	1.8	2.4	2.3	2.2

Table 33.—Steam Consumption, Pounds per Hour per Ton of Refrigeration for Engine-driven Ammonia Compression and Absorption Machines

Type of Machine	Condenser Pressures, lb. per sq. in., gage								
	140			170			200		
	Suction Pressures, lb. per sq. in., gage								
	0	15	30	0	15	30	0	15	30
Simple non-condensing engine.	78.3	44.5	31.1	90.5	52.5	37.2	104.0	61.4	44.5
Compound condensing engine.	42.0	23.8	16.6	48.4	28.0	19.0	55.6	32.7	23.9
Absorption machine.........	31.8	29.5	24.3	43.4	32.8	28.0	51.1	36.4	30.1

able, being about equal to latent heat. In practice, about 1.5 × latent heat is required to drive off adsorbed water.

NH_3-$CaCl_2$ ABSORPTION SYSTEM uses as a solid absorbent a calcium chloride sponge, specially treated to increase its absorbing capacity (set in cement). The general method of operation is similar to that of the silica-gel system.

6. VAPOR CONDENSERS

HEAT ABSTRACTED.—Let h = heat content of gas leaving the compressor and arriving at the condenser; q_c = heat content of liquid leaving condenser; p_c = condenser pressure. Heat abstracted in the condenser is

$$(h - q_c) = h_c + C_p\,(t - t_c) - q_c = r_c + C_p\,(t - t_c) \quad . \quad . \quad . \quad . \quad [31]$$

C_p = mean specific heat between temperatures t_c and t; t = final compression temperature; t_c = condenser temperature; h_c = heat content of saturated vapor for pressure p_c; r_c = latent heat for temperature t_c. Values of h may be obtained from Figs. 4, 5 and 7 for NH_3, CO_2, and SO_2, and values of q_c from the saturated vapor tables.

AMOUNT OF CONDENSING WATER REQUIRED.—Let t_x and t_y = initial and final temperatures, respectively, of condensing water; C = lb. of condensing water required per lb. of ammonia circulated. Then, with $(h - q_c)$ as given above,

$$C = (h - q_c)/(t_y - t_x) \quad . \quad . \quad . \quad . \quad . \quad . \quad . \quad . \quad . \quad [32]$$

The temperature of condensing water for the warmer summer months should be ascertained in advance for each individual case. Usual assumed initial temperatures are 68° to 70° F. if the source is a reservoir, river or lake, or 55° if the source is an artesian well. The final temperature of condensing water determines the final temperature of the condensed liquid and, therefore, should be as low as practicable. It will be approximately 80° to 85° F. with an initial temperature of 70°, and the final temperature of the condensed liquid then will be about 95°, which gives a final compression pressure of 182.6 lb. per sq. in., gage, when ammonia is used as the refrigerating medium.

The most economical compression pressure depends upon the cost of producing 1 I.Hp. in a compressor cylinder for various pressures. This, in turn, depends upon the cost of water per gallon, either by purchase or by pumping.

Example.—Required the cooling water per ton of refrigeration for an ammonia condenser operating under a condensing pressure and temperature of 182.6 lb. gage and 95° F. respectively, with suction pressure and temperature of 15.25 lb. gage and 0° F., respectively. Initial temperature of circulating water is 70° F., and its final temperature 85° F.

Solution.—Weight of refrigerating medium circulated per minute per ton of refrigeration = $M = 200/(h_s - q_c) = 200/(538.5 - 71.3) = 0.428$ lb. of ammonia circulated per ton per minute.

Table 34.—Relative Performance of Ammonia Compression and Absorption Machines

Condenser: Temperature in degrees F.	Condenser: Absolute pressure, lb. per sq. in.	Refrigerating Coils: Temperature in degrees F.	Refrigerating Coils: Absolute pressure, lb. per sq. in.	Temperature of Absorber, degrees F.	Pounds of Ice-melting Effect per lb. of Coal — Compression Machine: Using 3 lb. of coal per hr. per I.Hp.	Compression Machine: Using 1.6 lb. of coal per hr. per I.Hp.	Absorption Machine*: Ammonia circulating pump exhausts into the generator	Absorption Machine*: Ammonia circulating pump exhausts to atmosphere through heater, yielding 212° temp. to the feedwater	Heat furnished to generator of absorption-machine, B.t.u. per lb. of ammonia circulated
61.2	110.6	5	33.7	61.2	38.1	71.4	38.1	33.5	969
59.0	106.0	5	33.7	59.0	39.8	74.6	38.3	33.9	967
59.0	106.0	5	33.7	130.0	39.8	74.6	39.8	35.1	931
59.0	106.0	−22	16.9	59.0	23.4	43.9	36.3	31.5	1000
86.0	170.8	5	33.7	86.0	25.0	46.9	35.4	28.6	988
86.0	170.8	5	33.7	130.0	25.0	46.9	36.2	29.2	966
86.0	170.8	−22	16.9	86.0	16.5	30.8	33.3	26.5	1025
86.0	170.8	−22	16.9	130.0	16.5	30.8	34.1	27.0	1002
104.0	227.7	5	33.7	104.0	19.6	36.8	33.4	25.1	1002
104.0	227.7	−22	16.9	104.0	13.5	25.3	31.4	23.4	1041

* 5% of water entrained in the ammonia will lower the economy of the absorption machine about 15% to 20% below the figures given in the table.

Heat rejected to the condenser per ton per minute $= M(h - q_c) = 0.428 \times (663 - 71)$
$= 253$ B.t.u. lb. of water per ton per minute $= 253/(85 - 70) = 16.8$,
or $16.8/8.33 = 2$ gal. $= 2 \times 1.75 = 3.5$ gal. per minute per ton of ice manufactured per 24 hr.

Condensing Surface Area

TYPES OF CONDENSERS.—The types of vapor condensers in common use are: The atmospheric, the double pipe and the shell and tube type.

The atmospheric type is recommended when its location on the roof or in the open room is unobjectionable. It is economical of water and easy of access for repairs or cleaning. See Table 35 for dimensions, and Table 36 for amount of pipe per ton of refrigeration usually allowed in atmospheric condensers.

The double-pipe type is recommended where condensing water is used for other purposes. The water is under pressure and the absence of drip and dampness makes this type suitable for all locations. It is essential that the water be soft, with no tendency to form scale. Condensers for ammonia usually are made 12 pipes high, the three upper pipes being 2 1/2 in. diam., and the balance 2 in. diam. The water pipe is 1 1/4 or 1 1/2 in. diam. throughout. The larger pipes at the top provide a wider annular space around the water pipe for gas when it reaches the condenser in a rarefied condition. See Table 37.

The shell and tube type is similar to the surface type of steam condensers. It is the

Table 35.—Dimensions of Atmospheric Ammonia Condensers

No. of Sections	Capacity, Tons of Refrig.	Space Required, ft.			Pipe Size, in.	Pipe Length, ft.	No. of Pipes	Total Pipe, ft.	Shipping Weight, lb.
		Length	Width	Height					
1	7 1/2	19 1/2	2	8	1 1/4	18	20	360	2,300
2	15	19 1/2	3	8	1 1/4	18	40	720	4,600
1	12 1/2	21 1/2	2	11 3/4	2	20	24	480	3,200
2	25	21 1/2	3 1/2	11 3/4	2	20	48	960	6,400
3	37 1/2	21 1/2	5 1/2	11 3/4	2	20	72	1440	9,600
4	50	21 1/2	7 1/4	11 3/4	2	20	96	1920	12,800
6	75	21 1/2	12	11 3/4	2	20	144	2880	19,200
8	100	21 1/2	16	11 3/4	2	20	192	3840	25,600
12	150	21 1/2	24	11 3/4	2	20	288	5760	38,400
16	200	21 1/2	32	11 3/4	2	20	384	7680	51,200

Table 36.—Dimensions of Double-pipe Ammonia Condensers

Capacity, Tons	No. of Sections	No. of Pipes	Length of Pipe, ft.	Total Feet of Pipe	Space Required						Shipping Weight, lb.
					Length, ft.	in.	Width, ft.	in.	Height, ft.	in.	
1	1	6	8	48	10	6	1	8	6	0	900
2	1	6	12	72	14	6	1	8	6	0	1,100
3	1	8	12	96	14	6	1	8	6	9	1,500
4	1	8	14	112	16	6	1	8	6	9	1,600
5	1	8	16	128	18	6	1	8	6	9	1,700
6	1	10	15	150	17	6	1	8	7	6	1,950
7	1	10	16	160	18	6	1	8	7	6	2,000
8	1	10	17.5	175	20	0	1	8	7	6	2,100
9	1	10	16.5	198	19	0	1	8	8	3	2,400
10	1	12	17.5	210	20	0	3	8	8	3	2,500
12.5	1	12	19	228	21	6	1	8	8	3	2,600
15	1	14	19	266	21	6	1	8	9	0	2,900
15	2	8	17.5	280	20	0	3	4	6	9	3,900
18	2	10	17.5	350	20	0	3	4	7	6	4,500
20	2	10	19	380	21	6	3	4	7	6	4,800
25	2	12	19	456	21	6	3	4	8	3	5,400
30	3	10	18	540	20	6	5	0	7	6	6,800
35	3	12	18	648	20	6	5	0	8	3	7,900
40	4	10	19	760	21	6	6	8	7	6	9,400
45	4	12	17.5	840	20	0	6	8	8	3	10,400
50	4	12	19	912	21	6	6	8	8	3	10,800
55	5	12	17.5	1050	20	0	8	4	8	3	12,700
60	5	12	18	1080	20	6	8	4	8	3	12,900

Table 37.—Dimensions of Shell and Tube Type Ammonia Condensers

Diameter of shell, in.	16	20	24	34	38	42	46
No. of vertical 2-in. tubes	21	23	51	115	144	191	227
No. of horizontal 2-in. tubes	16	26	36	96	136	176	204
Shell thickness, in.	1/4	5/16	3/8	1/2	5/8	5/8	3/4

most widely used type for ammonia, and is available in both horizontal and vertical designs. Tube sizes range from 1 1/4 to 2 1/2 in., and generally are No. 10 gage.

RATING OF ATMOSPHERIC CONDENSERS.—The water required per min. per ton of refrigeration in atmospheric condensers is, according to J. Levey, as follows:

Temperature, deg. F.	50	55	60	65	70	75	80	85
Gal. per min........	1/2	5/8	3/4	7/8	1	1 1/5	1 1/2	2

These quantities are based on water leaving the condenser at 95° F. and are the smallest quantities that should be allowed.

RATING OF DOUBLE-PIPE AMMONIA CONDENSERS.—Table 38 gives data on double-pipe condensers. The capacities and horsepower per ton of refrigeration are based on one section of a counter-current double-pipe condenser of 1 1/4 and 2 in. pipe, 12 pipes high, 19 ft. outside of water bends, and on water velocities of from 100 to 400 ft. per min., with an initial temperature of condensing water of 70° F. The horsepower per ton is for a single-acting compressor, and 15.67 lb. suction pressure. The friction in the water pump and connections should be added to water horsepower and to total horsepower.

THE FLOODED CONDENSER.—In the flooded condenser sufficient liquid ammonia is injected into a mixing chamber in direct contact with the ammonia gas to absorb the superheat and insure full saturation. Condensation is effected by cooling water applied externally. The advantages of the flooded type of condenser are lower first cost, reduced space and reduced upkeep. The flooded condenser is built in the atmospheric, double-pipe and shell-and-tube types. One section of a flooded atmospheric condenser, 12 pipes high, 20 ft. long, made of 2-in. pipe, or a total of 150 sq. ft. of pipe surface is rated as follows when operating at a condensing pressure of 185 lb. and various temperatures of water:

Temperature of water, deg. F....	55	60	65	70	75	80	85	90
Tonnage per coil................	54.6	49.9	44.4	39.3	33.5	26.5	17.1	11.8
Water per min. per ton of refrigeration, gal....................	1.1	1.19	1.36	1.53	1.79	2.26	3.86	5.68

Similar data for a double-pipe coil, 8 pipes high, 18 ft. 2 in. long, made of 2- and 3-in. pipe are:

Temperature of water, deg. F....	55	60	65	70	75	80	85	90
Tonnage per coil................	62.5	55.9	49.3	42.5	35.8	28.8	21.3	13.0
Water per min. per ton of refrigeration, gal....................	1.44	1.61	1.83	2.12	2.51	3.13	4.22	6.9

COOLING TOWERS IN REFRIGERATING PRACTICE.—Natural draft water-cooling towers are rated on a basis of a wind velocity of 5 mi. per hr., with major axis of tower at right angles to prevailing wind movement of the locality.

Table 39 gives data for guarantees of final water temperatures, based on 5 mi. per hr. wind velocity; towers 17 to 32 ft. high; water circulation, 1.5 to 3.0 gal. per min. per sq. ft. of distributing deck; decks 4 to 12 ft. wide. Wet bulb temperature is that of entering outside air to tower.

In mechanical towers, air is circulated by fan or blower and is independent of outside wind movement. The two types are forced and induced draft. Table 40 gives the prac-

Table 38.—Double-pipe Condenser Data

Velocity through 1 1/4-in. Pipe, ft. per min.	Condensing Water: Total Gallons Used per min.	Condensing Water: Gallons per min. per Ton Refrig.	Friction Through Coil, lb. per sq. in.	Capacity, Tons Refrig. per 24 Hr.	Condensing Pressure, lb. per sq. in.	Hp. per Ton of Refrigeration: Engine Driving Compressor	Hp. per Ton of Refrigeration: Circulating Water through Condenser	Hp. per Ton of Refrigeration: Total Engine and Water Circulation
HIGH PRESSURE CONSTANT								
100	7.77	1.16	1.69	6.7	185	1.71	0.0012	1.7112
150	11.65	1.165	3.05	10.	185	1.71	.002	1.712
200	15.54	1.165	5.08	13.4	185	1.71	.004	1.714
250	19.42	1.18	7.89	16.4	185	1.71	.006	1.716
300	23.31	1.24	11.41	18.8	185	1.71	.009	1.719
400	31.08	1.30	20.51	24.	185	1.71	.016	1.726
CAPACITY CONSTANT								
100	7.77	0.777	1.69	10.	225	2.04	0.0008	2.0408
150	11.65	1.165	3.05	10.	185	1.71	.002	1.512
200	15.54	1.554	5.08	10.	165	1.54	.005	1.545
250	19.42	1.942	7.89	10.	155	1.46	.009	1.469
300	23.31	2.331	11.41	10.	148	1.40	.016	1.416
400	31.08	3.108	20.51	10.	140	1.33	.038	1.368

tice in mechanical draft towers, based on a water circulation of 2.07 gal. per min. per sq. ft. of wetted filling surface. In general the cooling surface or filling for towers is cypress or redwood, copper nailed. See also p. 9–21.

Table 39.—Final Water Temperatures Natural Draft Towers

Wet Bulb Temperature, deg. F.	Cooling Range $(t_1 - t_2)$ *				
	5	10	15	20	30
80	81.6	83.0	84.3	85.4	86.9
75	77.0	78.6	80.2	81.6	83.6
70	72.4	74.4	76.2	77.8	80.4
65	67.8	70.2	72.3	74.2	77.1
60	63.2	66.0	68.4	70.4	74.0

* t_1 = temp. water from condenser, deg. F.; t_2 = temp. water leaving tower, deg. F. Limit of t_2 is wet bulb temp. of entering air.

Table 40.—Final Water Temperatures of Mechanical Draft Towers

Wet Bulb Temp., deg. F..	80	75	70	65	60
$(t_1 - t_2) =$ 5............	81.8	77.0	72.3	67.8	63.4
	82.6	78.2	74.0	69.6	65.6
	83.4	79.4	75.4	71.5	67.8
	84.2	80.4	76.6	73.0	69.4

7. BRINE CIRCULATING SYSTEM

In the brine circulating system, Fig. 18, brine circulates through coils in the storage room. The evaporating coils of the refrigerating system are located in the brine tank, and the brine acts simply as a heat transfer medium. The principal advantage of this system is that the refrigeration can be stored and the refrigerating machine shut down for part of the time. The length of refrigerant piping and number of joints also are reduced to a minimum.

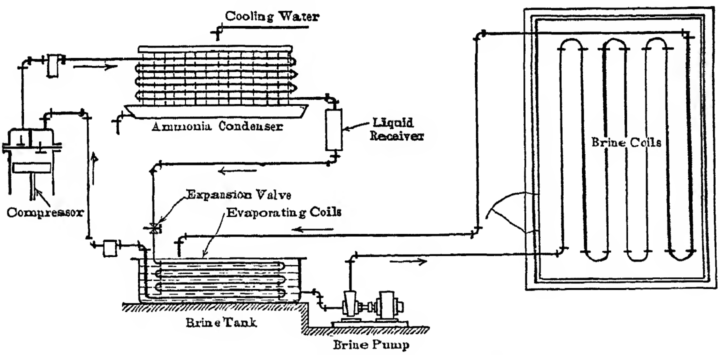


Fig. 18. Arrangement of Brine Circulating System

The brine system is more expensive to operate, as a lower back-pressure must be carried by the machine at all times, due to the fact that at least 10° difference in temperature must be maintained between outgoing brine from the tank and the evaporating coils, in order to obtain the necessary heat transfer from brine to the refrigerant. The brine tank sometimes is placed in the room to be cooled.

SIZE OF BRINE TANK.—For continuous operation about 60 cu. ft. of brine per ton of refrigeration is required. For non-continuous operation weight of brine required is

$$W = 12{,}000\, R(24 - h) \div \{S_b(t_1 - t_2)\} \quad . \quad . \quad . \quad . \quad . \quad . \quad [33]$$

where R = refrigeration, tons per 24 hr.; h = number of hours per 24 that machine operates; S_b = specific heat of brine solution (Table 41); t_1 = initial temperature of brine when machine is starting; t_2 = final temperature of brine when machine is shut down.

The specific gravity of the brine solution to be used depends on the lowest tempera-

ture t_2 to be carried, which fixes the percentage of salt in making up the solution. If x = specific gravity of solution at 60° F. the weight d, lb. per cu. ft. = $62.5x$, approximately, and the contents of the brine tank = W/d cu.ft. The size of the brine circulating pump is calculated in the same manner as that of a water pump. If a reciprocating pump is used, allowance for slip must be made.

RATING OF MACHINE.—The refrigerating machine and condensing apparatus must produce R tons of refrigeration in h hr., and also dispose of the heat absorbed by the brine tank. If the brine tank be located in one of the rooms to be refrigerated, the heat transmission of the tank produces useful refrigeration and is not an extra tax on the machine. The rating of the machine (tons per 24 hr.) required will be: $C = (R/h) \times 24$. Rating is based on a suction temperature in evaporator coils approximately 10° lower than lowest brine temperature required. A shell-and-tube, shell-and-coil or double-pipe brine cooler is used in preference to a submerged coil. The shell-and-tube type brings a larger body of ammonia in contact with the cooling surface than the others.

The three-pipe balanced system is used in larger installations, especially where brine is to be pumped against a great head. The balancing brine tank is placed slightly higher than the top coils in the building; pump and brine cooler are located in the basement. The pump discharges brine from the bottom of the balancing tank through the cooler to the room coils, whence it returns to the balancing tank.

BRINE SOLUTIONS.—Common Salt (NaCl) corrodes pipes so freely that leakage, repairs, etc., offset the lower first cost of salt. Refrigerated brines must circulate so close to 0° F., the freezing point of common salt brine, that crystals of salt separate out, clog pipes, increase friction and tend to insulate pipes, thus requiring that more brine be circulated than would be necessary with clean pipes. See Table 41 for properties.

Calcium Chloride ($CaCl_2$) brine has no action on the pipe and may be used at temperatures below 0° F. A somewhat greater cost of calcium chloride, compared to common salt, is offset by the fact that 25% less in weight is required. See Table 42 for properties.

BRINE PIPING.—Factors entering into design of brine piping are: 1. Heat absorbed by each coil, basing calculations upon refrigeration required for the various rooms or refrigerators. 2. Amount of brine to be circulated to absorb the calculated amount of heat with a temperature rise of from 2° to 5° F. 3. Pipe friction of the system.

Quantity of Brine to be Circulated per deg. F. rise of temperature per ton of refrigeration, when the specific heat of brine is 0.829, is 242.4 lb. or 25.2 gal. per min. If the number and character of fittings is not definitely known, an estimated friction loss in fittings, etc., of from 20% to 40% usually will be sufficiently accurate, using the latter figure for small jobs and the former for, say, large drinking-water systems.

Regulation of flow is facilitated if the system is so arranged that it always will be full of brine while in operation and also if it is kept under pressure. To insure this latter, it may be advisable to place a reducing valve on the discharge.

Direct Expansion versus Brine Circulating System with Ammonia

DIRECT EXPANSION.—Assume a temperature of 20° F. in a refrigerating room piped with direct expansion coils. The ammonia temperature then must be approximately 10° F., corresponding to a suction pressure of 23.3 lb., gage. If the condensing pressure is 182.6 lb., gage, corresponding to a temperature of 95° F., then $200/(541.2 - 71.3) = 0.425$ lb. of ammonia must be pumped per min. per ton of refrigeration. The specific volume of

Table 41.—Properties of Common Salt Brine at 60° F.

Degrees Baumé	Degrees Salinometer	Specific Gravity	Percent Salt	Wt. of 1 gal., lb.	Wt. of 1 cu. ft., lb.	Freezing Point, deg. F.	Specific Heat
5	20	1.037	5	8.7	64.7	25.4	0.96
10	40	1.073	10	9.0	67.0	18.6	.892
15	60	1.115	15	9.3	69.6	12.2	.855
19	80	1.150	20	9.6	71.8	6.9	.829
23	100	1.191	25	9.9	74.3	1.0	.783

Table 42.—Properties of Calcium Chloride Brine

Specific Gravity at 68° F.	Lb. 75% Solvay $CaCl_2$ per gal.	Lb. 75% Solvay $CaCl_2$ per cu. ft.	Freezing Point, deg. F.	Specific Heat
1.100	1.46	10.9	18.0	0.88
1.125	1.83	13.7	12.5	.87
1.150	2.20	16.5	+ 6.5	.85
1.175	2.59	19.4	− 2.0	.835
1.200	2.99	22.4	−12.5	.81
1.225	3.38	25.3	−23.5	.80
1.250	3.75	28.3	−36.5	.77

the gas is 7.34 cu. ft., and 0.425 × 7.34 = 3.12 cu. ft. of saturated vapor to be handled by the compressor per ton per min. requiring 1.22 compressor I.Hp. per ton per 24 hr.

BRINE CIRCULATING SYSTEM—CONTINUOUS OPERATION.—Brine temperature in room coils must approximate that of the ammonia in the direct expansion system; *i.e.*, 10° F. Temperature of ammonia in the evaporating coils in the brine tank will be approximately 0° F. in which case 0.427 lb. of ammonia must be circulated per min. per ton of refrigeration per 24 hr. The specific volume of saturated ammonia at 0° F. is 9.19 cu. ft. per lb. The compressor must handle 0.427 × 9.19 = 3.92 cu. ft. per min. per ton of refrigeration per 24 hr., the compressor I.Hp. being 1.46. Under the conditions, assumed, therefore, with the brine circulating system continuously operated the compressor capacity is 25.6% larger, and the power consumption 20% greater than with the direct expansion system, no allowance being made for the power required to operate brine circulating pump.

ICE MANUFACTURE

Two systems of ice manufacturing in general use are: 1. Can system. 2. Plate system. The former may be subdivided into *a*. Distilled water system; *b*. Raw water system. Ammonia is the commonly used refrigerant in ice plants.

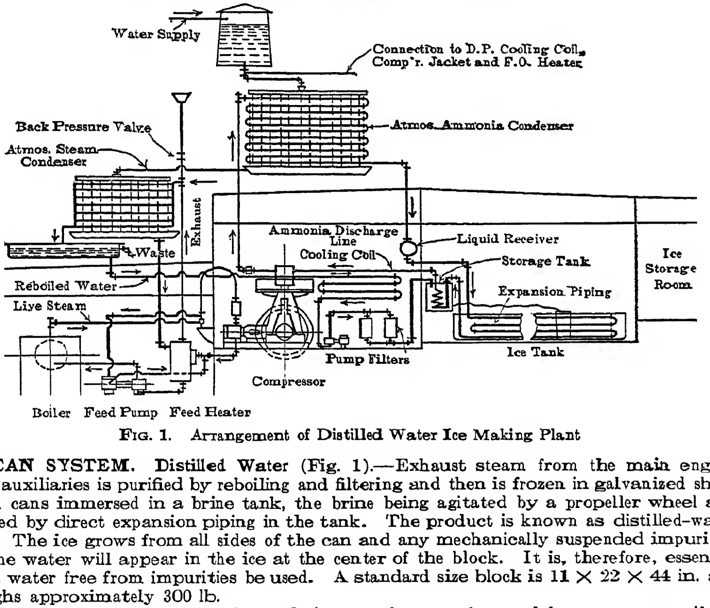


Fig. 1. Arrangement of Distilled Water Ice Making Plant

CAN SYSTEM. Distilled Water (Fig. 1).—Exhaust steam from the main engine and auxiliaries is purified by reboiling and filtering and then is frozen in galvanized sheet steel cans immersed in a brine tank, the brine being agitated by a propeller wheel and cooled by direct expansion piping in the tank. The product is known as distilled-water ice. The ice grows from all sides of the can and any mechanically suspended impurities in the water will appear in the ice at the center of the block. It is, therefore, essential that water free from impurities be used. A standard size block is 11 × 22 × 44 in. and weighs approximately 300 lb.

Raw Water.—A clear merchantable ice may be manufactured from raw water if the water in the cans is agitated during the entire freezing period by air discharged under a pressure of 3 to 6 lb. per sq. in. at or near the bottom of the cans through a $^1/_{16}$-in. opening. The unfrozen water, heavily charged with impurities, is pumped out of the center of the block before this part is entirely closed and frozen, and the space is filled with raw water containing only the initial amount of impurities. As this system does not require distilling apparatus, the most economical form of drive may be used, the majority of plants being motor-driven. The size of the motor should be ample to provide for a higher terminal pressure than ordinarily is considered for 70° F. condensing water, to prevent possible overloading of motor. See Fig. 2 for capacities of can system.

PLATE SYSTEM.—A tank approximately 10 ft. deep by 12 ft. wide, divided by $^1/_2$-in. plates, bolted to direct expansion piping, into compartments 30 in. wide is used. The plates form the freezing surface. Ice grows from the plates outwards and mechani-

Table 1.—Displacement and Brake Horsepower per Ton Ice-making in 24 Hours. (York Mfg. Co.)

Single-acting (S.A.) and Double-acting (D.A.) Ammonia Compressors—Dry Compression

Condenser Gage Pressure and Corresponding Temperature = Temperature of Liquid at Expansion Valve	Suction Gage Pressure and Corresponding Temperatures																	
	5 lb. −17.5° F.		7.5 lb. −12.7° F.		10 lb. −8.5° F.		12.5 lb. −4.6° F.		15.67 lb. 0° F.		17.5 lb. +2.5° F.		20 lb. 5.7° F.		22.5 lb. 8.7° F.		25 lb. 11.5° F.	
	Cu. In. Displacement per min. per ton Ice Making	Brake Hp. at the Machine	Cubic Inches Displacement	Brake Hp. at the Machine	Cubic Inches Displacement	Brake Hp. at the Machine	Cubic Inches Displacement	Brake Hp. at the Machine	Cubic Inches Displacement	Brake Hp. at the Machine	Cubic Inches Displacement	Brake Hp. at the Machine	Cubic Inches Displacement	Brake Hp. at the Machine	Cubic Inches Displacement	Brake Hp. at the Machine	Cubic Inches Displacement	Brake Hp. at the Machine
85 lb. = 56° F. S.A....	17,776	1.971	15,696	1.786	14,080	1.654	12,496	1.496	11,056	1.355	10,544	1.269	9,632	1.162	8,976	1,077	8,400	0,994
D.A...	20,640	2.253	18,000	2.024	15,968	1.866	14,528	1.681	12,560	1.505	11,840	1.426	10.880	1.285	10,096	1.188	9,312	1.091
105 lb. = 65.7° F. S.A....	18,560	2.288	16,368	2.059	14,640	1.918	13,056	1.769	11,520	1.602	10,960	1.522	10,048	1.408	9,312	1.302	8,736	1.223
D.A...	21,600	2.640	18,816	2.376	16,672	2.200	15,104	1.989	13,120	1.795	12,368	1.718	11,360	1.566	10,480	1.450	9,760	1.338
125 lb. = 74.3° F. S.A....	19,360	2.587	16,992	2.367	15,200	2.200	13,568	2.024	12,000	1.866	11,360	1.763	10,400	1.646	9,632	1.531	9,040	1.443
D.A...	22,528	3.010	19,600	2.731	17,360	2.517	15,648	2.302	13,680	2.103	12,832	2.003	11,776	1.857	10,848	1.711	10,080	1.610
145 lb. = 82° F. S.A....	20,173	2.911	17,632	2.666	15,698	2.464	14,080	2.288	12,526	2.103	11,776	2.015	10,824	1.874	9,984	1.760	9,338	1.660
D.A...	23,432	3.381	20,400	3.087	18,080	2.837	16,224	2.619	14,242	2.390	13,328	2.288	12,200	2.112	11,216	1.975	10,435	1.855
165 lb. = 89° F. S.A....	20,872	3.227	18,272	2.966	16,237	2.746	14,624	2.552	12,947	2.361	12,192	2.259	11,184	2.114	10,336	1.989	9,643	1.885
D.A...	24,325	3.761	21,184	3.432	18,752	3.171	16,800	2.930	14,758	2.691	13,824	2.572	12,637	2.388	11,616	2.241	10,802	2.112
185 lb. = 95.5° F. S.A....	21,586	3.543	18,880	3.265	16,779	3.027	15,152	2.823	13,379	2.617	12,624	2.514	11,550	2.352	10,688	2.218	9,957	2.107
D.A...	25,238	4.143	22,016	3.784	19,440	3.508	17,376	3.238	15,288	2.992	14,320	2.860	13,082	2.663	12,016	2.506	11,176	2.365
205 lb. = 101.4° F. S.A....	22,315	3.858	19,520	3.555	17,334	3.307	15,680	3.089	13,808	2.871	13,040	2.763	11,920	2.587	11,008	2.446	10,272	2.329
D.A...	26,179	4.525	22,816	4.136	20,144	3.843	17,936	3.552	15,824	3.291	14,816	3.144	13,534	2.939	12,400	2.763	11,555	2.619
225 lb. = 107.3° F. S.A....	23,120	4.173	20,160	3.854	17,920	3.608	16,192	3.353	14,240	3.142	13,472	3.010	12,320	2.851	11,360	2.679	10,592	2.587
D.A...	27,154	4.910	23,616	4.495	20,800	4.173	18,528	3.854	16,320	3.590	15,328	3.425	13,920	3.238	12,800	3.027	11,840	2.886
245 lb. = 112.5° F. S.A....	23,840	4.435	20,768	4.136	18,400	3.872	16,736	3.608	14,720	3.397	13,888	3.247	12,720	3.098	11,712	2.904	10,880	2.816
D.A...	28,000	5.280	24,448	4.831	21,520	4.506	19,072	4.162	16,832	3.872	15,840	3.714	14,400	3.502	13,184	3.291	12,288	3.133

Brake horsepower at machine means the effective Hp. actually delivered at the belt-wheel of the machine or at the shaft for direct-connected machines. For belt-driven machines, 5% should be added to the brake Hp. for friction and slippage of belt.

Table is based on 70° F. water to storage tank. For different temperatures of water multiply these figures by percentages in table below.

Tons of Refrigeration Required to Produce One Ton of Ice, When the Water To Be Frozen Is Delivered to the Apparatus Operated by the Machine, at the Temperatures Given:

Temperature of water, degrees F.	50	60	70	80	90
Tons of refrigeration per ton of ice	1.46	1.53	1.60	1.67	1.74
Refrigeration work, percent, based on 70° F. water = 100 percent	91 1/4	95 5/8	100	104 3/8	108 3/4

cally suspended impurities in the water are separated and fall to the bottom of the tank. The ice cake usually is thawed from the plate by passing hot gas through the coils. Distilled water is not required for the manufacture of clear ice, and electric drive may, therefore, be used. The ice gradually forms in from 8 to 10 days to a thickness of 12 to 14 in.

The freezing tank area required by the plate system is about 12 times that required by the can system and the cubical contents are about 4 times as great. The advantage of the plate system that clear ice is produced without special apparatus is offset by the fact that the building up of the ice is slow and expensive; also, for continual operation several tanks are required, so that one or more may be frozen while the others are being emptied. The cost of the plate system is about $^1/_3$ more than that of the can system. In the can system, ice is drawn throughout the 24 hours while in the plate system the entire product is harvested, cut and stored in a few hours.

THE DISTILLING SYSTEM.—Exhaust steam from the main engine and auxiliaries, after passing through an oil separator, is condensed at atmospheric pressure in a steam condenser. About 10% is diverted to the feedwater heaters. The feedwater first is used as cooling water in the ammonia and steam condensers and distilled water cooler. The condensed steam flows to the reboiler and skimmer, where a live-steam coil boils the water at atmospheric pressure, driving out air and absorbed gases. Oil and other impurities that rise to the surface are skimmed off. The steam coil either may be perforated or closed. If perforated, the high-pressure steam is reduced to atmospheric pressure and mixes with the distilled water. If closed, condensation returns to the feedwater heater.

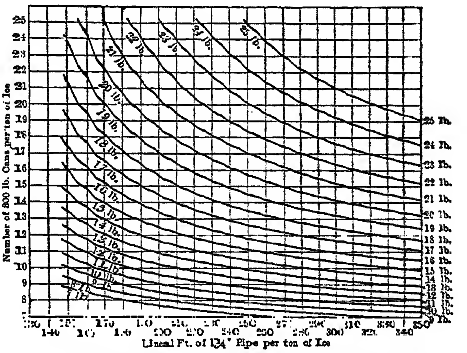


FIG. 2. Number of Cans per Ton of Ice for Various Suction Pressures and Areas of Evaporated Surface

The reboiled distilled water is reduced to a temperature of approximately 90° F. in the distilled water cooler, usually of the double-pipe type. After filtering through quartz, sand or maple charcoal filters, it is stored and further cooled in the storage tank or fore-cooler, containing a coil through which gas from the evaporating coils passes on its way to the compressor. When reduced to a temperature of approximately 40° F., it is used to fill the cans. A regulating device automatically controls the flow of water from the reboiler to the storage tank and also controls the pump circulating distilled water through coolers and filters.

ICE-MAKING CAPACITY.—The ice-making capacity of a refrigerating machine is approximately 61% of its refrigerating capacity, as given under Rating of Refrigerating Machine, p. 10-03. Table 1 gives data from which ice-making capacity may be determined.

SIZE OF FREEZING TANK.—Let W = weight of ice, lb., to be pulled every 24 hr. = tons rating × 2000; H = freezing time, hr., N = number of cans required; C = weight of 1 block of ice, lb. Then $N = WH/(C \times 24)$. For 300-lb. blocks, $H = 50$ and $N = W/144$, and the number of 300-lb. cans per ton capacity rating of plant is 14. Some manufacturers recommend 16 cans per ton of capacity. Dimensions of the freezing tank may be approximated from data given in Table 3.

Table 2.—Suction Pressures Required for Various Brine Temperatures

Back pressure, lb. per sq. in., gage..	10	15	20	25	30
Brine temperature, deg. F..........	0	10	15	20	25

Table 3.—Size of Standard Ice Cans and Freezing Time

Size of Can, in.	Weight of Ice Block, lb.		Gage of Metal		Time of Freezing, hr.	
	Normal	Actual	Sides	Bottom	15° Brine	18° Brine
6×12×26	50	56	20	20	15	20
8×16×32	100	110	18	16	30	36
8×16×42	150	165	18	16	30	36
11×22×32	200	220	18	14	50	60
11×22×44	300	315	16	14	50	60
11×22×57	400	415	16	14	50	60

EXPANSION PIPE.—Approximately 80 to 100 sq. ft. of pipe surface per ton of ice-making capacity is required with good brine agitation. The maximum length of pipe for one expansion is 1200 ft. See Table 5.

Table 4.—Dimensions of Freezing Tanks

Number Cans, Wide or Long	Width, ft.	Width, in.	Length, ft.	Length, in.	Number Cans, Wide or Long	Width, ft.	Width, in.	Length, ft.	Length, in.
6	9	0	16	6	22	28	3	49	9
8	11	3	20	9	24	30	9	53	9
10	13	9	24	9	26	33	0	58	0
12	16	3	29	0	28	35	6	62	3
14	18	6	33	0	30	38	0	66	3
16	21	0	37	3	32	....		70	6
18	23	6	41	3	34	....		74	6
20	25	9	45	6					

Table 5.—Lineal Feet of Pipe per Ton of Ice

Pipe size, in.........		1 1/4	/2	2
15° brine, ft. of pipe.	400	320	270	210
18° brine, ft. of pipe.	450	360	310	240

Table 6.—Water Consumption per Ton of Ice in Compression Plants (O. Gueth)

Initial temp. water over ammonia condenser, deg. F.....	55	60	70	80
Water temp. entering steam condenser, deg. F..........	80	85	90	95
Water temp. leaving steam condenser, deg. F...........	125	125	125	125
Gallons per minute	4	4.5	5.15	6

FUEL CONSUMPTION IN ICE PLANTS operating on the compression system may be computed as follows: Assuming an average steam consumption for the main engine (Corliss type) of 27 lb. per I.Hp.-hr. and an additional 4% to operate the feed pump, total steam required is 28 lb. per I.Hp.-hr. The agitator engine and distilled water pump will use an additional 8 lb. of steam per hr. per ton of ice-making capacity. Then, since each ton of ice made requires 1.75 tons of refrigeration, and each ton of refrigeration requires 1.5 compressor I.Hp., the steam required per ton of ice-making capacity for main engine and feed pump will be $(1.75 \times 1.5 \times 28)/0.85 = 87$ lb. of steam, when the mechanical efficiency of the engine and compressor is taken at 85%. Exhaust steam available is $87 + 8 = 95$ lb. Adding approximately 10% for waste in the reboiler and in filling the cans, 105 lb. of water must be evaporated per hour, equivalent to approximately 3 1/2 boiler Hp. per ton of ice-making capacity.

Section 11

HEATING, VENTILATING AND AIR CONDITIONING

By Louis A. Harding

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HEATING, VENTILATING AND AIR CONDITIONING

By Louis A. Harding

HEATING

1. ESTIMATING HEATING REQUIREMENTS OF BUILDINGS*

HEAT REQUIRED TO BE SUPPLIED.—The amount of heat, in B.t.u., to be supplied by the heating apparatus to a building to maintain the inside temperature above that of the outside, commonly termed heat losses, is: (*a*) The heat required to offset the heat transmission of the walls, ceiling or roof, and floor. This loss of heat depends upon the type and materials of construction used and the temperature difference to be maintained between the inside and the outside of the building. (*b*) The heat required to warm the air entering the building from the outside, either by infiltration or purposely introduced for ventilation. (*c*) The heat supplied by persons, lights, machinery and motors, which may be deducted from the sum of Items (*a*) and (*b*) to obtain net amount of heat to be supplied by the heating system. [Item (*c*) usually is not considered.] In the design of heating installations it is customary to estimate the amount of heat per hour to be supplied by the apparatus. The total heat to be supplied, B.t.u. per hour, is $H = [(\text{Item } a) + (\text{Item } b) - (\text{Item } c)]$.

This is the assumed amount of heat that must be supplied by the heat emitting apparatus, *i.e.*, radiators, unit heaters, etc., after a condition of heat equilibrium has been established for the assumed temperature differential. The amount of heat to be supplied by the boiler or furnace during the "starting up" period with cold rooms and apparatus will be considerably greater. See Rating of Heating Boilers, p. 11–11.

TEMPERATURES.—The inside temperatures to be maintained for various classes of work are given in Table 1. The outside temperature for which the heating installation should be designed is fixed by the lowest outside temperature that is liable to continue for several days during the heating season.

HEAT TRANSMISSION OF BUILDING CONSTRUCTION.—Heat is transferred to and from a surface by *radiation* and *convection*. The transfer of heat through a material is by *conduction*.

As it is difficult to measure separately the heat transfer by radiation and convection, they ordinarily are combined. Let t and t_1 = temperature of the inside air and wall surfaces respectively; t_0 and t_2 = temperature of the outside air and outside wall surface respectively. K_1 and K_2 = the combined coefficient of radiation and convection for the inside and outside wall surfaces respectively. (B.t.u. transferred per sq. ft. per hour per

Table 1.—Usual Inside Temperature Specified

Kind of Building	Degrees F.	Kind of Building	Degrees F.
Public buildings	68–72	Bath-rooms	85
Factories	65	School-rooms	68–70
Machine-shops	60–65	Hospitals	72–75
Foundries, boiler-shops, etc.	50–60	Paint-shops	80
Residences	70		

Table 2.—Values of K in Still Air Tests

Brickwork	1.40	Glass window	1.50
Concrete	1.30	Sheet asbestos	1.40
Corkboard	1.25	Magnesia board	1.45
Cement plaster	0.93	Wood (finished)	1.40

The average of the above values of K is 1.34.

* Largely condensed from Heating, Ventilating and Air Conditioning, by L. A. Harding and A. C. Willard. John Wiley & Sons, Inc.

degree difference in temperature, between the surface and air in contact with the surface.) Then $K_1(t - t_1) = K_2(t_2 - t_0)$. Table 2 gives values of K determined in a series of conductivity, absorption and emissivity tests by Harding and Willard at Univ. of Ill. in 1915.

A wind velocity of 15 miles per hour over the surface increases the values of K approximately 3 times the tabular values. Hence, it is recommended that for inside wall surfaces the tabular values of K be used, and for outside surfaces (tabular values $\times$ 3).

CONDUCTION.—Let C = coefficient of conductivity, or B.t.u. transferred through a material per sq. ft. per hr. per deg. F. difference in temperature of the two surfaces, per in. of thickness; x = thickness of material, in. Then $(C/x)(t_1 - t_2)$ = heat transmitted by conduction per sq. ft. per hour. Tables 4, 5 and 6 give conductivities for various materials for the thickness stated. See *The Guide*, published by A.S.H.V.E., each year for a complete report of conductivity tests covering building and insulation materials. Tables 4, 5 and 6 were abstracted from the 1932 *Guide*.

Table 3.—Climatic Conditions Compiled from United States Weather Bureau Records

State	City	Av. Temp., deg. F.*	Lowest Temp., deg. F.	Av. Wind Velocity, miles per hr.†	Direction of Prevailing Wind	State	City	Av. Temp., deg. F.*	Lowest Temp., deg. F.	Av. Wind Velocity, miles per hr.†	Direction of Prevailing Wind
Ala.	Mobile	57.7	−1	8.3	N	Nev.	Tonopah	39.6	−7	9.9	SE
	Birmingham	53.9	−10	8.6	N		Winnemucca	37.9	−28	9.5	NE
Ariz.	Phoenix	59.5	16	3.9	E	N. H.	Concord	33.4	−35	6.0	NW
	Flagstaff	34.9	−25	6.7	SW	N. J.	Atlantic City	41.6	−7	10.6	NW
Ark.	Fort Smith	49.5	−15	8.0	E	N. Y.	Albany	35.1	−24	7.9	S
	Little Rock	51.6	−12	9.9	NW		Buffalo	34.7	−14	17.7	W
Cal.	San Francisco	54.3	29	.....	N		New York	40.3	−6	13.3	NW
	Los Angeles	58.6	28	.....	NE	N. M.	Santa Fe	38.0	−13	7.3	NE
Colo.	Denver	39.3	−29	7.4	S	N. C.	Raleigh	49.7	−2	7.3	SW
	Grand Junction	39.2	−16	5.6	SE		Wilmington	53.1	5	8.9	SW
Conn.	New Haven	38.0	−14	9.3	N	N. D.	Bismarck	24.5	−45	.....	NW
D. C.	Washington	43.2	−15	7.3	NW		Devil's Lake	18.9	−44	11.4	W
Fla.	Jacksonville	61.9	10	8.2	NE	Ohio	Cleveland	36.9	−17	14.5	SW
Ga.	Atlanta	51.4	−8	11.8	NW		Columbus	39.9	−20	9.3	SW
	Savannah	58.4	8	8.3	NW	Okla.	Okla. City	48.0	−17	12.0	N
Idaho	Lewiston	42.5	−13	4.7	E	Ore.	Baker	34.1	−20	6.0	SE
	Pocatello	36.4	−20	9.3	SE		Portland	45.9	−2	6.5	S
Ill.	Chicago	36.4	−23	17.0	SW	Pa.	Philadelphia	41.9	−6	11.0	NW
	Springfield	39.9	−24	10.2	NW		Pittsburgh	40.8	−20	13.7	NW
Ind.	Indianapolis	40.2	−25	11.8	S	R. I.	Providence	37.6	−9	14.6	NW
	Evansville	44.1	−15	8.4	S	S. C.	Charleston	56.9	7	11.0	N
Iowa	Dubuque	33.9	−32	6.1	NW		Columbia	53.7	−2	8.0	NE
	Sioux City	32.1	−35	12.2	NW	S. D.	Huron	28.1	−43	11.5	NW
Kan.	Concordia	38.9	−25	7.3	N		Rapid City	32.3	−34	7.5	W
	Dodge City	40.2	−26	10.4	NW	Tenn.	Knoxville	47.0	−16	6.5	SW
Ky.	Louisville	45.2	−20	9.3	SW		Memphis	50.9	−9	9.6	NW
La.	New Orleans	61.5	7	9.6	N	Texas	El Paso	53.0	−2	10.5	NW
	Shreveport	56.2	−5	7.7	SE		Fort Worth	54.7	−8	11.0	NW
Me.	Eastport	31.1	−23	13.8	W		San Antonio	60.7	4	8.2	N
	Portland	33.6	−17	10.1	NW	Utah	Modena	38.1	−24	8.9	W
Md.	Baltimore	43.6	−7	7.2	NW		Salt Lake City	40.0	−20	4.9	SE
Mass.	Boston	37.6	−13	11.7	W	Vt.	Burlington	29.3	−27	12.9	S
Mich.	Alpena	29.1	−27	11.3	W	Va.	Norfolk	49.1	2	9.0	N
	Detroit	35.4	−24	13.1	SW		Lynchburg	45.2	−7	5.2	NW
	Marquette	27.6	−27	11.4	NW		Richmond	47.4	−3	7.4	S
Minn.	Duluth	25.1	−41	11.1	SW	Wash.	Seattle	45.3	3	9.1	SE
	Minneapolis	29.6	−33	11.5	NW		Spokane	37.5	−30	.....	SW
Miss.	Vicksburg	56.0	−1	7.6	SE	W. Va.	Elkins	38.8	−21	4.8	W
Mo.	St. Joseph	40.3	−24	9.1	NW		Parkersburg	41.9	−27	6.6	S
	St. Louis	43.3	−22	11.8	NW	Wis.	Green Bay	28.6	−36	12.8	SW
	Springfield	43.0	−29	11.3	SE		LaCrosse	31.2	−43	5.6	NW
Mont.	Billings	34.7	−49	.....	W		Milwaukee	33.0	−25	11.7	W
	Havre	27.7	−57	8.7	SW	Wyo.	Sheridan	31.0	−45	5.3	NW
Nebr.	Lincoln	37.0	−29	10.9	N		Lander	28.9	−36	3.0	NE
	North Platte	34.6	−35	9.0	W						

* Oct. 1 to May 1. † In Dec., Jan., :

CALCULATION FOR THE HEAT TRANSMISSION OF WALLS.—The amount of heat absorbed by the inside wall surface, the amount conducted through the wall and the amount emitted by the outside wall surface is evidently the same. Let u = heat transmission of the actual wall per sq. ft. per hour per degree F. difference in temperature of the air on the two sides. Then

$$u = 1 \div \{(1/K_1) + (1/K_2) + (x/C) + (x_1/C_1) + (x_2/C_2) + \text{etc.}\} \quad . \quad . \quad [1]$$

The value of (x/C) for thin plates, glass or roofing material is so small that it safely may be neglected in ordinary calculations. For example, the unit heat transmission u for a window is assumed as $(1/1.5) + (1/4.5) = 1.12$.

EXAMPLE.—Determine the heat transmission of an 8-in. brick wall, furred with 1 1/2-in. split furring tile laid up in plaster, and finished with 1/2-in. plaster. $K_1 = 1.34$; $K_2 = 3 \times 1.34 = 4.02$; C, for brick, $= 5$; C, for split tile, $= 1.25$; C, for plaster, $= 2.32$ (see Table 4).

$$u = 1 \div \left(\frac{1}{1.34} + \frac{1}{4.02} + \frac{8}{5} + \frac{1}{1.25} + \frac{0.50}{2.32}\right) = 0.28 \text{ B.t.u. per sq. ft. per hr. per deg. F. difference in temperature of air inside and outside.}$$

Heat is transmitted through air space construction from one surface to another by radiation and convection. The conduction of one air space, as shown by the experiments of Prof. F. B. Rowley, is approximately $C = 1.0$ for the usual temperatures encountered in heating practice.

EXPOSURE FACTOR.—Some engineers increase the calculated heat transmission loss of wall and glass surfaces when they are exposed to high winds, to compensate for the increased convection and the leakage of cold air through the wall. The following are commonly used arbitrary factors by which heat transmission losses are multiplied: North, northeast and northwest exposure, if winds are an important factor, 1.15 to 1.30; east or west walls, moderately exposed, 1.10 to 1.20; south walls, 1.0. If the data from Tables 4, 5 and 6 are used, *it is not necessary* to use an exposure factor.

The heat transmission of stone walls is approximately 50% greater than that of brick

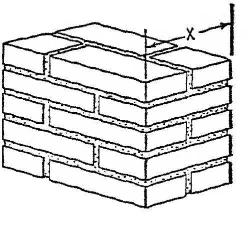
Table 4.—Coefficients of Transmission (C) of Solid Brick Walls

NOTE.—These coefficients are expressed in B.t.u. per hr. per sq. ft. per 1° F. temperature difference between the air on the two sides, and are based on an outside wind exposure of 15 mi. per hr.

Values of C are based on the following internal conductivities or conductances, B.t.u. per hr. per sq. ft. per 1° F.

Brick	5.00 per 1 in.
Cement mortar or concrete	8.00 per 1 in.
Plasterboard	3.73 per 3/8 in.
Plaster (gypsum)	2.32 per 1 in.
Wood lath and plaster	2.00 as applied
Corkboard	0.30 per 1 in.
Rigid insulation (board form)	0.33 per 1 in.
Cellular gypsum (18 lb.)	0.59 per 1 in.
Flaked gypsum, dry (24 lb.)	0.48 per 1 in.

Split furring tile: 1 1/2-in. = 1.40; 2-in. = 1.25
Hollow tile: 6-in. = 0.54; 8-in. = 0.49; 10-in. = 0.46; 12-in. = 0.33
Y = thickness of insulation, in., where specified



No.	Interior Construction	Y	Thickness of Brick, X 8 in.	12 in.	16 in.
	Plain walls—no interior finish		0.385	0.295	0.238
	1/2 in. plaster on brick		.356	.277	.227
	1/2 in. plaster on metal lath, furred		.261	.216	.184
	1/2 in. plaster on wood lath, furred		.250	.208	.178
	1/2 in. plaster on 3/8-in. plasterboard, furred		.251	.209	.179
	1/2 in. plaster on wood lath on 2-in. furring strips, cellular gypsum fill *	1 5/8†	.171	.151	.135
	1/2 in. plaster on wood lath on 2-in. furring strips, flaked gypsum fill *	1 5/8†	.154	.137	.124
	1/2 in. plaster on rigid insulation (board form), furred	1/2	.191	.166	.146
		1	.148	.132	.120
	1/2 in. plaster on corkboard set in 1/2 in. cement mortar		.127	.115	.104
			.105	.097	.090
	1/2 in. plaster on 1 1/2-in. split furring tile set against wall		.284	.231	.195
	1/2 in. plaster on 2-in. split furring tile set against wall		.277	.227	192

* The coefficient used for cellular gypsum, 0.59 is for 18 lb. weight; weights as low as 12 lb. may be used.

† Based on 1 5/8-in., the actual thickness of 2-in. furring strips.

of equal thickness. The B.t.u. loss per foot of sash perimeter (Table 8), is based on the leakage determinations by Voorhees and Meyer, *Trans.* A.S.H.V.E., 1916.

HEAT TRANSMISSION OF ROOFS AND FLOORS.—The temperature of the air in contact with the under side of a ceiling or roof is higher than the temperature at the breathing line, where the temperature usually is measured. This is due to the natural tendency of the warmer or less dense air to rise. It is recommended that an increase of approximately 20% be made to the specified inside temperature for the temperature at the ceiling for ceiling or wall heights exceeding 15 ft. Thus, if 65° F. is the specified inside temperature to be maintained in a room the height of which is 25 ft., the temperature of the air in contact with the under side of the roof may be assumed to be 65° + 20%, or 78° F. The loss of heat through the ceiling of a room over which a large air space exists, through partitions between a heated and a cold room, or through the first floor to the cellar, may be estimated on the assumption that the warmed rooms give off sufficient heat to maintain the temperature of these colder spaces according to the following schedule:

Closed attics under metal or slate roofs.................. 14° F.
Closed attics under tile, cement, tar, or gravel roofs....... 23° F.
Cellars and rooms kept closed......................... 35° F.

The heat transmission of floors that are laid directly upon the ground may be estimated on the assumption that the ground in contact with the under side of the floor has an approximate temperature of 55° F. Thus the estimated heat loss through a 6-in. concrete floor, laid directly on the ground, assuming an inside temperature of 65° F., is

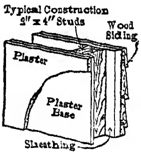
$$u = 1 \div \{(1/1.34) + (6/8)\} = 0.66;\ 0.66\ (65 - 55) = 6.6 \text{ B.t.u. per sq. ft. per hr.}$$

HEAT LOSS BY INFILTRATION.—The heat required to warm the outside air which may enter by leakage through the cracks or clearances around windows and doors is that required to raise the temperature of the weight of incoming air per hour from the outside to the inside temperature.

Table 5.—Coefficients of Transmission (*C*) of Wood Siding on Clapboard Frame Walls

Values of *C* are based on the following internal conductivities or conductances, B.t.u. per hr. per sq. ft. per 1° F.

Wood (yellow pine or fir).................. 1.00 per 1 in.
Plaster (gypsum).......................... 2.32 per 1 in.
Plasterboard..............3.73 per 3/8 in., 2.80 per 1/2 in.
Corkboard................................ 0.30 per 1 in.
Wood lath and plaster..................... 2.00 as applied
Rigid insulation (board form)............... 0.33 per 1 in.
Flexible insulation........................ 0.27 per 1 in.
Cellular gypsum (18 lb.).................... 0.59 per 1 in.
Flaked gypsum (24 lb.)..................... 0.48 per 1 in.



No.	Type of Sheathing.	Insulation between Studding	Wood lath	Metal lath	Plaster Base: 3/8-in. plaster-board	Plaster Base: 1/2-in. rigid insulation	Plaster Base: 1-in. rigid insulation	Plaster Base: 1 1/2-in. cork-board	Plaster Base: 2-in. cork-board
	1-in. wood †	None	0.262	0.275	0.263	0.198	0.153	0.117	0.098
		Flaked gypsum fill (24 lb.) *............	.096	.098	.096	.086	.076	.067	.060
		Cellular gypsum fill (18 lb.)* ‡.........	.111	.113	.111	.097	.085	.073	.065
		1/2-in. flexible insulation...............	.152	.157	.153	.128	.107	.089	.077
	1/2-in. rigid insulation (board form)	None................	.220	.229	.221	.173	.137	.108	.092
		Flaked gypsum fill (24 lb.)*..........	.090	.091	.090	.081	.072	.063	.058
		Cellular gypsum fill (18 lb.)* ‡.........	.102	.104	.103	.090	.080	.069	.062
		1/2-in. flexible insulation................	.137	.140	.137	.117	.099	.083	.073
	1/2-in. plaster-board	None...........	.295	.312	.297	.216	.163	.123	.102
		Flaked gypsum fill (24 lb.)*...........	.099	.102	.100	.089	.079	.068	.062
		Cellular gypsum fill (18 lb.)* ‡..........	.116	.118	.116	.102	.088	.075	.067
12		1/2-in. flexible insulation................	.163	.167	.163	.135	.112	.092	.080

* Thickness of fill assumed 3 5/8 in., based on 2 × 4 in. studding.
† Based on 25/32 in., the actual thickness of 1-in. or 7/8-in. sheathing. Building paper neglected.
‡ The coefficient used for cellular gypsum, 0.59 is for 18 lb. weight; weights as low as 12 lb. may be used.

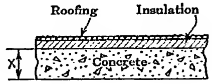
Table 6.—Coefficients of Transmission C of Various Types of Flat Roofs Covered with Built-up Roofing without Ceilings.

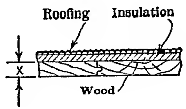
(Underside of Roof Exposed)

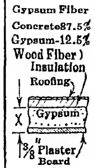
These coefficients are expressed in B.t.u. per hr. per sq. ft. per 1° F. temperature difference between the air on the two sides and are based on an outside wind exposure of 15 mi. per hr.

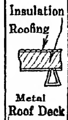
Values of C are based on the following internal conductivities or conductances, B.t.u. per hr. per sq. ft. per 1° F.

Roofing	1.325 per 1 in.
Concrete	8.30 per 1 in.
Gypsum (fiber concrete)	1.66 per 1 in.
Wood (yellow pine or fir)	1.00 per 1 in.
Rigid insulation (board form)	0.33 per 1 in.
Flexible insulation	0.27 per 1 in.
Corkboard	0.30 per 1 in.
Plasterboard	3.73 per 3/8 in.
Cellular gypsum (24 lb.)	0.77 per 1 in.
Lt. wt. heat-treated clay aggregate cement	1.62 per 1 in.









No.	Thickness of Insulation, in.	Kind of Insulation	Concrete Roofs, Thickness, X, in. 1 5/8*	2	3	4	5	6	Wood Roofs, Thickness, X †, in. 1	1 1/2	2	3	4	Gypsum Fiber Concrete Roofs, Thickness, X, in. 2 1/2	3 1/2	Flat Metal Roofs	Corrugated Iron Roofs
1	0	No insulation	0.680	0.658	0.610	0.568	0.532	0.500	0.485	0.386	0.345	0.256	0.204	0.354	0.292	0.781	1.50 ‡
2	1/2	Rigid insulation (board form)	.334	.330	.317	.306	.295	.285	.280	.243	.226	.184	.156	.230	.202	.358	....
3	1		.222	.220	.214	.209	.203	.199	.196	.178	.168	.144	.126	.171	.155	.232	....
4	1 1/2		.166	.165	.162	.159	.156	.153	.151	.140	.134	.118	.106	.136	.126	.172	....
5	2		.133	.132	.130	.128	.126	.124	.123	.116	.112	.100	.091	.112	.106	.136	....
6	1	Corkboard	.208	.206	.201	.196	.192	.188	.185	.169	.160	.138	.121	.163	.148	.217	....
7	1 1/2		.154	.153	.151	.148	.145	.143	.142	.132	.126	.124	.101	.128	.119	.160	....
8	2		.123	.122	.120	.119	.117	.115	.114	.108	.104	.095	.086	.106	.099	.126	....
9	1/2	Flexible insulation (not compressed)	.301	.297	.286	.277	.268	.260	.256	.225	.210	.174	.148	.214	.190	.319	....
10	1		.193	.191	.187	.183	.179	.175	.174	.159	.151	.131	.116	.153	.140	.201	....
11	2	Cellular gypsum	.246	.243	.236	.229	.223	.217	.214	.193	.182	.154	.133	.....	.....	....	....

* Pre-cast cement tile. For heat-treated clay aggregate cement tile covered with roofing, C = 0.55 (1 5/8-in. slab) and C = 0.62 (1-in. slab), based on a conductivity of 3.00 per 1 in.

† Nominal thicknesses specified; actual thicknesses used in computations.

‡ No built-up roofing. The value for corrugated iron is obtained by assuming that the surface area is increased 50%.

Let b = B.t.u. required per hour to heat the incoming air; t = inside room temperature, deg. F.; t_0 = outside temperature; C_p = specific heat of air at constant pressure = 0.24; d = density of the air at temperature t, = 0.075 for 70° inside temperature, = 0.076 for 60° inside temperature; Q = cubic feet of air per hour entering building by infiltration, measured at temperature t; W = weight of air per hour entering building by infiltration = $d \times Q$. Then

$$b = C_p(t - t_0)Q \times d = 0.24 \times W \times (t - t_0)$$
$$= 1.26\,Q \text{ for } 70° \text{ inside temperature and } t_0 = 0.$$
$$= 1.08\,Q \text{ for } 60° \text{ inside temperature and } t_0 = 0.$$

Two assumptions are made by engineers in practice for obtaining the value of Q. The common method is to assume a certain number of air changes, n, per hour in the cubical contents of the room. Table 7 may be used as guide when this method is employed.

Table 7.—Air Changes per Hour

	n		n
Halls	3	Offices and stores, 2d floor	1 1/2 to 2
Rooms on 1st floor	2	Churches, public assembly-rooms	3/4 to 2
Rooms on 2d floor	1	Large rooms with small exposure	1/2 to 1
Offices and stores, 1st floor	2 to 3	Factory buildings	1/2 to 1

Table 8.—Air Infiltration Around Window Sashes

Plain wooden sash	114 cu. ft. air per hr. per ft. perimeter
Plain wooden sash, weather-stripped	24 cu. ft. air per hr. per ft. perimeter
Hollow metal sash	216 to 268 cu. ft. air per hr. per ft. perimeter
Hollow metal sash, weather-stripped	72 to 150 cu. ft. air per hr. per ft. perimeter
Copper-covered sash	132 cu. ft. air per hr. per ft. perimeter

Example.—Required the heat loss, by infiltration, from a room containing 20,000 cu. ft., the temperature of which is maintained at 70° F. in zero weather, the estimated number of air changes n being two per hour.

Solution.—$Q = 2 \times 20{,}000 = 40{,}000$ cu. ft. of air entering per hour at 70° F., $b = 1.26 \times 40{,}000 = 50{,}400$ B.t.u. per hour.

The second method is to use the estimated amount of air leaking in the building through the cracks around the sash perimeter and meeting rail. Table 8 may be used in this connection. The data are based on a wind movement of approximately 20 miles per hour (Voorhees and Meyer tests).

For a room with more than one outside wall use only the sum of the perimeters of the windows, in the side having the greater number.

Example.—An office, 14 by 16 ft. with a 10-ft. ceiling, has two 3 × 7-ft. wooden sash windows. The maintained inside temperature is 70°, and the outside temperature 0° F. Required the heat loss by infiltration.

Solution.—By the first method, assuming two air changes per hour, the loss is

$$b = 1.26 \times 2 \times (14 \times 16 \times 10) = 5645 \text{ B.t.u. per hr.}$$

By the second method this loss is:

$$b = 1.26 \times 2(3 + 3 + 3 + 7 + 7 \text{ perimeter}) \times 114 = 6607 \text{ B.t.u. per hr.}$$

HEAT SUPPLIED BY PERSONS, LIGHTS, MOTORS, MACHINERY, ETC.—See Air Conditioning, p. 11–54.

CALCULATING THE HEAT LOSSES FROM A BUILDING.—It is evident from the foregoing data that the heat transmission calculations for each wall, each roof or ceiling and floor for every room must be performed separately and tabulated for future reference as required for the proper selection of the amount of radiation to be installed, or the amount of warmed air to be introduced into the various rooms.

2. HEATING SYSTEMS

HEATING SYSTEM is generally understood to mean the kind of heating medium and the type of apparatus used to release or transfer heat from the medium to the enclosure to be warmed.

All heating systems comprise at least two principal parts or necessary apparatus to effect the desired result: *a*. **Heat generator** where the heat released from burning fuel is transferred to the heating medium. This apparatus is termed the boiler in steam and hot water systems, or simply a furnace in warm air furnace systems. *b*. **Distribution system** used to convey the medium from the generator to the heat emitting apparatus; for example, steam or hot water piping and air ducts. *c*. **Heat emitting or releasing apparatus**

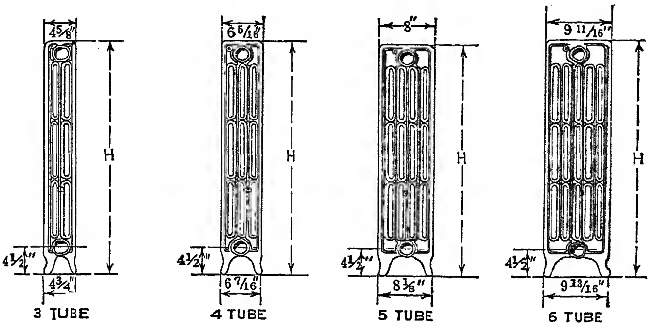


Fig. 1. Cast-Iron Radiators, Tubular Type

to which the distributing system is connected, termed the radiator. In a simple warm air furnace system, only *a* and *b* are required, whereas in a steam or hot water system all three parts are essential.

Radiator commonly means a heat emitting unit located within the enclosure to be warmed. If no provision is made for introducing outside air for ventilation, to be circulated over the radiator, it is termed a direct radiator. Radiators often are installed in recesses in the walls of the building.

The most common type of direct radiator is made of hollow cast-iron sections joined by malleable-iron nipples. Practically only two types of direct cast-iron radiators are now (1935) manufactured: *a.* Tubular type; *b.* Wall type. See Figs. 1 and 2. Several special types of tubular radiators are not illustrated. A variety of direct radiators is built of brass or copper pipe coils, with extended surface in the form of thin metal fins or spiral wound ribbons. Direct radiators are built of steel or iron pipe and standard fittings still are employed to some extent for industrial work. See Fig. 3.

Iron pipe coils, through which hot water is circulated, buried in the wall and ceiling construction of buildings and termed panel heating, are used to some extent in England in place of direct radiators.

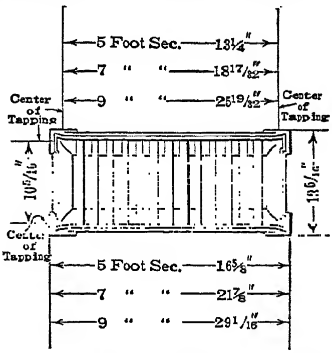


Fig. 2. Cast-iron Radiators, Wall Type

A **UNIT HEATER** is a combination of a radiator and fan located within a common enclosure or casing and placed within or adjacent to the space to be warmed. The unit heater largely used for industrial heating is made in two general types: *a.* Floor type; *b.* Ceiling or hung type. The air passed through the casing is warmed by convection. Comparatively little heat is emitted by radiation. See p. 11–44.

Air for industrial installations almost invariably is recirculated, with no provision for

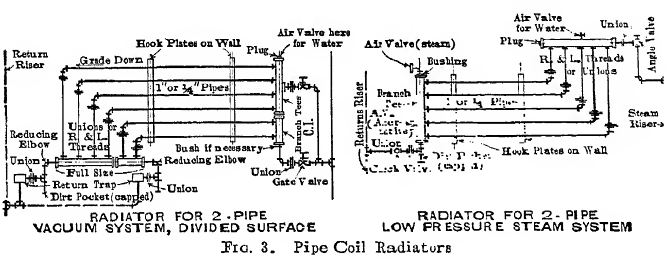


Fig. 3. Pipe Coil Radiators

introduction of outside air to be passed through the unit. Ordinarily, sufficient air for adequate ventilation is provided by infiltration in single-story industrial buildings having monitors or saw tooth roofs. Unit type heaters are now (1935) practically standard heating equipment for industrial establishments. Steam ordinarily is used as the heating medium, although hot water occasionally is used. The circulation of air at a velocity of 1000 ft. per min. or more over a heating surface greatly increases the convection coefficient of the heat emitting surface. This results in a large reduction in installed heating surface, as compared with direct radiation, to obtain the same heating results. The ratio is often as low as 1 to 5.

UNIT VENTILATOR is defined as a floor-type unit heater, designed to circulate all or part outdoor air, with or without recirculation. It is designed primarily for school and office building use where a positive ventilation is required. See Fig. 31. Provision often is made to introduce moisture into the air circulated through the ventilator, and thermostatic control ordinarily is provided. These units may supply both heating and ventilating requirements or ventilation alone in conjunction with direct radiation.

FAN OR BLAST HEATING is understood to mean a combination of heat emitting surfaces, enclosed by a casing through which air is blown or drawn by a fan. As usually installed, the blast heater is located on the suction side and the duct system is connected to the fan outlet. See Figs. 25 to 27 inclusive and Figs. 29 and 30.

RATING OF RADIATORS. EQUIVALENT DIRECT RADIATION.—All types of direct radiators are tested to determine the heat emission, in B.t.u. per hr., of the assembled radiator in still air at 70° F., using dry steam at a temperature of 215° F., corresponding to a pressure of 0.50 lb. per sq. in. gage at the radiator inlet. Manufacturers publish the rating in terms of square feet of *equivalent direct radiation* (e.d.r.) 1 sq. ft. of equivalent direct steam radiation being assumed equal to a heat emission of 240 B.t.u. per hour. No fixed relation exists between the actual measured external heating surface of the many and varied types of radiators, particularly when they are of the extended surface type, and their heat emission; hence the necessity of either publishing radiator ratings in B.t.u. or in terms of e.d.r.

Cast-iron direct radiators, being constructed of sections, the e.d.r. per section is stated. See Tables 9 and 10. When a number of these sections are assembled more or less heat actually may be emitted, depending on height and number of sections. If the same radiator is to be used in conjunction with a hot-water system, based on supplying the radiator with water at 180° F. in still air at 70° F. with a 20° drop in temperature passing through the radiator, the heat emission is assumed to be 150 B.t.u. per sq. ft. of e.d.r. per hour. The temperatures stated for the steam or water and air surrounding the radiator are known as *standard conditions*. Thus, if the published rating of any type of direct radiator, unit heater or blast heater is stated as R sq. ft. of e.d.r., the expected heat emission, when the heat emitting unit is supplied with steam at a temperature of 215° F., will be $R \times 240$ B.t.u. per hour. The latent heat at this pressure is 967 B.t.u. per lb. The weight of steam condensed per sq. ft. of e.d.r. per hr. is, therefore, 240/967, or approximately 0.25 lb. per hr.

AMOUNT OF E.D.R. TO BE INSTALLED.—If H is the calculated heat loss, B.t.u. per hr., the sq. ft. R of e.d.r. to be installed is $R = H/240$ sq. ft. for steam at 0.50 lb. per sq. in. gage in 70° F. air. For a hot-water installation, $R = H/150$ sq. ft. with initial temperature of water of 180° F., and a 20° drop through the radiator.

DIRECT RADIATION FOR CONDITIONS OTHER THAN STANDARD.—Cases arise where both steam temperature and temperature of the room air are other than 215° and 70°, as for example in drying rooms, etc. The coefficient of heat emission of direct radiators, when the steam or room temperature is other than standard, increases with the temperature difference between the radiator and the air. In order to correct for changed conditions, the following formula safely may be applied within the limits of the usual variations met in practice. Let t = temperature of steam or water, deg. F.; t_a = room temperature, deg. F.; R = equivalent direct radiation (e.d.r.) as calculated for standard conditions, sq. ft.; r = equivalent direct radiation (e.d.r.) to be installed, sq. ft. The variation in heat emission from a direct radiator is approximately 0.2% per deg. F. above or below a temperature difference of 145° F. between the radiator and room temperature. Then

$$r = \frac{R}{\{(t - t_a)/145\} + 0.002\,(t - t_a - 145)} \quad . \; . \; . \; . \; . \; . \; . \; [2]$$

RADIATOR DIMENSIONS.—Table 9, with example following, gives the necessary data to determine a tubular type cast-iron radiator, Fig. 1. Table 10 gives corresponding data for wall type radiators, Fig. 2. Manufacturers' catalogs must be consulted for the dimensions of the many and varied types of iron, copper or brass tube extended surface type radiators.

Table 9.—Corto Cast-iron Radiators. (Fig. 1)
(Tubular Type Radiation—Am. Radiator Corp.)

Height of Section H, in.	Square Feet of Equivalent Direct Radiation per Section (240 B.t.u. per sq. ft. of rating). Length of Section = 2 1/2 in. Number of Tubes and Width of Section			
	3–4 5/8 in.	4–6 3/16 in.	5–8 in.	6–9 11/16 in.
20	1 3/4	2 1/4	2 2/3	3
26	2 1/2	2 3/4	3 1/2	4
32	3	3 1/2	4 1/3	5
38	3 1/2	4 1/4	5	6

Table 10.—Dimensions and Heating-surface of Wall Radiators. 2)
(American Radiator Corp.)

Section No.	Length, in.	Width, in.	Thickness, in.	Thickness (with Bracket), in.	Heating-Surface, sq. ft.
5–A	16 5/8	13 5/16	2 7/8	3 1/2	5
7–A, 7–B	21 7/8	13 5/16	2 7/8	3 1/2	7
9–A, 9–B	29 1/16	13 5/16	2 7/8	3 1/2	9

E.—Required the dimensions of a 5-tube, 32-in. high, tubular type, cast-iron direct radiator to supply 14,400 B.t.u. per hr. Standard conditions: steam 0.50 lb. per sq. in., gage; room temperature, 70° F. The e.d.r. rating required is 14,400/240 = 60 sq. ft. The e.d.r. surface for a 5-tube, 32-in. high section is 4 1/3 ft. (Table 9.) The number of sections required is therefore 60/4 1/3, or (approx.) 14. The length of the assembled radiator will be 14 × 2 1/2, or 35 in., and the width will be 8 in. The actual installed e.d.r. is 14 × 4 1/2 = 63 sq. ft. The expected weight of steam to be supplied (or resulting condensate) per hour will be 63 × 0.25 or 15 3/4 lb.

HEAT EMISSION OF DIRECT CAST-IRON RADIATORS.—The coefficient of heat emission K of direct radiation in still air, or the B.t.u. emitted per hour by the radiator per measured square foot of external surface per degree difference in temperature between the heating medium and the surrounding air, is found by tests to be a variable. Its value depends on the length, width, number of sections in the radiator, and the temperature difference. The value of K varies from 1.45 to 1.95, the average value being 1.70. The values may be applied to either steam or hot water.

The coefficient K is increased or decreased at the approximate rate of 0.2% per degree of temperature difference above or below the standard temperature difference of 145° F.

Effect on Heat Emission of Location of Direct Radiators.—The maximum heat emission is obtained when the radiator is placed in the center of the room. The Committee "Report on Best Position for a Radiator in Room," *Trans.* A.S.H.V.E., 1916, states that the following values of K were obtained by test: 1.76 with radiator placed in center of room and 1.588 with radiator placed under a window in the outside wall.

The heat emission of direct radiation should be reduced by the following percentages when inclosures are provided. (See report by Prof. K. Brabbee, Royal Tech. Inst. of Berlin; translated by Geo. Stumpf, Jr., *Heat. & Vent. Mag.*, May, 1914):

Radiator set in recess of wall without front grill, 8%; radiator with shelf over top, 10%; radiator with shelf over top and front grille, 20%.

Effect of Painting on Heat Emission.—Tests by J. R. Allen indicate that the effect of painting the surface of radiators is to influence the loss of heat from the surface only, and depends largely on the radiation factor for the surface in question. The following are the relative transmissions of various surfaces: Bare iron, 1.00; aluminum and copper-bronze, 0.75; snow-white enamel, 1.01; white-lead paint, 0.987; white-zinc paint, 1.01.

3. RATING OF HEATING BOILERS

HEATING BOILER LOAD.—In order to make a satisfactory selection of a heating boiler a number of factors must be considered. As a result, manufacturers have fixed certain standard conditions under which the ratings of their boilers are established.

The *Code of Minimum Requirements for the Heating and Ventilation of Buildings* of the A.S.H.V.E., in discussing the Minimum Capacity and Installation Requirements for Low-Pressure Steam and Hot-water Heating Boilers summarizes the more important factors or conditions as follows:

Estimated Heating Boiler Load.—For the purposes of the Code, the estimated connected load to the boiler or boilers using solid fuels, shall be taken as $B = (a + b + c) \times F$, where $B =$ esti-

mated load connected to the boiler, B.t.u. per hr.; a = estimated heat emission, B.t.u. per hr., of the connected radiation, direct, indirect or both, to be installed as determined by computation from data given for normal operation. (Heat transmission calculations, see p. 11–03); b = estimated maximum heat, B.t.u. per hr., required to supply water heaters or other apparatus to be connected to the boiler or boilers; c = the estimated heat loss, B.t.u. per hr., of the piping connecting radiation and other apparatus with the boiler or boilers (see below); F = estimated increase in the normal load, B.t.u. per hr., due to starting with cold radiation, if boilers are to be in service intermittently. This increase is to be based on $a + b + c$, and assumed to be not less than the following:

Sum of a, b and c in B.t.u. per Hour	Multiplying Factor, F
Up to 100,000	1.65
100,000 to 200,000	1.60
200,000 to 600,000	1.55
600,000 to 1,200,000	1.50
1,200,000 to 1,800,000	1.45
Above 1,800,000	1.40

The values of F are based on heating 1 sq. ft. of cast-iron radiation weighing 7 lb. (specific heat = 0.12) from 35° F. to 220° F. during the first hour of operation, or $7 \times 0.12 \times (220 - 35) = 156$ B.t.u., which is $(156/240) \times 100 = 65\%$ of the heat emission of the radiator.

Heat losses from the piping system, which comprises all connections not considered as direct radiation, and including all mains, branches and risers, may be estimated as follows: Let H = heat loss, B.t.u. per hr. per sq. ft. of external surface of uncovered piping; t_1 = temperature of surrounding air; t = temperature of steam or hot water; $t_2 = (t - t_1)$; K = a constant = 2.0 for steam, and 1.8 for hot water. Then $H = t_2 \times K$. Based on steam and hot-water temperatures of 219.4° F. and 180° F., respectively, the following are values of H for various values of t_1.

t_1	40	45	50	55	60	65	70	75
H (steam)	358.8	348.8	338.8	328.8	318.8	308.8	298.8	288.8
H (hot water)	252	243	234	225	216	207	198	189

If pipe covering $^3/_4$ in., or more, thick is used, the above figures may be reduced 75%. If the covering is less than $^3/_4$ in. thick, the piping should be considered as bare.

Boiler Capacity to be Installed.—The boiler capacity installed should be able to supply, at the boiler outlets, the total B.t.u. per hour as computed by the method outlined in the preceding paragraph, under the conditions of operation which are stated in the specifications covering the installation for which the boilers are intended.

STANDARD CONDITIONS.—American manufacturers of heating-boilers rate the boilers in terms of B.t.u. per hr. or sq. ft. of direct cast-iron radiating surface (e.d.r.) that the boiler is capable of supplying, for the firing period stated (fuel available in hours), under the following conditions: 1. *Steam boilers,* steam pressure, 2.3 lb. gage at the boiler; 2. *Hot-water boilers,* water temperatures, 180° F. leaving and 160° F. entering the boiler; 3. *Fuel,* stove size anthracite, of a heating value of at least 12,000 B.t.u. per lb. of dry coal.

The rate of combustion, or coal required per hour for the boiler to develop this rating, has, until recently, seldom been given. The method of determining rating has varied with different makers and often is not stated. Also, it is possible for a boiler to be placed on the market and given a certain rating, although such rating never actually has been checked by tests. It is, therefore, important to establish standard conditions for rating tests, and the manufacturer should be in a position to produce certified test sheets of such tests. The standard conditions under which a boiler should be tested to develop its rating are generally understood by manufacturers to be as follows:

1. Pressures, temperature and fuel as stated above. 2. Fuel capacity sufficient to carry the boiler for the period stated, and leave 20% reserve for igniting a fresh charge. 3. Draft of sufficient intensity to burn the fuel at the required rate to produce the rating stated. 4. Each square foot of equivalent direct radiation (e.d.r.) is assumed to transmit 240 B.t.u. per hr. for steam, and 150 B.t.u. per hr. for hot-water radiators, respectively. 5. The condensation from steam radiators returns to the boilers at the same temperature as the steam, so that the boiler simply supplies the latent heat of evaporation at 2.3 lb. gage pressure, or 965.4 B.t.u. per lb. of water evaporated. 6. The water from hot-water radiators returns to the boiler at 160° F., allowing a 20° drop in the radiators. No loss of temperature is allowed in the return main.

EQUIVALENT BOILER HORSEPOWER RATING OF HEATING BOILERS.—The capacities of heating-boilers may be stated in boiler horsepower, and its equivalent in square feet of radiation may be determined as follows: 1 boiler Hp. = 34.5 lb. water evaporated from and at 212° F. = 34.5×970.2 (latent heat at 212° F.) = 33,472 B.t.u. per hr. 1 sq. ft. of standard cast-iron steam radiation is assumed to transmit 240 B.t.u. per hr., and 1 boiler Hp. = $33{,}472 \div 240 = 139.5$ sq. ft. of equivalent direct radiation.

The equivalent boiler Hp. rating of a hot-water boiler is $33{,}472 \div 150 = 223.1$ sq. ft. of direct cast-iron hot-water radiation.

GRATE-SURFACE.—It is advisable to check the grate-area required for heating-boilers. The total heat loss must include not only the calculated loss, due to transmission through walls and glass for which the radiation is proportioned, but also about 50% additional for heat losses from the piping system, boiler, etc. If B is the B.t.u. to be supplied by the boiler, $G = B \div (C \times F \times E)$, where G = grate-area, sq. ft.; C = rate of combustion of dry coal, lb. per hr.; F = calorific value of fuel, B.t.u. per lb.; E = combined efficiency of boiler and grate (usual assumption, 65%). The boiler selected should not have a smaller grate-area than that given by the above formula. Special attention is called to the distinction between grate-area and firebox or fuel-pot area. (See Depth of Fuel-pot, below.) The rate of combustion C varies from approximately 5 lb. for small boilers up to approximately 10 lb. for the largest sizes. Table 11 gives the maximum rate of combustion during the starting up period, and the rate for the normal load as ordinarily found in practice.

Table 11.—Rates of Combustion for Heating Boilers

Based on 13,000 B.t.u. Anthracite, and on Boiler and Grate Efficiency of 65%

Estimated Normal Boiler Load, Sum of Items, $a+b+c$ (see p. 11-03)		Maximum rate of Combustion for Starting Up Load, C_m lb. per sq. ft. of grate per hr.†	Rate of Combustion of Normal Load, C, lb. per sq. ft. of grate per hr.‡	Factor for Starting Up Load, $F=C_m/C$
B.t.u. per hr.	Equivalent Direct Steam Radiation, sq. ft.*			
Up to 100,000	Up to 420	8	4.85	1.65
100,000 to 200,000	420 to 840	8	5.00	1.60
200,000 to 600,000	840 to 2,500	9	5.80	1.55
600,000 to 1,200,000	2,500 to 5,000	10	6.70	1.50
1,200,000 to 1,800,000	5,000 to 7,500	11	7.50	1.45
1,800,000 to 2,880,000	7,500 to 12,000	12	8.60	1.40
2,880,000 to 3,840,000	12,000 to 16,000	14	10.00	1.40

*B.t.u. per hour/240. † $C_m = (a \times F \div (G \times 0.65 \times 13{,}000)$.
‡ $C = (a+b+c) \div (G \times 0.65 \times 13{,}000)$.

EXAMPLE.—The estimated boiler load (sum of items $a + b + c$) is 150,000 B.t.u. per hr. The maximum starting load is assumed as $150{,}000 \times 1.6 = 240{,}000$ B.t.u. per hr., or 1000 sq. ft. of e.d.r. *Required*: The grate area with 13,000 B.t.u. anthracite, and an assumed boiler efficiency of 65%. From Table 11, assume a combustion rate of 8 lb. per sq. ft. of grate per hr. for the maximum starting up load. Then $G = 240{,}000/(8 \times 0.65 \times 13{,}000) = 3.55$ sq. ft. The rate of combustion at normal load will be about 5 lb. per sq. ft. of grate per hr.

DEPTH OF FUEL-POT.—The average area of the firebox is usually somewhat larger than the grate-area in sectional boilers. It may be less than the grate-area in certain types of round boilers. In any event, the capacity of the firebox or fuel-pot, measured from grate to middle of firedoor, should be sufficient to hold all the coal required for an 8-hr. firing period, plus at least 20% reserve for igniting the new charge.

To determine the depth of pot or firing period, as the case may be, let G = grate-area, sq. ft.; C = rate of combustion, lb. per hr.; A = average area of fire-pot, sq. ft.; h = firing period, hr.; W = weight of fuel per cu. ft. (50 lb. for anthracite, 40 lb. for bituminous); D = depth of fuel bed from grate to center of firedoor, ft. Then $(G \times C \times h) + 20\%$ or $A \times W \times D$ = total weight of one charge. Hence, $D = 1.2\, GCh/AW$, or $h = AWD/1.2\, GC$. This formula allows for the greater bulk of soft coal.

EXAMPLE.—In a boiler with a grate-area of 8 sq. ft., average firepot area, 9 sq. ft., height to center of firedoor, 18 in., and a rate of combustion of 6 lb. per sq. ft. of grate for anthracite, $h = (9 \times 50 \times 1.5) \div (1.2 \times 8 \times 6) = 11.7$ hr.

MANUFACTURERS' RATING OF STEEL HEATING BOILERS.—The Steel Heating Boiler Institute adopted (1929) a rating code for steel heating boilers based on grate-area as follows: For boilers with ratings of 300 sq. ft. to 4000 sq. ft. of e.d.r. for steam, grate area, sq. ft. = $\sqrt{(\text{catalog rating, sq. ft. steam radiation} - 200) \div 25.5}$. For boilers with ratings of 4000 sq. ft. of steam radiation and larger, grate-area, sq. ft. = $\sqrt{(\text{catalog rating, sq. ft.} - 1500) \div 16.8}$. The code further states that the boiler heating surface must not be less than $^1/_{14}$ of the e.d.r. rating in square feet. Catalogs of steel boilers give both grate area and boiler heating surface. The heating surface ratio is based on 10 sq. ft. per nominal boiler Hp. One boiler Hp., as previously shown, will supply 139.5 sq. ft. of e.d.r. Therefore, $10/139.5 = {}^1/_{14}$ (approx.).

EFFECT OF FUELS ON RATINGS.—All ratings are based on anthracite, unless otherwise stated. If bituminous coal is used and the boiler is selected by catalog rating, one with a fire-pot of at least 25% greater capacity should be selected. With soft coal, additional heating-surface also is required, as soot accumulations render the heating-

surfaces less effective. Boilers for pea coal should have a larger fire-pot than those for stove or furnace coal. As the small sizes of anthracite contain more ash than the larger sizes, they have greater bulk for the same heating effect and so require larger fuel-pots for the same capacity. Firing periods differing from that on which the boiler is rated, also affect the fuel-holding capacity. For example, if boilers designed for an 8-hr. period are operated on a 12-hr. basis, at least 50% greater fuel-holding capacity will be necessary and larger boilers must be selected.

FUEL CONSUMPTION.—The estimated fuel consumption of heating-boilers per heating season may be based on grate-area, square feet of radiation installed, or cubic contents of the building to be heated. The U. S. Treasury Dept. allows for government buildings 5 tons of coal per sq. ft. of grate-area per season of 240 days, or 1 lb. of coal per cu. ft. of building content. District steam-heating companies estimate 500 lb. of steam per sq. ft. of direct radiation per season or about 70 lb. of good coal. This is approximately equivalent to assuming that $^1/_3$ of the radiation installed is operated continuously for 240 days. The amount of coal for maximum conditions is determined as follows:

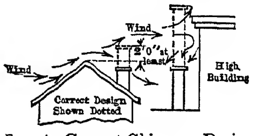


Fig. 4. Correct Chimney Design

Assume 1 sq. ft. of direct steam radiation to emit 240 B.t.u. per hr., 70° air surrounding the radiator, and assume the piping on the average job to be 25% of the direct radiation. Assume 8000 B.t.u. per lb. of anthracite burned. If R = direct radiation, sq. ft., and C = coal burned per hr., lb., $C = (1.25 \times R \times 240) \div 8000$. In a heating season from Oct. 1 to May 1, 210 days of 24 hr. each, there would be burned under maximum conditions, a total of

$$(1.25 \times R \times 240 \times 210 \times 24) \div (8000 \times 2000) = 0.0945\,R \text{ tons of coal,}$$

and for hot-water heating, 0.0197 R tons.

The estimated seasonal consumption is the product of the preceding figure and the "demand factor," or $(70 - T_a)/T_d$, where T_a = average outside temperature (Table 1), and T_d = temperature for which the system was designed, usually 70° F.

Reference.—Engg. Economics of Heating, M. W. Ehrlich, *Trans.* A. S. H. V. E., 1919.

Table 12.—Average Steam Consumption of Various Type Buildings, per Year
(Indianapolis, Ind.)

Buildings	No.	Pound Steam per sq. ft. Radiation	Buildings	No.	Pound Steam per sq. ft. Radiation
Hotels	11	482	Public garages	3	280
Apartments	13	562	Manufacturing	4	447
Department stores	6	315	Public buildings	4	292
Theaters	5	364	Clubs, lodges	3	500
Office buildings	20	354	Printing	5	655
Auto sales	8	498	Warehouses	2	590

Average of above is 400 lb. per sq. ft. per year.

TYPES OF HEATING BOILERS.—Cast-iron steam-heating boilers are designed to operate at a maximum pressure of 15 lb. per sq. in. The sections are tested in manufacture to about 100 lb. per sq. in. hydrostatic pressure. The maximum size of round-type boilers manufactured is rated at about 1400 sq. ft. Sectional boilers are built up to 10,000 sq. ft. rating (see manufacturers' catalogs for dimensions, capacities, etc.).

SELECTION OF CAST-IRON BOILERS.—The selection of cast-iron boilers should not be influenced too largely by price and the ease of carrying them into a building, where structural conditions interfere with the introduction of a steel boiler. The character of the service or attendance, especially in government or other public buildings, often is such that steel equipment, capable of standing more abuse, should be used, particularly if the returns are handled by a pump.

STEEL HEATING BOILERS.—The two general types of all-steel boilers used for heating are the firebox type and the return tubular type. In the firebox type, the grate and combustion chamber are surrounded by a water-jacketed extension of the shell. The products of combustion pass directly through the tubes to the smoke flue at the rear. The return tubular type is similar to the ordinary horizontal tubular boiler used for power plants (see Steam Boilers).

Capacities of the usual firebox type boilers range from 500 to 13,000 sq. ft. of direct radiation. Detailed information as to capacities, dimensions, etc., are available in makers' catalogs. The return tubular boiler commonly is used in heating systems of 10,000 sq. ft. or more of direct radiation, and is rated at 10 sq. ft. of boiler heating-surface per boiler Hp. A special design of setting is required for smokeless combustion when bituminous

coal is used. The so-called standard setting should not be used in this connection. (See Steam Boilers.)

CHIMNEYS FOR HEATING BOILERS.—The minimum height of chimneys for low-pressure heating boilers, hot-water boilers, and hot-air furnaces is 35 ft. measured from the grate. No flue should be less than 8 × 8 in. Many heating installation failures may be traced to insufficient draft to burn the fuel at the rate required for the rated capacity of boiler or furnace. Flue-gas temperatures should range between 400° and 500° F. when the apparatus is worked at its rated capacity. The chimneys should be so located with reference to nearby higher buildings that wind currents will not form eddies and force air downward in the shaft. (See Fig. 4.) The flue should be as straight as possible from

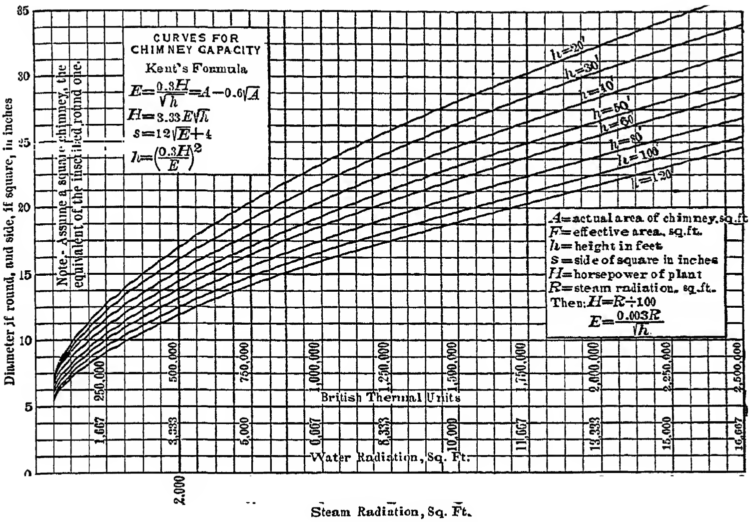


Fig. 5. Capacity of Chimneys

base to top outlet and should have no opening except the boiler smoke pipe. The outlet must not be so capped that its area is less than the flue area. Sharp bends and off-sets in the flue may reduce the area and choke the draft. The flue must have no feature which reduces the full area. In tile flues, joints must be well cemented and all space between tile and brickwork tightly filled in. If crevices open into the flue where tile sections meet, the draft will be checked. With brick flues, the stacks should have outside walls at least 8 in. thick. Exposed bricks at the top should be laid in cement mortar to prevent acid

Table 13.—Fireclay Flue-linings

(Robinson Clay Products Co., Akron, Ohio)

Rectangular			Round	
Nominal Size, in.	Actual Size, Outside, in.	Actual Size, Inside, in.	Inside Diam., in.	Outside Diam., in.
4 1/2 × 8 1/2	4 3/4 × 8 5/8	3 1/4 × 7	6	7 1/2
4 1/2 × 13	4 3/4 × 13 1/4	3 1/16 × 11 3/4	7	8 1/2
4 1/2 × 18	4 1/2 × 17	3 7/8 × 15 1/2	8	9
6 × 16	6 × 12	4 1/2 × 10 1/2	9	10 1/2
7 × 7	7 1/4 × 7 1/4	5 3/4 × 5 3/4	10	12
8 1/2 × 8 1/2	8 1/2 × 8 1/2	7 1/4 × 7 1/4	12	14
8 1/2 × 13	8 1/2 × 13	6 7/8 × 11 5/8	15	17 1/8
8 1/2 × 18	8 1/2 × 18	6 1/2 × 16	18	20 7/8
13 × 13	13 × 13	11 1/4 × 11 1/4	20	23
13 × 18	13 × 18	10 3/4 × 15 3/4	24	27
18 × 18	18 × 18	15 1/2 × 15 1/2	30	35

fumes and rains from cutting out the joints, as will occur with lime mortar. The best location for a chimney is near the center of the building, as all walls then are kept warm. If there is a soot pocket in the flue below the smoke-pipe opening, the clean-out door always should be tightly closed. Other openings into it, from fire-places, etc., check the draft and prevent best results. The smoke-pipe should not extend into the flue beyond the inside surface of the latter, as its end cuts down the area of the flue. Joints where the smoke-pipe fits the smoke-hood of the boiler, or where the pipe enters the chimney, should be made tight with boiler putty or asbestos cement. The best practice uses fire-clay linings for small and medium-sized flues. Rectangular flue linings are rated by outside dimensions and round linings by inside dimensions.

Reference.—An Ordinance for Construction of Chimneys, recommended by Nat. Board of Fire Underwriters, *Jour.* A. S. H. V. E., Dec. 1921.

Table 14, giving dimensions and heights of chimneys, has been used with success in many heating installations. For large installations and for power boilers, draft losses should be estimated and a height of chimney chosen to give sufficient intensity of draft to balance the sum of the losses. See William Kent's Chimney Formula, p. 6–104 and Fig. 5.

The loss of draft through a cast-iron sectional boiler, normal rating, is approximately 0.15 in. water column, and 0.07 in. for loss through fuel bed, with a combustion rate of 8.6 lb. of egg coal per hr.

Stacks for Tall Buildings are special cases and should be designed by the methods used in the design of chimneys for power boilers. See Chimneys and Draft, p. 6–102.

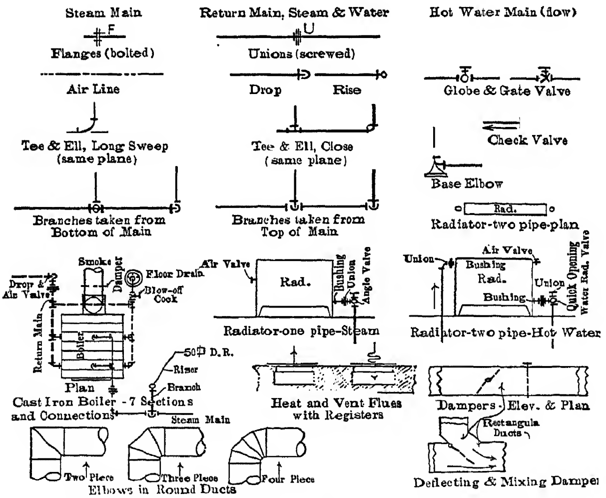


FIG. 6. Symbols Used for Heating and Ventilating Apparatus

4. DIRECT STEAM HEATING

Systems using direct steam radiators are: 1. Gravity circulating. 2. Mechanical circulating. The distinguishing characteristic is the manner in which the condensate from the radiators returns to the boiler. In Type 1 the condensate returns by gravity, due to the static head existing in the returns, and the system is a closed circuit. The steam pressure is the same in boiler, mains and radiator, except for friction-pressure losses due to the flow of steam. In Type 2, the condensate returns to a receiver or feedwater heater, and then is forced into the boiler by a pump or return traps, or both. The system is not closed, and boiler-pressure may be higher than that in mains and radiators. The receiver usually is vented to atmosphere, and in vacuum systems an additional pump, attached directly to the returns, discharges the condensate into the receiver or heater. Gravity circulating systems are also divided into 1-pipe and 2-pipe systems, with basement mains supplying

Table 14.—Dimensions of Chimneys for Low-pressure Steam- and Hot-water Boilers and Hot-air Furnaces

(American Society of Heating and Ventilating Engineers)

Warm Air Furnace Capacity, sq. in. of Leader Pipe	Steam Boiler Capacity, sq. ft. of Radiation	Hot-Water Heater Capacity, sq. ft. of Radiation	Rectangular Flues			Round Flues		Height of Chimney from Grate, ft.
			Nominal Dimensions of Fire-clay Lining, in.	Actual Inside Dimensions of Fire-clay Linings, in.	Actual Area, sq. in.	Inside Diameter of Lining, in.	Actual and Effective Area, sq. in.	
590	590	973	8 1/2 × 13	7 × 11 1/2	81	......	......	35
1000	690	1,140	......	......	......	10	79	35
	900	1,490	13 × 13	11 1/4 × 11 1/4	127	......	......	35
	900	1,490	8 1/2 × 18	6 3/4 × 16 1/4	110	......	......	35
	1,100	1,820	......	......	......	12	113	40
	1,700	2,800	13 × 18	11 1/4 × 16 1/4	183	......	......	40
	1,940	3,200	......	......	......	15	177	40
	2,130	3,520	18 × 18	15 3/4 × 15 3/4	248	......	......	40
	2,480	4,090	20 × 20	17 1/4 × 17 1/4	298	......	......	45
	3,150	5,200	......	......	......	18	254	50
	4,300	7,100	......	......	......	20	314	50
	5,000	8,250	24 × 24	21 × 21	441	......	......	55
	4,600	7,590	......	20 × 24	480	......	......	55
	5,570	9,190	......	24 × 24 *	576	......	......	60
	5,580	9,200	......	......	......	22	380	60
	6,980	11,500	......	......	......	24	452	65
	7,270	12,000	......	24 × 28	672	......	......	65
	8,700	14,400	......	28 × 28	784	......	......	65
	9,380	15,500	......	......	......	27	573	65
	10,150	16,750	......	30 × 30	900	......	......	65
	10,470	17,250	......	28 × 32	896	......	......	65
	11,800	19,500	......	......	......	30	707	70
	14,700	24,300	......	......	......	33	855	70
	17,900	29,500	......	......	......	36	1018	70

* Dimensions below are larger than those in which rectangular fire-clay flue linings are commercially available, and hence are for unlined rectangular flues—requiring thicker walls than when lined.

risers to the various floors, or with overhead mains supplying drop risers to the floors below. In the latter arrangement, steam and the water of condensation in the risers flow in the same direction. As there are no counter-currents, less friction is produced, and somewhat smaller pipes may be used. The overhead system commonly is known as the Mills system. See p. 11–18.

The following types of steam heating systems are in common use: One-pipe circuit systems, Fig. 7; One-pipe Relief systems, Figs. 8 and 9; Two-pipe systems, Fig. 10; Air-line systems, Fig. 11; Vapor or Air Return systems (2-pipe), Fig. 12; Vacuum systems, Fig. 13. In all systems provision must be made to maintain the water in the boiler at the normal water-line level. A most prolific cause of cracking of sections in a cast-iron boiler is the lowering of the water line, thereby uncovering heating surface which is practically in contact with the fire. Due to loss of pressure in a gravity return system caused by frictional resistance in piping, valves, etc., a static head of water must exist in the return piping, above the boiler water line, equivalent to this pressure loss (30 in. per lb. loss in pressure). When the system is started with cold radiation, a greater volume of steam is moved through the piping. Consequently, greater loss in pressure results, and more water is drawn from the boiler than is necessary during the normal heating period to create the necessary static head in the return piping. It is during this starting period that cracking of cast-iron boiler sections sometimes occurs.

ONE-PIPE GRAVITY SYSTEM.—The 1-pipe *circuit system* (Fig. 7) with basement mains commonly is used for small residence heating. The main rises close to the basement ceiling, just above the boiler, grading down from this high point with a fall of 3/4 or 1 in. per 10 ft. to the last radiator riser. The main then drops below the boiler water-line, and now being required to carry only condensation, is reduced in size. This construction is called *wet return,* and is the most satisfactory arrangement whenever its use is possible. If the return main is above the boiler water line, it is a *dry return.* Return mains slope to the boiler 1 in. per 30 ft. An automatic air valve should be placed on the main, at the drop, to remove air from the pipe system.

In mains of unusual length the height of the end of the main above the boiler water line must be carefully determined to prevent water backing up from the boiler and flooding the main, air valve and branches. For steam mains up to 80 ft. long, there should be

at least 20 in. between the under side of the steam main at the low point and the normal water level in the boiler. This height should be increased 2 in. for every 10 ft. of run above 80 ft. in all types of gravity systems. In operation, steam and water flow in the same direction in the steam main, and in opposite directions in basement branches, risers and radiator branches. This necessitates larger piping and valves than in any other steam system. The main, especially, must be full size from boiler to drop, unless dripped at intervals.

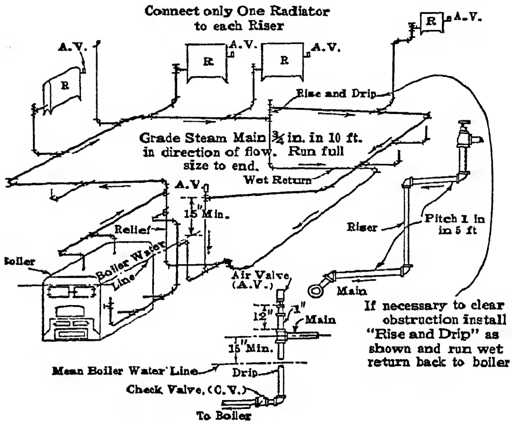


FIG. 7. One-pipe Circuit System

THE ONE-PIPE RELIEF SYSTEM, Fig. 8, resembles the 1-pipe circuit system, except that the individual risers drip to the return main, which may be either wet or dry. A wet return is preferred. The steam main carries no condensate, and also drips at intervals to the return. A *rise and drip*, as shown, is used when the head room under the steam main would be too much reduced. In this system it is possible to reduce the size of the main at each branch, and to run the main closer to the basement ceiling, which is important where basement space is valuable. This is the system most commonly used in large installations. For tall buildings, the 1-pipe system with basement mains and gravity circulation frequently is used. It is satisfactory if the piping is properly designed for the circulation of steam and return of condensate. In long, narrow buildings, using a gravity system, a deep boiler pit is necessary, otherwise the elevation of water in the return connections may flood the far end of the steam lines.

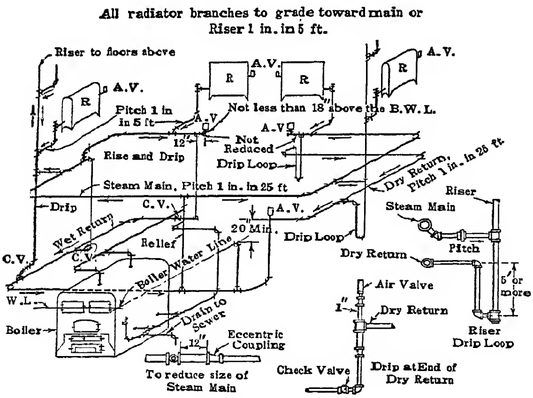
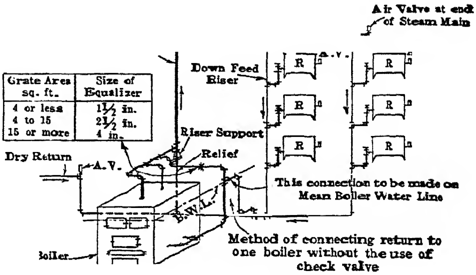


FIG. 8. One-pipe Relief System

MILLS SYSTEM.—A more satisfactory arrangement, Fig. 9, for tall buildings and factories is to run the steam main near the ceiling of the top floor or the roof and install down-feed risers to the radiation. This arrangement of the piping is known as the Mills System. In it, the steam and condensate flow in the risers in the same direction and with higher velocities. Consequently smaller pipe may be used than with an up-feed system. The risers drip at the bottom to the return as previously indicated. These systems ordinarily operate at 2 to 5 lb. boiler pressure at normal load. The steam piping usually is designed for a loss in pressure of approximately 1 oz. per 100 ft. of run, including allowances for ells and other fittings.

THE TWO-PIPE GRAVITY SYSTEM, Fig. 10, with basement mains often is used in large buildings, and *always* with indirect radiation. A thermostatic valve on each radiator will adapt it to vapor and mechanical vacuum systems. When used as a gravity

system, the return from each radiator is separately sealed either by dropping below the boiler water line to a wet return or by using drip loops, before connection to a dry return. Even in 1-pipe systems, all drips or reliefs should be sealed as in Fig. 8. If this is not done, steam may enter a drip or return from the outlet and cause water-hammer, due to counter-currents of steam and condensate. All drips, reliefs, return-risers and connections from the steam to the return side of the system must be sealed, either by connection below the water line or by using a running or return-trap on the connecting line. Failure so to seal will result in unsatisfactory operation.



Grate Area sq. ft.	Size of Equalizer
4 or less	$1\frac{1}{2}$ in.
4 to 15	$2\frac{1}{2}$ in.
15 or more	4 in.

FIG. 9. Mills System

SPECIAL GRAVITY SYSTEMS.—Many special steam-heating systems, known as air-line and vapor systems, also operate with gravity return of the condensate. The air-line system may be applied to any 1- or 2-pipe gravity system, by connecting the automatic air valve of each radiator, by small size piping, to an exhauster maintaining a slight vacuum in the air piping and removing accumulated air from the radiators. The application of this scheme to the ordinary 1-pipe or 2-pipe gravity system, will improve its operation. The exhauster for less than 2500 sq. ft. of radiation is a water-driven vacuum-pump with a pressure of at least 20 lb. per sq. in. Larger systems use a high-pressure steam jet, or if steam is not available, a motor-driven vacuum pump of about $^1/_4$ Hp.; 1-in. air mains in the basement are used, with a gate valve on each riser. See Fig. 11. The steam used varies from 1 to 5% of the total condensation.

The Vapor Systems, so-called, Fig. 12, are 2-pipe gravity systems in which the accumulated air in the radiators is removed through the return, the air valve on the radiator being omitted. The return on each radiator has a check valve or thermostatic trap, and the dry main return in the basement terminates in a small receiver, having an automatic air valve of sufficient capacity to remove all accumulated air. Each radiator ordinarily is fitted with a graduated fractional valve on the steam connection, permitting partial heating of the radiator when desired.

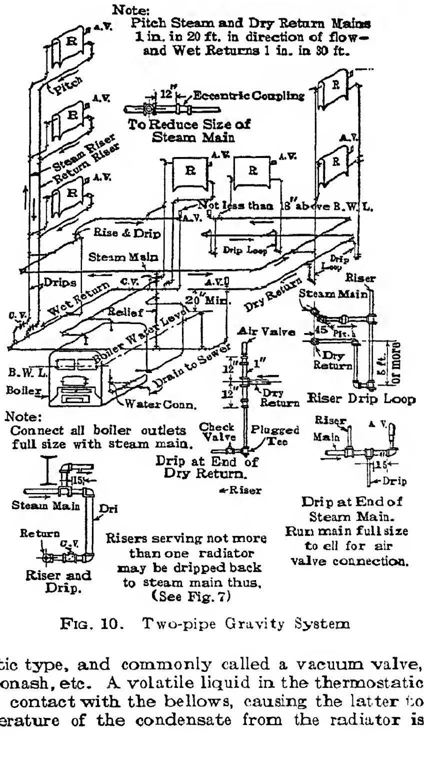
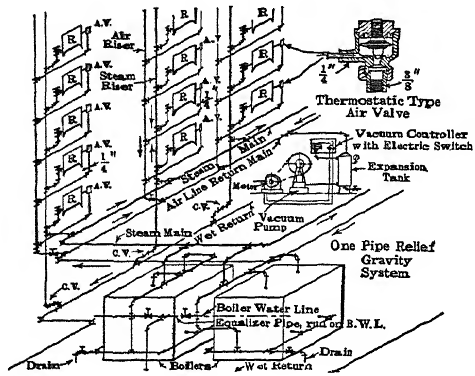


FIG. 10. Two-pipe Gravity System

MECHANICAL VACUUM SYSTEMS are of the 2-pipe type and have a vacuum pump attached directly to the returns. See Fig. 13. This pump must be capable of handling both air and water, as no air valves can be used on the radiators in this system. The return end of each radiator has a radiator trap, usually of the thermostatic type, and commonly called a vacuum valve, such as the Dunham, Webster, Illinois, Monash, etc. A volatile liquid in the thermostatic bellows vaporizes when steam comes in contact with the bellows, causing the latter to expand and close the valve. The temperature of the condensate from the radiator is



Note:
All Connections to radiator air Valves $\frac{1}{4}$ in.
Air risers are $\frac{1}{2}$ in., up to 8 stories high.
Air mains in basement are: 1 in. for $\frac{1}{2}$ in. Risers and $1\frac{1}{4}$ - $1\frac{1}{2}$ in. for $\frac{3}{4}$ -in. Risers.
Air Valves are of the thermostatic type.
This illustration shows the recommended method of connecting piping for a two-boiler installation without the use of check valves.

Fig. 11. Air-line Vacuum System

Note: Pitch Steam and Dry Return Mains 1 in. in 20 ft. in direction of flow. All Radiator Branches to grade toward Main or Riser 1 in. in 5 ft.

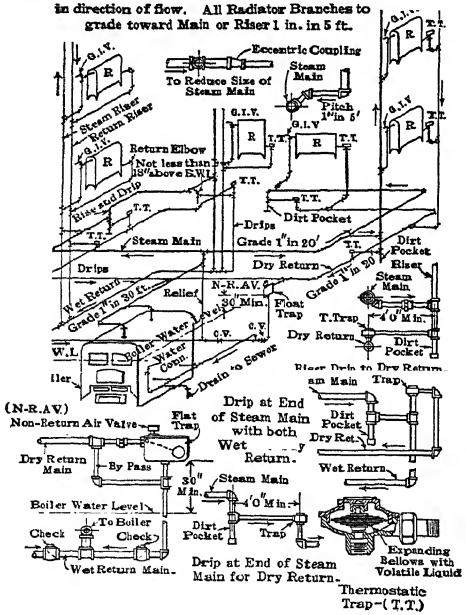


Fig. 12. Vapor System

not sufficiently high to vaporize the liquid, and the valve, therefore, remains open to pass condensate and air until the steam starts to flow. The valves are very sensitive, and when in proper adjustment and repair will not blow steam. It is good practice to connect a $^1/_2$-in. cold water line to the main return at the pump, to condense steam that may leak past the vacuum valves, due to dirt getting under the seat.

Figs. 14*A* and 14*B* show the application of vacuum traps to the 2-pipe system. The vacuum valve or trap is placed 4 ft. from the riser or main which it drains. Otherwise, conduction of heat through the connection to the trap will keep the valve open. Return connections for a vacuum system are smaller than for the ordinary 2-pipe system.

Vacuum systems are used with exhaust steam heating, where the back pressure from engines or turbines ordinarily should not exceed 5 lb. per sq. in. A by-pass with a reducing valve cross connects the live-steam main with the heating system, allowing live steam at reduced pressure (usually 2 to 5 lb.) automatically to enter the heating system whenever the demand is greater than the supply from the engines, or when they are not operating (see Fig. 18). The pump on the main return-line ordinarily maintains a vacuum of about 10 in. of mercury. It is under automatic control, the negative pressure in the return line operating the controller.

RADIATOR VALVES.—Table 15, giving radiator valve ratings, is based on average cast-iron radiation for a 20-minute heating-up period. If the 20-min. quick heating-up feature is disregarded and ratings are desired for normal requirements after the radiator is fully heated, multiply values in Table 15 by 2. The capacity and rating of various types of fractional valves as manufactured by various concerns will naturally vary somewhat.

PRESSURE LOSSES IN PIPING.—The pressure loss due to friction in steam mains, valves and traps in low-pressure gravity systems, ordinarily should not exceed, approximately, 1 oz. or 0.062 lb. per sq. in. per 100 ft. of run. Fig. 15 shows the reason for limiting the pressure loss. As the steam flowing through the main loses pressure, the pressure at the last riser will be lower than in the boiler. The difference in pressure, or pressure loss, p, causes the water line in the return main to be higher than in the boiler. The added height of water in the return is that of a column of water which pressure p will support. Thus, if boiler-pressure is 2 lb. per sq. in., and the pressure at the far end of the main is $1\,^1/_2$ lb., with water weighing 61 lb. per cu. ft. (0.035 lb. per cu. in.), the water in the return will stand $(2 - 1.50) \div 0.035$ or 14 in. above the water-line of the boiler. In this instance, unless the water-line of the boiler is 18 in. or more below the last riser or radiator-connection, water is liable to flood the steam-main and cause hammering and poor circulation in the radiators near the end of the run.

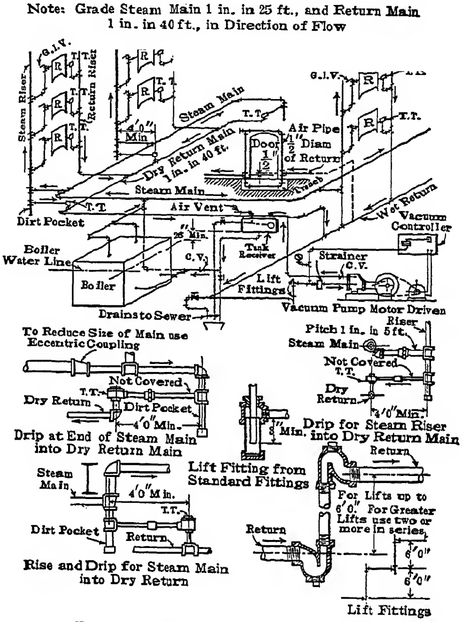


FIG. 13. Mechanical Vacuum System

Fig. 15, showing the various pressure losses existing in a vapor system, illustrates the method of approximately estimating pressure losses in any low-pressure heating system. The height of the water line in the return piping above the boiler water line then may be determined. The example is based on normal or designed load conditions of operation. The pressure losses are calculated as follows:

Let P_1 = pressure loss, or effective pressure, required to impart initial velocity in the steam main and overcome friction at entrance to main, lb. per sq. in.; H = head of water column, ft., $= P/0.415$; then $P_1 = 1.65 \times$ velocity pressure $= 1.65\, v^2d/9262$; v = velocity of steam, ft. per sec.; d = density of steam. (See Steam Tables, p. 5–04.) The velocity of the steam in the main at

Table 15.—Rating of Radiator Valves When Full Open

Square Feet Equivalent Direct Radiation
(Warren Webster Co., Camden, N. J.)

Type of Valve	Size, in.	Pressure at Inlet Valve 2 oz.	4 oz.	8 oz.	1 lb.
Standard angle...	1/2	30	42	60	84
" " ...	3/4	62	87	124	175
" " ...	1	102	147	204	294
" " ...	1 1/4	180	252	360	504
" " ...	1 1/2	258	364	516	728
Modulating *....	1/2	27	38	54	76
"	3/4	57	80	113	160
"	1	94	132	187	265
"	1 1/4	160	225	319	450

* Fractional or graduated inlet type. Use 2-oz. rating for vapor and 8-oz. rating for vacuum systems.

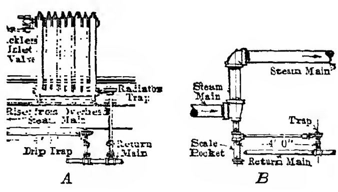
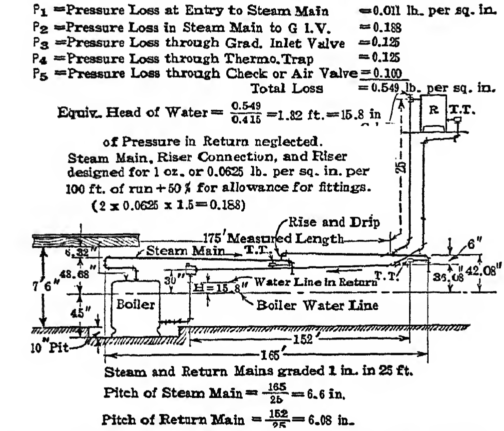


FIG. 14. Application of Vacuum Traps to 2-Pipe System. A. Riser Drip. B. Riser and Drip Connection

the boiler, as ordinarily designed, usually does not exceed 40 to 50 ft. per sec. at rated load. Assuming 1 lb. gage pressure at the boiler, d = 0.04, and

$$P_1 = 1.65\,d(40^2 \times 0.04/9262) = 0.011 \text{ lb. per sq. in.}$$

P_2 = pressure loss due to friction in steam main, horizontal connection to riser and riser to inlet valve of radiator farthest from the boiler, lb. per sq. in., taken as 1 oz. or 0.0625 lb. per sq. in. per 100 ft. of run, including all allowances for loss through ells and tees. (See Figs. 16 and 17.) If no actual estimate for ells or tees is made, 1.5 L is usually sufficiently accurate, L being the measured length, ft., from boiler outlet to last radiator valve.



Note: In the case of Long Steam Mains it sometimes becomes necessary to install "Rise and Drip", as indicated, to keep end of Steam Main high enough to Drip into the Dry Return Main at end of run.

Fig. 15. Pressure Losses in Piping

Then P_2 (approx.)

$$= 0.0625 \times (1.5L/100)$$
$$= 0.094\,L/100 \text{ lb. per sq. in.}$$

P_3 = pressure differential required to overcome pressure loss through a graduated inlet valve wide open. With proper valve size, P_3 should not exceed 0.125 lb. per sq. in. at normal rated operation. Exact data should be obtained from the maker of valve used.

P_4 = pressure differential required to overcome pressure loss through the radiator return valve as applied to a vapor system. In thermostatic return valves for vapor systems in normal operation, P_4 (approx.) = 0.125 lb. per sq. in. With thermostatic valves for pressure systems (1 to 2 lb.), P_4 = (approx.) 0.25 lb.

P_x = pressure loss in radiator and return lines; it is negligible if returns are properly graded.

P_5 = effective pressure causing air-vent check valve to open at end of return line, or the effective pressure to operate check valve on return line at boiler, whichever is greater. Air-vent check valves usually are rated on a basis of completely venting the system of air in 40 min. at starting, with a pressure differential of 0.0625 lb. per sq. in. In normal operation perhaps 0.05 lb. or less is required. A horizontal swing check may require approximately 0.10 lb. per sq. in. in normal operation. A swing check in the return main adjacent to the boiler, or on a drip line, requires an effective head of approximately 3 in. to 4 in. of water, $\approx$ 0.103 to 0.138 lb. per sq. in. The cubic content of the radiation and connecting piping is approximately 3 cu. ft. per 100 sq. ft. of installed direct radiation.

The sum of the losses P as above indicated is

$$P = P_1 + P_2 + P_3 + P_4 + P_5 = 0.011 + (0.094\,L/100) + 0.125 + 0.125 + 0.10$$
$$= 0.361 + (0.094\,L/100).$$

Assuming L = 200 ft., P = 0.361 + 0.188 = 0.549 lb. per sq. in.; then $H = P/0.415$ = 1.32 ft. = 15.8 in.

During the starting period (cold radiation and maximum pressure condition) the condensation rate, and consequently, the weight of steam flow and pressure loss in the main, inlet valve, and radiator trap may approximate twice the amounts stated. For the example given, H then approximately equals 30 in. Vapor systems using thermostatic traps place the air trap on the return main 24 to 30 in. above normal water line of boiler to prevent flooding of the dry return main.

PRESSURE LOSS IN STEAM MAINS.—The pressure loss in a pipe full of flowing steam may be approximated by the Unwin or Babcock formula

$$\sqrt{\frac{y\,p\,d^5}{L \times (1 + 3.6/d)}} \quad . \quad . \quad . \quad . \quad . \quad . \quad . \quad . \quad . \quad [3]$$

where W = weight of steam flow, lb. per min.; L = length of pipe, ft.; d = diam. of pipe, in.; y = density of steam, lb. per cu. ft.; p = loss of pressure, lb. per sq. in. Figs. 16 and 17 are based on this formula. Under normal conditions of operation, 1 sq. ft. of direct radiation will condense 0.25 to 0.30 lb. per hour. The density of steam, y, is 0.04 lb. for 2.3 lb. per sq. in. pressure. The capacities of steam mains in Table 19 were

calculated by the above formula, the pressure loss p being limited to 1 oz. or 0.062 lb. per sq. in. per 100 ft. of straight pipe. To allow for fittings, use the data in Table 16. The steam main should not be smaller than the riser connected to it.

ALLOWABLE PRESSURE DROP IN LOW-PRESSURE, VAPOR AND VACUUM STEAM-HEATING MAINS.—Usual practice has been to design steam-heating mains in buildings on a basis of 1 oz. pressure drop per 100 ft. of run regardless of length. Good practice (1935) limits the total drop in pressure from boiler to the farthest radiator to approximately the values given in Table 17.

The allowable pressure loss per 100 ft. of run is $p_l = \{p_t \div (L/100)\}$, where p_t = total allowable pressure drop, and L = length of run, ft. See Table 18. The length of run includes allowances for elbows, side outlet tees and valves.

Table 16.—Resistance of Valves and Fittings

Length of pipe to be added to measured length of run to obtain equivalent length of run, in feet.

Pipe size, in.	2 1/2		10	12	14
Standard elbow.		35	39	47	53
Side outlet tee..		69	76	90	105
Gate valve......		13	15	18	20
Globe valve.....		105	118	140	160
Angle valve.....		47	52	63	72

Table 17.—Pressure Drop in Steam Mains

Type of System	Total Drop
One-pipe low-pressure gravity systems, equivalent length of run 200 ft. or less......	2 oz.
Two-pipe low-pressure gravity systems, equivalent length of run 200 ft. or less.....	2 oz.
Two-pipe vapor systems, equivalent length of run 200 ft. or less..................	2 oz.
One-pipe low-pressure gravity systems, equivalent length of run 200 ft. to 600 ft....	4 oz.
Two-pipe low-pressure gravity systems, equivalent length of run 200 ft. to 600 ft...	4 oz.
Two-pipe vapor systems, equivalent length of run 200 ft. to 400 ft................	2 oz.
Two-pipe vapor systems, equivalent length of run 200 ft. to 600 ft................	4 oz.
Vacuum pump systems, equivalent length of run 200 ft. to 600 ft.................	4 oz.
Vacuum pump systems, equivalent length of run 200 ft. to 1200 ft...............	8 oz.

Table 18.—Pressure Drop, Ounces per Square Inch per 100 Feet of Run

Total Drop, oz. per sq. in.	Equivalent Length of Run, feet									
	100	200	300	400	500	600	700	800	900	1000
2	2	1	0.67	0.50	0.40	0.33	....	....	....	....
4	4	2	1.33	1	.80	.67	....	....	....	....
8	....	4	2.66	2	1.6	1.33	1.14	1	0.89	0.80

Table 19.—Capacities of Steam Mains, Branches and Risers

Capacities stated in equivalent square feet of direct radiation. One sq. ft. of equivalent direct radiation assumed to condense 0.25 lb. of steam per hr.

Nominal Pipe Size, in.	Steam Mains and Down-feed Risers Dripped; Branches to Risers Dripped, Steam and Condensate Flowing in Same Direction. Pressure Loss, oz. per 100 ft.			Branches to Risers Not Dripped*		Up-feed Supply Risers	
	1	2	4	One-pipe Gravity Systems	Two-pipe Gravity, Vapor and Vacuum Systems	One-pipe Gravity Systems †	Two-pipe Gravity, Vapor and Vacuum Systems ‡
3/4	§	§	§	....	....	25	30
1	55	80	110	20	26	45	55
1 1/4	120	175	245	55	58	98	120
1 1/2	190	270	380	80	95	152	190
2	385	550	770	165	195	288	385
2 1/2	635	900	1,270	260	395	464	635
3	1,165	1,645	2,325	475	700	799	1165
3 1/2	1,735	2,460	3,475	745	1150	1144	1735
4	2,460	3,475	4,915	1110	1700	1520	2460
5	4,545	6,430	9,090	2180	3150	....	....
6	7,460	10,555	14,925	....	....	....	....
8	15,335	21,970	31,070	....	....	....	....
10	28,345	40,085	56,690	....	....	....	....
12	45,490	64,335	90,990	....	....	....	....

* Radiator branches more than 8 ft. long to be one pipe size larger than table. † Based on tests by A.S.H.V.E. Research Laboratory. ‡ Based on 1 oz. pressure loss per 100 ft. run. § See Table 21 for size of Radiator Valves.

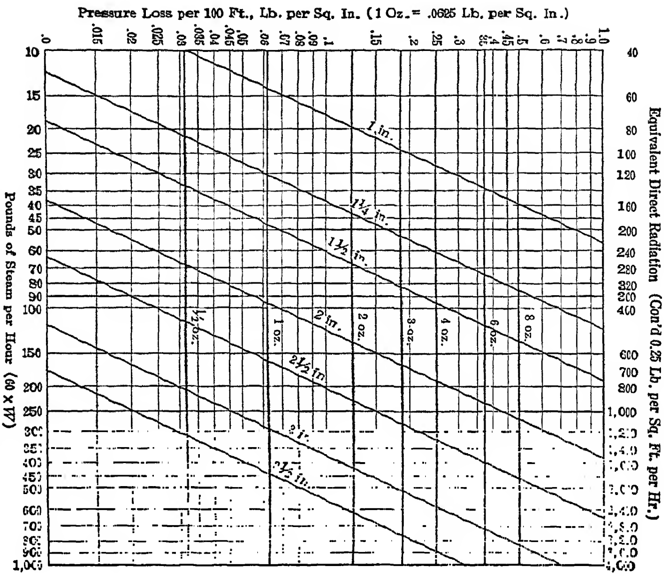


Fig. 16. Pressure Losses in Steam Mains, 40 to 4000 sq. ft. E.D.R.

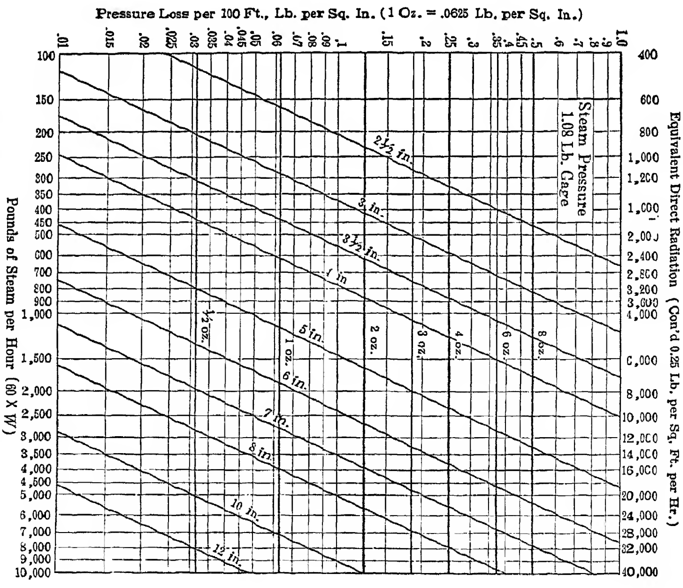


Fig. 17. Pressure Losses in Steam Mains, 400 to 40,000 sq. ft. E.D.R.

EXAMPLE.—In a vapor system the measured distance from the boiler to the farthest radiator, including allowances for ells, is 350 ft. Total allowable drop is 2 oz. Allowable drop per 100 ft., 2 ÷ (350/100) = 0.57 oz. = 0.0356 lb. per sq. in. A horizontal line through this pressure loss per 100 ft. in Figs. 16 and 17, intersects the diagonal pipe size lines. The equivalent direct radiation for a 2-in. pipe is read at the top of the chart as 285 sq. ft.

PIPE SIZES FOR LOW-PRESSURE STEAM, VAPOR AND VACUUM SYSTEMS.—Tables 19 and 20 may be used to determine pipe sizes in buildings for all types of low-pressure steam and vapor systems. They represent present (1935) practice. The rating of the steam mains is based on pressure losses of 1 oz., 2 oz., and 3 oz. per 100 ft. of run. To design the steam main for a fixed total pressure loss, P, for a length, L, determine the pressure loss per 100 ft. of run, equal to $P \div L/100$; locate this pressure loss on the chart; from the intersection of the horizontal pressure loss line with the vertical line corresponding to the weight of steam to be carried by the pipe per hour or the equivalent direct radiation, determine the nearest size of pipe required. It is advisable in any large gravity steam system to check the total pressure loss in the system.

Table 20.—Capacities of Dry and Wet Return Mains

Capacities stated in equivalent square feet of direct radiation.

Nominal Pipe Size, in.	Dry Return Mains: 1- and 2-pipe Gravity and Vapor Systems Up to 200 Ft.*	1- and 2-pipe Gravity Systems Exceeding 200 ft. Length *, Length, L, ft.: 300	400	600	2-pipe Vapor Systems Exceeding 200 ft. Length*, Length, L, ft.: 300	400
3/4	......	......	......	......	......	......
1	320	370	320	275	285	250
1 1/4	670	770	670	480	595	520
1 1/2	1,058	1,210	1,058	757	945	820
2	2,300	2,640	2,300	1,630	2,140	1,880
2 1/2	3,800	4,380	3,800	2,770	3,470	3,040
3	7,000	8,000	7,000	5,000	6,250	5,480
3 1/2	10,000	11,500	10,000	7,200	8,800	7,800
4	......	......	15,000	10,700	13,400	11,700

Nominal Pipe Size, in.	Vacuum System: Return Mains and Return Risers *, Length, L, ft.: 100	300	600	Wet Return Mains: Gravity and Vapor Systems. Pressure Loss, 1/2 in. Water per 100 ft. Run †, Length, L, ft.: 100	200	400	600
3/4	800	462	326	......	......	......	......
1	1,400	810	570	1,525	1,083	762	625
1 1/4	2,400	1,387	976	3,255	2,311	1,627	1,335
1 1/2	3,800	2,195	1,547	4,541	3,224	2,270	1,862
2	8,000	4,622	3,256	8,450	6,000	4,425	3,465
2 1/2	13,400	7,745	5,453	13,176	9,355	6,588	5,402
3	21,400	12,360	8,710	21,122	15,000	10,511	8,660
3 1/2	32,000	18,490	13,020	32,500	23,075	16,250	13,325
4	44,000	25,430	17,910	45,077	32,000	22,538	18,482

* Recommendations of Joint Committee, A.S.H.V.E. and H.P.C.N.A., also A.S.H.V.E. Minimum Requirements Code.

† Calculated from formula proposed by Dr. Biel. (See Heating, Ventilation and Air Conditioning, p. 416, John Wiley & Sons, New York.)

NOTE.—For capacities for any length of run L_1, divide capacities given in table in the column under Length, L by $\sqrt{L/L_1}$. Minimum grade for steam and dry return mains 1 in. per 40 ft. Minimum grade for horizontal branches to radiators 1 in. per 20 ft. Above table applies to pipes properly reamed and first-class workmanship.

Table 21.—Radiator Valve Capacities and Vertical Connections

Square feet, equivalent direct radiation

Size, in.	Single Pipe Gravity Systems	Two-pipe Gravity Systems: Radiator Supply Valve	Two-pipe Gravity Systems: Return Trap	Vapor and Vacuum Systems
3/4	...	30	120	Use manufacturers listed capacities for valves and return traps
1	20	55	190	
1 1/4	55	120	335	
1 1/2	81	190	...	
2	165	385	...	

5. EXHAUST STEAM HEATING

The economy of using exhaust steam for heating is apparent, since approximately only 7% of the heat above 32° F. supplied to the average non-condensing engine appears as work in the steam cylinder. Approximately 80% of the exhaust may be utilized for heating, drying, etc. The steam consumption of *non-condensing* automatic high-speed engines and turbines in first class condition with atmospheric exhaust is given in Table 22, when operating at normal load. The arrangement of piping for an exhaust steam heating system is shown in Fig. 18.

Direct-acting feed pumps consume approximately 4% of the total steam generated by boilers; forced draft equipment approximately 2% to 3%. A feedwater heater will condense approximately 17% of the total weight of exhaust steam when heating feedwater from 50° to 210° F., and 6% when heating the water from 150° to 210° F. The latter assumption may be used when all of the exhaust is utilized and the heating returns are piped back to the feedwater heater. Allow 20% loss by radiation in piping and heater in determining the net direct radiation which the power equipment will supply.

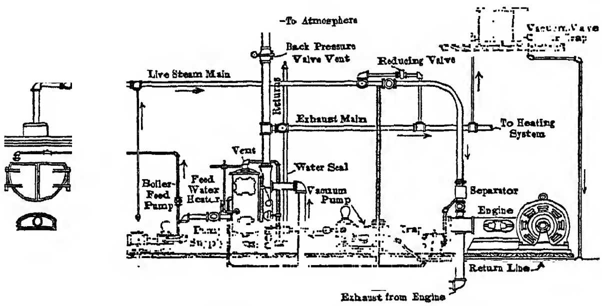


Fig. 18. Piping for Exhaust Steam Heating System

Example.—Required the amount of direct radiation (0.25 lb. condensation per sq. ft. per hour) which a 200-kw. non-condensing engine-driven unit will supply; hand-fired natural-draft boiler plant and 5 lb. back-pressure on the engine.

Solution.—Boilers must evaporate: $200 \times 48 = 9600$ lb. of water per hour for engine; $(9600 \times 0.04)/(1.00 - 0.04) = 400$ lb. per hour for feed pump, or a total of 10,000 lb. per hour. The feedwater heater will condense approximately 6% of this amount or 600 lb., leaving $(10{,}000 - 600) = 9400$ lb. of steam per hour for heating. Deducting 20% from this amount for unavoidable losses, 7520 lb. of steam per hour is available for the radiation. Maximum amount of radiation which plant will supply is: $7520 \div 0.25 = 30{,}000$ sq. ft.

A vacuum system should be used in conjunction with exhaust steam heating in order to obtain good steam circulation with a minimum of back pressure on the engine. For additional information, see L. A. Harding, Power from Process and Space Heating Steam, *Trans.* A. S. H. V. E., 1930.

Table 22.—Approximate Steam Consumption of High-speed Engines and Turbines

Size of Unit		Steam Consumption per Hour	
I.Hp.	kw.	Per I.Hp.	Per kw.
Engines 100 lb. Gage Initial Pressure			
10–25	....	45	....
50	30	33	51
100	65	29	55
300	200	28	43
Turbines 150 lb. Gage Initial Pressure			
...	50	...	56
...	100	...	47
...	200	...	41
...	500	...	36
...	1000	...	34

1 kw. = 1.34 electrical Hp. Allowing for efficiency of engine and generator, 1 kw. at the generator terminals requires approximately 1.55 I.Hp. The above water rates will be increased approximately 3% for 2 lb. back-pressure and 10.5% for 5 lb. back-pressure.

6. DIRECT HOT-WATER HEATING

SYSTEMS IN USE.—Direct hot-water radiator heating systems may be divided into two general classes: 1. All systems operating by gravity only, depending on the difference in density of the water columns in the flow- and return-lines to cause circulation; 2. Systems in which a forced circulation is maintained by a pump placed just in front of the boiler or heater on the return-line. Class 2 systems usually are employed only in large installations, or in district heating service.

GRAVITY HOT-WATER HEATING SYSTEM.—The gravity systems are: *a.* Up-feed systems, using basement mains; *b.* Down-feed systems, using overhead or attic mains. Up-feed systems may have either a 1-pipe or 2-pipe basement main; and the latter type may have either a direct or a reversed return main. (See Fig. 20, for reversed return.) The down-feed systems may have either single or double risers. Either system may operate with an open or a closed expansion tank, as shown in Fig. 19. In general, the down-feed or overhead systems are more positive, permit the use of smaller mains and risers, and provide for the automatic removal of air from radiators and piping. For proper installation of overhead mains and branches, the headroom in the attic must be at least 4 ft. If the overhead mains can be run at the ceiling of the top floor, this restriction does not apply. Mains in attics must be well insulated to prevent freezing.

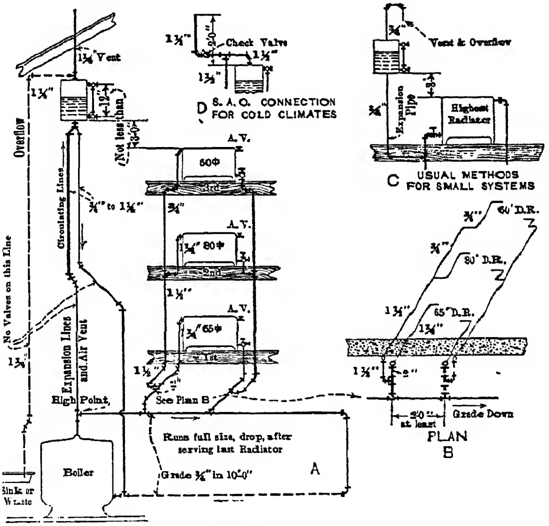


Fig. 19. Expansion Tank Connections

Under-feed systems are used where basement space of little or no value is available, and the radiation is located on two or more floors; or where attic space is so limited that overhead mains and branches cannot be installed. Under-feed systems are liable to be unsatisfactory in buildings less than two stories high, as the motive head, with radiators on the first floor only, is so slight, that faulty or deficient circulation is probable.

The only rational method for designing gravity flow hot-water piping is to balance the friction head against the head available. The head available is calculated from the difference in weight of the water in the flow and return lines. The friction head formulas for pipe of American manufacture, valves and fittings, was determined by Dr. F. E. Giesecke, in 1924. Space limitations do not permit of the reproduction of the method or tables required to use the rational basis for determining pipe sizes.

See The Friction of Water in Pipes and Fittings, F. E. Giesecke, Univ. of Texas *Bull.* No. 1759; The Art of Heating Buildings by Gravity-flow Hot Water Heating Systems, F. E. Giesecke, a thesis for the doctor's degree conferred by the Univ. of Illinois, June, 1924. See also *Heating* and *Vent. Mag.*, for The Friction of Water in Elbows, F. E. Giesecke, C. P. Reming, and J. W. Knudson, Jr., Bureau of Engineering Research, Univ. of Texas *Bull.* 2712, March 22, 1927.

Up-feed, One-pipe Systems.—The up-feed, 1-pipe system consists of a supply main in the basement, sloping down $^3/_4$ in. per 10 ft., from a point close to the ceiling above the boiler to beyond the last supply branch, after which it drops and returns to the boiler, together with flow-risers and radiator branches taken off at the top, and return-branches entering at the side or bottom of the main. The main is of the same diameter throughout the circuit. In the case of branches near the boiler, or branches supplying only upper-floor radiators, flow connections may be made to the supply main at 45° instead of at the

top. Radiators on upper floors will assist the circulation in radiators on the first floor, if the upper-floor risers are taken from the side of the branches supplying the first floor radiators. See Fig. 19*A* and *B*. First-floor branches usually are all full size.

Radiators should be connected at the top to the supply by a union-elbow, and at the bottom to the return, by a quick-opening hot-water radiator valve. Only one valve is required to control a radiator. The area of the last radiators on a main should be increased from 5% to 10%, as the temperature of the water is gradually decreased in passing through the preceding radiators; it is advisable to increase the size of branch and riser connections at the end of a main by one pipe size.

Tables 23 and 24 (J. J. Hogan) give data for proportioning piping. In using the tables, all mains must be measured back to the boiler. For mains over 100 ft. long, reduce the capacity in the ratio of $\sqrt{100 \div L}$. Risers to a given floor must be made large enough to supply not only the radiators on that floor, but also on all the floors above it.

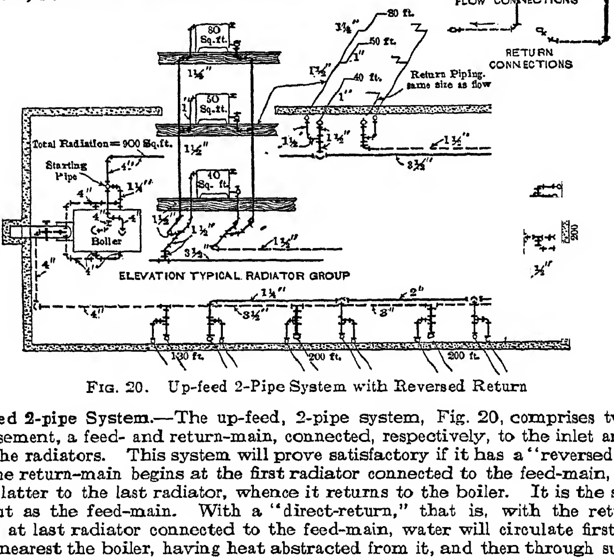


Fig. 20. Up-feed 2-Pipe System with Reversed Return

Up-feed 2-pipe System.—The up-feed, 2-pipe system, Fig. 20, comprises two mains in the basement, a feed- and return-main, connected, respectively, to the inlet and return risers of the radiators. This system will prove satisfactory if it has a "reversed return," that is, the return-main begins at the first radiator connected to the feed-main, and parallels the latter to the last radiator, whence it returns to the boiler. It is the same size throughout as the feed-main. With a "direct-return," that is, with the return-main beginning at last radiator connected to the feed-main, water will circulate first through radiators nearest the boiler, having heat abstracted from it, and then through succeeding

Table 23.—Sizes of Basement Mains for Hot-water, Up-feed, Open Tank Heating Systems. (Mains up to 100 ft. long)

Pipe Size, in.	Direct Radiation, sq. ft.	Indirect Radiation, sq. ft.	Pipe Size, in.	Direct Radiation, sq. ft.	Indirect Radiation, sq. ft.	Pipe Size, in.	Direct Radiation, sq. ft.	Indirect Radiation sq. ft.
1 1/4	135	100	3 1/2	850	650	7	4,800	3,900
1 1/2	220	135	4	1100	850	8	6,200	5,000
2	350	225	4 1/2	1350	1050	9	7,700	6,300
2 1/2	460	320	5	1700	1350	10	9,800	7,900
3	675	500	6	3600	2900	12	14,000	11,400

Table 24.—Sizes of Branches and Risers for Hot-water, Up-feed, Open Tank Heating Systems with Basement Mains

Pipe Size, in.	Direct Radiation, sq. ft.				Pipe Size, in.	Direct Radiation, sq. ft.			
	1st Floor	2d Floor	3d Floor	4th Floor		1st Floor	2d Floor	3d Floor	4th Floor
3/4	30	45	55	70	2 1/2	400	490	525	550
1	60	75	85	95	3	620	650	690	730
1 1/4	110	120	135	150	3 1/2	820	870	920	970
1 1/2	180	195	210	230	4	1050	1120	1185	1250
2	290	320	350	370	4 1/2	1325	1400	1485	1560

radiators in turn. The last radiators thus will be slow in warming up and the system may prove unsatisfactory. With the reversed return each radiator offers the same resistance to the flow of water and all become warm at the same time. With the reversed return, the flow is in the same direction as in the flow main. The return increases progressively in size while the flow main decreases. Flow mains should slope up from, and return mains down to, the boiler 3/4 in. per 10 ft. Pipe sizes in Tables 23 and 24 will apply to 2-pipe systems, and the size of main should be reduced or increased as rapidly as the change in radiation supply will permit. A "starting pipe" of from 1 1/4 to 2 1/2 in. diam. is, in government work, installed between the flow main and the return at the boiler in under-feed systems, to assist establishing initial circulation between flow and return headers.

EQUALIZATION OF PIPES.—The relative capacities of different sizes of pipe for the same friction loss per 1000 ft. of run are as follows:

Pipe Size, in.	1/2	3/4	1	1 1/4	1 1/2	2	2 1/2	3	3 1/2	4	5	6	7	8
Relative Capacity.	2	5	10	20	30	60	110	175	260	380	650	1050	1600	2250

The equivalent numbers are proportional to the 5/2 powers of the diameters and the quantity of water W flowing equals $Kd^{5/2}$, where K = a constant and d = pipe diameter, in.

Example.—Find a pipe of equivalent capacity to a 1 1/4-, 1 1/2- and 2-in. pipe.

Solution.—The equivalent capacity numbers are 1 1/4 = 20; 1 1/2 = 30; 2-in. = 60. 20 + 30 + 60 = 110, which is the number equivalent to a 2 1/2-in. pipe.

DETAILS OF PIPING SYSTEMS FOR GRAVITY HOT-WATER HEATING.—**Mains and Branches** must be uniformly graded with provision for expansion and contraction by means of flexible double elbow joints or otherwise. Air traps and pockets must be avoided and automatic air outlets provided at the top of all points where such pockets may occur. Eccentric fittings must be used wherever the mains are reduced in size to keep the tops of the pipe in the same plane and avoid air pockets. The piping must be so arranged that the system will completely drain of water when the blow-off cock at the boiler is open. Branch mains from a header at the boiler must all rise to the same elevation; the tops of all branches must lie in the same plane as they start away from the boiler.

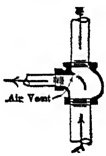


Fig. 21. O. S. Fitting

Long Sweep Fittings must be used on all main piping and branches. Risers supplying radiators on two or more floors should be connected through special tees, known as O. S. fittings, to the branches. See Fig. 21. The deflector to divert the current of flow into the outlet of the tee will favor the radiators on the intermediate or lower floors. The maximum length of branch employed above the floor for connecting either steam or water radiators is 9 in. The branch must run in the floor or under the ceiling if longer than 9 in. Risers to upper floors should not be over 2 in. from finished walls.

Radiators should be connected to the flow at the top and to the return at the bottom. A single valve on the return thus will control the radiator. With both connections made at the bottom, two valves are necessary.

Air Removal.—Small air cocks or automatic air relief valves should be attached to the high point of each radiator on all up-feed systems and opened periodically to relieve air accumulations. Automatic air valves heretofore have been liable to derangement and to passing water as well as air. Unless air accumulations are removed, faulty circulation and failure to heat the radiator will result.

EXPANSION TANKS.—A suitable tank to take care of the expansion and contraction of water in the system must be provided on all low-pressure hot-water heating systems. This tank should be opened and connected to the nearest return riser or to a separate expansion line at an elevation at least 3 ft. above the highest radiator. The capacity of the tank depends on the amount of water in the system and on its temperature range. Water heated from 32° to 212° F. increases in volume approximately 4.33%. Hence, for every 23 gal. of water in the system at 32° F., a tank capacity of 1 gal. must be provided, when water is heated to 212° F. Cast-iron radiators have an internal volume of about 1 1/2 pint per sq. ft.; steel radiators and 1-in. pipe contain about 1 pint per sq. ft. of surface. It generally is assumed that the internal volume of radiators is 50% of the volume of the entire system and the capacity C of the tanks on this basis, therefore, would be, $C = (R \times 2) \div 0.125$, where C = capacity, gal. and R = sq. ft. of radiation in radiators.

Table 25 shows the sizes and capacities of commercial expansion tanks. These are slightly smaller in the larger sizes than given by the above formula, but will prove satisfactory with all systems employing up to 6000 sq. ft. of radiation. For larger systems, the size of the tank should be determined separately.

Expansion tanks on 1-pipe systems preferably are connected through an expansion line from the high point of the main, just above the boiler connection, to a return bend just below the tank, with a return circulating line connected through the side of the bend

with the return main at the boiler. See Fig. 19. A 1 1/4-in. vent from the top of the tank should lead through the roof with a 1 1/4-in. overflow connected in the vent line just above the tank. The vent should discharge into an open sink or a drain near the boiler. In small installations, the expansion line may be connected to a radiator return riser.

CLOSED TANK SYSTEMS.—In open-tank systems, the highest temperature practically possible is 212° F. At this temperature, the water, rising into the open tank, will boil and empty the system of water. The remedy is to raise the pressure on the system, which, by increasing the hydrostatic head, will raise the boiling point. The hydrostatic head may be increased by interposing a column of mercury in the expansion line. A device for this purpose is the Honeywell "heat generator," Fig. 22. Water entering the generator forces the mercury up in tube *A*, until a head of 20 in. or 10 lb. is established, and the entrance to the tube is uncovered. Water and air then may pass to the expansion tank. Any excess of mercury over that required to fill tube *A*, returns by tube *B* to the reservoir. When the system cools, water flows back in tube *A* as the mercury column drops, the small head of mercury at the outlet of the tube being overcome by the head of water in the expansion tank. The 10-lb. increase in static pressure makes possible a temperature of 240° F. in the system, and allows the use of smaller radiators. A greater difference in temperature will exist between flow and return risers than in open tank systems, giving a greater motive head, and permitting smaller mains and risers to be used. As the system will contain less water, it will be more sensitive both in heating up and in cooling off. The "generator" may be located any place in the expansion line, excepting close to the expansion tank. Other devices for the same purpose as described for the generator are available.

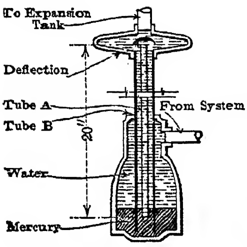


FIG. 22. Heat Generator

Table 25.—Sizes and Capacities of Expansion Tanks

Size, No.	Size, in.	Capacity, gal.	Sq. Ft. of Radiation	Size, No.	Size, in.	Capacity, gal.	Sq. Ft. of Radiation
0	10×20	8	250	5	16×36	32	1300
1	12×20	10	300	6	16×48	42	2000
2	12×30	15	500	7	18×60	66	3000
3	14×30	20	700	8	20×60	82	5000
4	16×30	26	950	9	22×60	100	6000

7. FURNACE HEATING

The term applied to warming or heating a building by a hot-air furnace is *furnace heating*. Furnaces for soft coal usually are designed with a secondary air supply or over-draft, to admit heated air at the surface of the fire to consume the volatile gases liberated immediately after firing. The over-draft should be so controlled that it may be checked or closed when the fuel has coked. Soft coal also may be burned efficiently in the under-feed type of furnace.

The furnace should be centrally located in the basement with reference to the rooms to be heated, and preferably toward the side or sides exposed to prevailing winds in winter. This arrangement favors the more exposed rooms by shortening the leaders to them and reduces the length of cold-air duct which should run from the exposed side of the building. In operation, some or all of the air circulated is cold air drawn from outdoors through the cold-air duct, passed through the space between the heater and its jacket, and warmed by contact with the surface of the combustion chamber and the radiator just above it.

LEADERS AND STACKS.—Leaders are the nearly horizontal round pipes in the basement, connecting the furnace to the vertical rectangular pipes or stacks leading to the register grilles. Leaders should pitch upward toward the base of the stack at least 1 in. per ft. For best results, they should not be over 12 to 15 ft. long. Stacks should run between the studding of interior walls or partitions. If placed in outside walls, the cooling effect reduces their efficiency, both in temperature of air and velocity of flow. Leaders and stacks usually are made of bright IX tin, although for leaders larger than 12 in., galvanized steel, No. 26 U. S. Std. gage, is used. Leaders, boots and stacks, and also the furnace, should be covered with first-class insulation. As stacks are usually in a 4-in. studding-space, with a net depth of about 3 3/4 in., they should be kept as deep as possible. Steel lath or expanded metal should be used in front of all such stacks which usually have but one layer of asbestos-paper covering. The double-wall stack, with space between

inside and outside pipes, and without asbestos covering, is more effective. Attention is called to the fact that where large second-floor rooms are heated by one register, 6-in. stud partitions usually are required on the first floor.

Design of Furnace Heating Systems

HEAT LOSS AND AIR REQUIRED.—The size of the furnace, and of the connecting leaders, stacks, etc., depends on: 1. The actual heat loss from each room in the building, including wall and glass transmission losses, as well as loss due to infiltration. 2. The amount of air to be circulated per hour, which, in turn, is based on this heat loss. A building is heated by hot-air by introducing the air into the rooms at a temperature above that maintained in the rooms at the breathing line (approximately 70° F.). The air, in cooling, gives up 0.24 B.t.u. per lb. for each degree drop in temperature, thus supplying the heat necessary to offset the heat transmission of the walls, etc. The maximum temperature of the air leaving the furnace cap is approximately 190° F.; it leaves the registers at 175°. These figures are maximum values, not to be exceeded. If all the air is drawn from outdoors, and the outdoor temperature is 0° F., then the air is heated, from 0 to 190° F., and cooled on entering the room from 175° to 70° F., or 105° F. That is, 0.24×105 or 25.2 B.t.u. apparently is thrown away for every pound of air circulated. If all the air for ventilation must be brought from outdoors, this is the price that must be paid for ventilation. It would be the same for equally good ventilation, irrespective of the system of heating. Usually, however, much, if not all, of the air may be recirculated. In this event, for equal ventilation, the furnace system requires no greater expenditure of fuel than a direct steam or hot-water system, and is just as economical when correctly designed, installed and operated. The head, producing the flow, is due to the difference in weight between the ascending column of heated air and the weight of an imaginary similar column of the colder intake air. The system generally is proportioned for recirculating all the air during extreme cold weather.

WEIGHT OF AIR TO BE CIRCULATED PER HOUR.—Let W = air to be circulated per hour, lb.; t = inside temperature to be maintained, deg. F.; T = temperature of air leaving registers deg. F. (assumed 15° lower than temperature leaving furnace cap); H = B.t.u. to be supplied to room per hr., as determined by heat loss calculations (see page 11–03). $0.24 \times (T - t)$ = B.t.u. given up per lb. of air circulated. Then,

$$W = H \div \{0.24(T - t)\} \qquad [4]$$

The maximum value for T is approximately 175° F. and t = 70° F. Then, $W = H/25.2$, and $Q = W/d = W/0.062 = H/1.6$, where Q = warm air entering the room, cu. ft. per hr., and d = density of entering air = 0.062 at 175° F.

HEAT REQUIRED FROM FURNACE, BASED ON RECIRCULATION.—The heat required per hour from the furnace depends on the temperature of the entering air. It will be a maximum if all air circulated is taken from outdoors, and a minimum if all the air is recirculated. Let h = B.t.u. required from heater per hr.; T_e = temperature of air entering heater = 65° F.; T_h = temperature of air leaving heater = 190°. Then

$$h = 0.24\,(T_h - T_e)\,W \qquad [5]$$

Substituting the values given above for T_h and T_e, $h = 1.2\,H$.

SIZE OF FURNACE.—The capacity of the furnace for heating air depends primarily on the amount of coal that may be burned per hour or (rate of combustion) × (grate area). With an assumed rate of combustion, capacity depends on grate area. Grate area, therefore, is the basis for rating and comparison of hot-air furnaces. The average rate of combustion in furnace heating ranges from 3 to 4 lb. per sq. ft. of grate per hr. In zero weather this rate may be as high as 6 lb., and readily is obtainable with the ordinary height of residence chimney, that is, at least 35 ft. A properly-designed furnace will have a combined furnace-and-grate efficiency of 55 to 60%.

FURNACE RATING BASED ON EFFICIENCY AND RATE OF COMBUSTION.—The B.t.u. per hr. that a furnace can impart to the air (not to the room) also may be estimated from the grate area by assuming that the average coal used has a heat value of 12,000 B.t.u. per lb. A combined furnace-and-grate efficiency of 55% and a maximum combustion rate of 6 lb. per sq. ft. of grate per hr. for coldest weather conditions also usually are assumed.

GRATE SURFACE REQUIRED BASED ON RECIRCULATION.—The area of the grate is readily calculated when the heat to be supplied per hr. has been determined. Let H = B.t.u. to be supplied per hr.; h = B.t.u. required from furnace per hr. for heating the air = $1.2\,H$; C = B.t.u. of coal per lb.; E = combined furnace-and-grate efficiency; R = rate of combustion, lb. of coal per sq. ft. of grate per hr.; G = grate area, sq. ft.

Table 26.—Air-heating Capacity of Warm-air Furnaces

Fire-pot		Casing *	Total Cross-sectional Area of Heat Pipes, sq. in.	Number and Size of Heat Pipes that May be Supplied
Diameter, in.	Area, sq. ft.	Diameter, in.		
18	1.8	30–32	180	3–9″ or 4–8″
20	2.2	34–36	280	2–10″ and 2–9″ 2– 9″ and 2–8″
22	2.6	36–40	360	3–10″ and 2–9″ 4– 9″ and 2–8″
24	3.1	40–44	470	3–10″ and 1–9″ 2–10″ and 5–8″
26	3.7	44–50	565	5–10″ and 3–9″ 3–10″, 4– 9″ and 2–8″
28	4.3	48–56	650	2–12″, 3–10″ and 3–9″ 5–10″, 3– 9″ and 2–8″
30	4.9	52–60	730	3–12″, 3–10″ and 3–8″ 5–10″, 5– 9″ and 1–8″

* The casing diameter should be such that the minimum cross-sectional area between casing and radiator will be at least 20% greater than the total cross-sectional area of all the heat-pipes.

Table 27.—Capacities and Dimensions of Warm-air Piping and Registers

Diameter of Round Cellar or Riser-pipe, in.*	Proper Size of Rectangular Riser-pipe, in.*	Area of Riser-pipe, sq. in.	Required Area of Register-face, sq. in.*	Diameter of Round Cellar or Riser-pipe, in.*	Proper Size of Rectangular Riser-pipe, in.*	Area of Riser-pipe, sq. in.	Required Area of Register-face, sq. in.*
6	3 × 9 1/2	28	52	13	6×22	132	242
6 1/2	3 1/2× 9 1/2	33	62	13 1/2	8×18	143	254
7	3 1/2×11	38	72	14	8×19	154	276
7 1/2	3 1/2×12 1/2	44	84	14 1/2	8×20 1/2	165	298
8	3 1/2×14	50	96	15	8×22	176	320
8 1/2	4 ×14	57	108	16	8×25	201	358
9	4 ×16	64	120	17	10×22 1/2	227	410
9 1/2	4 ×18	71	134	18	10×25 1/2	254	450
10	4 ×20	78	142	19	12×23 1/2	283	508
10 1/2	6 ×14 1/2	86	158	20	12×26	314	554
11	6 ×16	95	176	21	12×28 1/2	346	618
11 1/2	6 ×17 1/2	104	194	22	14×27	380	686
12	6 ×19	113	204	23	14×29 1/2	415	707
12 1/2	6 ×20 1/2	122	222	24	14×32	452	770

* When the required size of pipe falls on the odd half-inch (as 7 1/2, 8 1/2, etc.), the size may be increased to the even inch (as 8 for 7 1/2, 9 for 8 1/2, etc.), for the first-floor rooms and bath-rooms; provided that the pipes for upper-floor rooms, other than bath-rooms, be decreased by 1/2 in. when the required sizes fall on the odd half-inch. It is better, however, to use pipes of the sizes given above, with proper allowances for length of pipe, extra bends, etc., beyond straight runs 12 ft. long.

Then $h = G \times C \times E \times R \quad 1.2$; $H = G \times 12{,}000 \times 0.55 \times 5.5 = 36{,}300\,G$; $G = (1.2 \times H) \div 36{,}300$.

The usual assumptions with anthracite are: $C =$ 12,000 B.t.u. per lb., $R = 4$ lb., ordinary rate, and 5.5. lb., coldest weather conditions. $E = 0.55$. Then $G = h \div 36{,}300 = 1.2\,H \div 36{,}300 = H \div 30{,}250$.

SIZE OF LEADERS AND STACKS.—The area of the air pipes (leaders and stacks) required for a room depends on the quantity of air to be introduced per minute and the velocity with which the air will flow with natural circulation. The theoretical velocity of air in a duct depends on the difference in weight of the column of heated air and of an imaginary column of equal height of air at the temperature of air entering the heater.

$Q/60$ = warm air to be introduced per min., cu. ft.; V = attainable velocity of air, ft. per min.; H = heat loss, B.t.u.; A = area of pipe, sq. in. Then, $Q/60 = AV/144$, and substituting value of $Q = H/1.6$; $A = 1.5\,H/V$ sq. ft.

The theoretical velocity of air in a duct is expressed by the formula

$$V = \qquad \times$$

where V = velocity in the duct, ft. per sec.; t_1 = absolute temperature of air in the vent duct; t_2 = absolute temperature of external air; h = vertical height of duct, ft.; $2g = 64.4$. Actual velocities will be considerably lower due to loss by friction, which will vary with the form and cross-sectional area of the duct and its connections, and the degree of smoothness of its interior surface. The following approximate velocities are obtained in leaders and

stacks for the several floors given: First floor, 173 ft. per min.; second floor, 240 ft. per min.; third floor, 322 ft. per min. Then, substituting in the equation in the preceding paragraph, we have: $A_1 = H/115$ (first-floor pipes, leaders, etc.); $A_2 = H/160$ (second-floor pipes, etc.); $A_3 = H/215$ (third-floor pipes, etc.).

Fig. 23 shows graphically the results of tests made under the direction of Prof. A. C. Willard, and described in *Bull.* No. 112, Engg. Experiment Station, Univ. of Illinois, 1919. From the test data it was found that the temperature at the registers on the second floor is approximately 10° lower than on the first floor, so that if 175° is used for the register temperature on the first and third floors, 165° should be used for second-floor registers.

Actual leader and stack sizes are based on the above areas, using the nearest $^1/_2$ in. for leader diameter (Table 27), and keeping the stacks of such proportions that the cross-sectional dimensions are never in a greater ratio than 3 to 1. For example, a stack 4 × 20 in. is seldom effective over its full area, it being too narrow and the large rubbing surface causing excessive friction. The actual velocities obtained, however, will depend on the head causing the flow, and on the friction-head, and seldom will exceed 50% of the theoretical velocity. Table 27 gives sizes of round pipe for leaders and of wall pipe for stacks, and free areas of registers to connect with them. Leaders over 12 ft. long should be increased 1 in. diameter for each 5 ft. over 12 ft.

REGISTERS.—The free area through the ordinary register grille is approximately 55% of the gross area. Hence, the register must have double the gross area of the pipe connected to it to avoid contraction of the air passage and reduced capacity. Commercial register sizes are based on actual inside dimensions of grille opening.

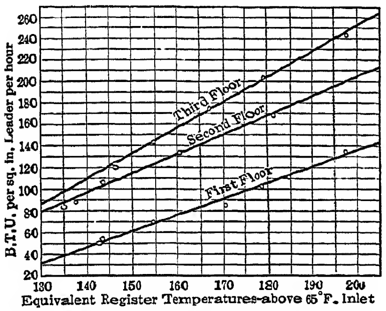


FIG. 23. Effect of Register Temperature on Leader Capacity

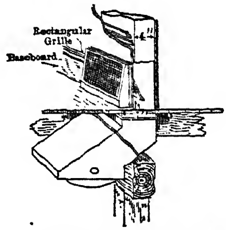


FIG. 24. Baseboard Register

Warm-air registers may be placed in the floor, but preferably in inside partitions for first-floor rooms. The base-board register (Fig. 24) usually gives the required capacity without resort to floor registers. Base-board registers can be connected to a flue 3 to 4 $^1/_2$ in. deeper than the studding by making it project 2 in. into the room at the floor line, and utilizing the 1 in. space occupied by the lath and plaster, a total increase in depth of flue of about 3 in. For upper-floor rooms, registers should be placed in inside partition walls, using convex registers for shallow stacks. As a general rule, warm-air registers should be so placed as to shorten leader and stack connections as much as possible.

8. FAN OR BLAST HEATING SYSTEMS

The mechanical indirect system of heating known as the fan system or blast system, particularly adapted to the warming and ventilating of large structures, consists of 3 units: 1. A heater of pipes, tubes, or cast-iron sections through which steam, hot water or hot gas is passed; 2. A fan or blower to circulate air over the heater surfaces, the air being the medium of heat transfer; 3. A system of ducts or pipes to convey the heated air from the heater. If the heater is located between the fan and main duct the system is termed *blow-through*. If the fan is installed between the heater and the duct, it is termed *draw-through*. (See Fig. 27.) The draw-through system is used for shops where compactness is desirable. The blow-through apparatus is used principally for hot and cold systems installed in schools and public buildings.

If ventilation is not required, or is relatively unimportant, the air simply may be recirculated, sufficient fresh air for ventilation being obtained by infiltration. The heat to be supplied to the heater in this case is the same as in a direct-radiation installation.

When ventilation is required, a cold-air intake is provided and sufficient fresh air is drawn into the system from outdoors to meet the ventilation requirement. The remainder of the air necessary for heating is recirculated. This is effected by an arrangement of ducts and dampers on the suction side of the fan. If the fresh air is to be washed or conditioned, the washer or humidifier and tempering coil is placed between the inlet for the recirculated air and the fresh-air intake. In factory work, the unit heater has largely supplanted the fan or blast system when heating is the only requirement. See Unit Heaters, p. 11–44.

ADVANTAGES OF THE FAN SYSTEM.—*The advantages of the fan system over direct radiation*, briefly, are: 1. It affords positive ventilation, entirely independent of changing climatic conditions; 2. When a standard humidity of air is to be maintained, as is desirable in heating and ventilating, and essential to the manufacture of some materials, the air-conditioning apparatus may be made integral with the system; 3. Less radiating surface is required for equal heating duty, with a consequent reduction in the number of steam-tight joints to keep in repair. 4. Air leakage being mostly outward, the building will be freer from drafts and more uniformly heated. If air is recirculated and no outside air is taken into the heating system from outside, this statement does not apply. The pressure of air in the building, even if all air enters the system from outside, is comparatively feeble, and some air will infilter around windows and doors on the windward side of the building, despite the often-made statement that the outward leakage prevents all infiltration of cold air. 5. The fan system is more easily regulated than direct radiation, and readily responds to changing outside temperature. 6. Air entering for ventilation, may be cooled in summer by circulating cold water or brine, previously cooled by mechanical refrigeration, through the heater. 7. Running the fan will, in itself, relieve oppressiveness in sultry weather, and if cold water is circulated through the coils, the difference is noticeable.

HEATER CALCULATIONS.—Let H = heat loss; H_1 = heat to be supplied by heater; H_2 = heat to be supplied by tempering coil; H_3 = heat to be supplied to humidifier, all in B.t.u. per hr. Let M_o = outside air introduced; M_r = recirculated air; $M = M_o + M_r$ = total air circulated, all in lb. per hr. Let T_o = temp. of outside air; T_1 = temp. of air entering heater; t = temp. to be maintained in room; t_1 = temp. of air leaving heater, t_2 = temp. loss in ducts; t_3 = temp. of air leaving ducts; t_4 = temp. of air leaving tempering coil; t_m = temp. of air entering ducts, all in deg. F.; n = number of occupants of room; Q = ventilation requirements, cu. ft. of air per hr. per occupant = 900 to 1800 (usual school requirements). Specific heat of air at constant pressure = 0.24. Weight of air per cu. ft. at 70° F. = 0.075 lb.

Amount of Air to be Circulated.

For heating, $M = H \div 0.24\,(t_3 - t)$ . . . . [6]

For ventilation, $M_o = 0.075 \times n \times Q$ . . . . [7]

Heat to be Supplied by Heater. $H_1 = 0.24\,(t_1 - T_1)\,M$ . . . . . . [8]

Temperature of Air Leaving Ducts. $t_3 = (H/0.24\,M) + t$ . . . . . . [9]

Temperature of Air Entering Ducts. $t_m = t_3 + t_2$.

Temperature of Air Entering Heater. *a.* When all air circulated is outside air, $T_1 = T_o$; *b.* When all air is recirculated, $T_1 = t$; *c.* When part of air is recirculated,

$$T_1 = \{M_o\,(T_o + 460) + M_r\,(t + 460)\} \div (M_o + M_r) \quad . . . . [10]$$

Temperature of air leaving the ducts, $t_3 = t_1 - t_2$. It depends on temperature T_1 of air entering heater, its velocity through clear area of heater, area of heating surface, steam temperature, and temperature loss in ducts. The usual range of t_3 is from 125° to 150° F. Tables 31 and 32 give values of t_1 for specific conditions. Values of t_2 depend on the location of the ducts. Heat loss from ducts in inside walls helps to heat the building, and t_3 may be taken as equal to t_1; that is, $t_2 = 0$, and $t_m = t_1$. Heat losses from ducts in outside walls or underground must be compensated by increasing t_1 by an amount equal to t_2.

A constant value of t cannot be maintained in several different rooms with different heat losses by controlling the temperature t_1 of air leaving the heater. Temperature t_3 usually will be different for each room. It is controlled by means of the double-plenum chamber system described below.

Air that is recirculated and outside air are passed through a tempering coil that raises it to a temperature t_4, usually from 64 to 70° F. Part of the tempered air passes through the heater. The remainder is by-passed and is mixed with the hot air leaving the heater at temperature t_1. Calculations for the tempering coil are the same as for the heater. The required proportions of heated and tempered air are found by the *method of mixtures* as follows: Let x and $(1 - x)$ = respectively the proportions by weight of heated and tempered air. Then

$$x\,(t_1 + 460) + (1 - x)\,(t_4 + 460) = t_m + 460 \quad . . . . . [11]$$

120° and 64° I

EXAMPLES.—Determine quantity of air to be circulated, and the heat to be supplied to it to offset heat loss from a room maintained at $t = 70°$ F., and whose heat loss is 120,000 B.t.u. per hr. Room contains 60 persons. Outside temperature T_o assumed as 0° F.

Solutions.—Several arrangements are possible. I. All circulated air drawn froı
Assume $t_2 = 5$. $t_3 = 120$. $t_1 = t_2 + t_3 = 125$.

$$M = M_o = H \div 0.24\ (t_3 - t) = 120{,}000/0.24\ (120 - 70) = 10{,}000 \text{ lb.}$$

$$H_1 = 0.24\ (t_1 - T_1) \times M = 0.24\ (125 - 0) \times 10{,}000 = 300{,}000 \text{ B.t.u.}$$

II. All air to be recirculated. Here $T_1 = t = 70$.

$$M = M_r = H \div 0.24\ (t_3 - t) = 120{,}000/0.24\ (120 - 70) = 10{,}000 \text{ lb.}$$

$$H_1 = 0.24\ (t_1 - T_1) \times M = 0.24\ (125 - 70) \times 10{,}000 = 132{,}000 \text{ B.t.u.}$$

III. Part of air to be recirculated, and 1800 cu. ft. of outside air per hr. per person to be su for ventilation.

$$M_o = n \times Q \times 0.075 = 60 \times 1800 \times 0.075 = 8100 \text{ lb.}$$

$$M = M_o + M_r = H \div 0.24\ (t_3 - t) = 120{,}000/0.24\ (120 - 70) = 10{,}000 \text{ lb.}$$

$$M_r = M - M_o = 10{,}000 - 8100 = 1900 \text{ lb.}$$

$$T_1 = [\{M_o\ (T_o + 460) + M_r\ (t + 460)\} \div (M_o + M_r)] - 460$$

$$= [\{(8100 \times 460) + 1900\ (70 + 460)\} \div 10{,}000] - 460 = 17.3° \text{ F.}$$

T_1 is the temperature of the mixture of outside and recirculated air entering the heater.

$$H_1 = 0.24\ (t_1 - T_1) M = 0.24\ (125 - 17.3) \times 10{,}000 = 258{,}480 \text{ B.t.u.}$$

IV. Room heated by direct radiation. Air for ventilation supplied by a fan system. This combination is called a split system. Air conditioned to maintain a constant relative humidity of 35% in room. Outside air is passed through a tempering coil to bring its temperature to 35° F. before it enters washer. With $t = 70°$ F., a relative humidity of 35% corresponds to a dew point temperature of 41° F. Air passing through spray chamber of air washer is saturated and leaves at a temperature of $T_1 = 41°$ F., at which temperature it enters the heater. Assume temperature t_3 of air entering the room to be 80° F. Then

$$M = M_o = n \times Q \times 0.075 = 60 \times 1800 \times 0.075 = 8100 \text{ lb.}$$

For heater, $H_1 = 0.24\ (t_3 - T_1) M = 0.24\ (80 - 41) \times 8100 = 75{,}816$ B.t.u.
For tempering coil, $H_2 = 0.24\ (t_4 - t_0) M = 0.24\ (35 - 0) \times 8100 = 68{,}040$ B.t.u.
For washer $H_3 = 0.73\ M = 59{,}130$ B.t.u.
It is assumed here that a washer with a water heater is used that will fully saturate the air. The heat to be supplied to the washer is the difference between the heat content of dry air entering the washer at 35° F. and of saturated air leaving it at 41° F., or 7.3 B.t.u. per lb.

$$\text{Total heat} = H_1 + H_2 + H_3 = 202{,}986 \text{ B.t.u.}$$

V. Constant temperature $t = 70°$ F. to be maintained in each of several rooms which have different heat losses, as given in Table 28. Ventilation to be provided at rate of 1800 cu. ft. per hr. per occupant. Determine temperature t_3 of entering air. Using room A-2 as an example,

$$M_o = Q \times n \times 0.075 = 1800 \times 53 \times 0.075 = 7125 \text{ lb.}$$

$$t_3 = (H/0.24\ M_o) + t = 21{,}000/(0.24 \times 7125) + 70 = 82.2° \text{ F.}$$

AIR SUPPLIED FOR VENTILATING PURPOSES ONLY.—A combination of direct radiation to offset the heat-loss H, and a fan system, to supply fresh air needed for ventilation, is considered best practice for schools and public buildings. The heater capacity usually is made sufficient to warm the air for ventilation to about 80° F.

HOT-AND-COLD OR DOUBLE-PLENUM-CHAMBER SYSTEMS.—In so-called Hot-and-cold or Double-plenum-chamber Systems all air drawn from outdoors first passes through a tempering-coil, designed to heat air to approximately 70° F. Part of the tempered air then passes through a heater and is raised to from 125° to 150° F. If varying proportions of the hot and tempered air are correctly mixed the resulting temperature T is controlled without varying the quantity of air discharged, which must remain constant on account of the ventilation requirement.

The two methods of distribution used are shown in Figs. 25 and 26. In the single-duct system, Fig. 25, the hot and tempered air meet at the entrance to the ducts, at the

Table 28.—Data for Example

Room Number	Number of Occupants, n	Ventilation: Cubic Feet per Hour at 70° F. $1800 \times n$	Ventilation: Weight per Hour, lb., M_o	Heat Loss, H	Temperature, T, of Air Leaving Ducts, by Formula
A-1	50	90,000	6750	32,000	89.5
A-2	53	95,400	7125	21,000	82.2
Hall	..	30,000	2250	4,000	77.4

end of the plenum-chamber. The temperature of the mixture is controlled by mixing-dampers, which may be operated by hand or automatic thermostatic control. The plenum-chamber is divided, and each duct serving a room has an independent set of mixing-dampers. In the double-duct system, Fig. 26, two ducts run from the plenum-chamber to the base of each vertical flue, carrying hot air and tempered air, respectively, which are mixed at the base of the flue. The mixing-dampers may be controlled by hand or by automatic thermostatic control through a compressed-air-operated damper.

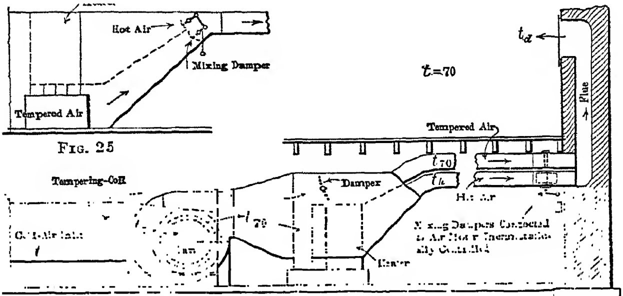


Fig. 25

Fig. 26

Figs. 25 and 26. Hot-blast Plenum Chambers

Blast Heaters

IRON PIPE-COIL HEATERS.—The standard pipe-coil heater for hot-blast work comprises 4 vertical rows of 1-in. pipe, spaced $2\,^1/_8$ to $2\,^3/_4$ in. centers, screwed into a cast-iron base. Nipples and ells cross-connect at the top the pipes in each row. The heater comprises a number of sections, each consisting of a base and its pipes, enclosed in a sheet-steel jacket, usually No. 22 gage. This type of heater has been supplanted largely by the cast-iron sectional and copper tube types. Fig. 27 shows types of heater jackets.

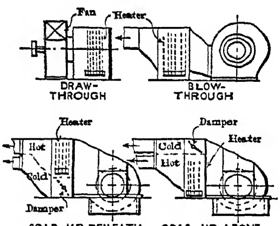


Fig. 27. Types of Heater Jackets

CAST-IRON INDIRECT HEATERS.—Special cast-iron sections for indirect heaters, such as the Vento, made by American Radiator Co., are used extensively in this class of work. A stack of several sections has fewer joints than a pipe-coil section of equal heating-surface. There is practically no deterioration of the cast-iron sectional type of heater, except for the right- and left-hand hexagonal nipples connecting the units of a stack. The four standard lengths of Vento heater-sections are indicated in Table 29, which includes other data needed in design. The Vento heater is designed for pressures not over 10 lb. per sq. in.

BRASS OR COPPER TUBE EXTENDED SURFACE HEATERS.—Some commercial heaters are built of copper tubing on which is wound a spiral copper ribbon, forming the extended surface. These are known under trade names as Aerofin, Arco Blast, Superfin, etc. This construction, due to its compactness and light weight, has largely supplanted the cast-iron sectional heater. See manufacturers' catalogs for physical data, temperature rise of air passing over various number of rows of pipe and resulting condensation for various steam pressures. These heaters also may be obtained for high-pressure work up to 350 lb. per sq. in., gage.

Table 29.—Vento Hot-blast Heater Data

Length of Section, in.	Heating-surface, Regular Units, sq. ft.	Inches on Centers		
		4 5/8	5	5 3/8
		Free Area of Unit Section, Square Feet		
40	10.75	0.52	0.62	0.72
50	13.50	0.65	0.77	0.91
60	16.00	0.78	0.92	1.08
72	19.00	0.94	1.10	1.30

TEMPERATURE RISE.—The temperature rise of air passing through blast heaters of various types has been well established by experiment. Manufacturers have published the results in the form of bulletins and catalogs. See Tables 30 and 31.

Example.—Determine size of Vento hot-blast heater necessary to heat a public building, whose calculated heat-loss H = 1,420,000 B.t.u. per hr. for 70° F. inside and 0° F. outside temperature;

Table 30.—Final Temperatures and Condensations, Vento Heaters

Regular section, standard spacing, 5-in. center to center, of loops. Steam, 5-lb., gage. C = condensation, lb. per hr. per sq. ft. of heating-surface. FT = final temperature of air leaving heater. Velocity measured through free are:

Number of Stacks Deep	Temperature of Entering Air, deg. F.	Velocity through Heater, ft. per min., Measured at 70° F.							
		1000		1200		1400		1600	
		FT	C	FT	C	FT	C	FT	C
1	20	51	1.99	49	2.23	47	2.42	45	2.56
	30	60	1.92	58	2.17	56	2.33	54	2.46
	40	68	1.80	66	2.00	64	2.16	62	2.26
	60	84	1.54	82	1.69	81	1.89	80	2.05
	70	92	1.41	90	1.54	89	1.71	88	1.85
2	20	76	1.80	72	2.00	69	2.20	66	2.36
	30	83	1.70	79	1.89	76	2.06	73	2.21
	40	90	1.60	86	1.77	83	1.93	81	2.10
	60	103	1.38	100	1.54	98	1.71	96	1.85
	70	110	1.28	107	1.42	105	1.57	103	1.69
3	20	97	1.65	92	1.85	88	2.06	85	2.22
	30	103	1.56	98	1.75	94	1.91	91	2.08
	40	109	1.47	104	1.64	100	1.79	97	1.95
	60	120	1.28	116	1.44	113	1.58	110	1.71
	70	126	1.20	122	1.34	119	1.46	116	1.57
4	20	115	1.52	110	1.73	105	1.91	101	2.08
	30	120	1.44	115	1.63	110	1.80	106	1.95
	40	124	1.35	119	1.52	115	1.68	111	1.82
	60	134	1.19	129	1.33	125	1.46	122	1.59
	70	138	1.09	134	1.23	131	1.37	128	1.49
5	20	130	1.41	124	1.60	119	1.78	114	1.93
	30	134	1.33	128	1.51	123	1.67	118	1.80
	40	138	1.26	132	1.42	127	1.56	123	1.70
	60	145	1.09	140	1.23	136	1.36	133	1.50
	70	149	1.01	144	1.14	141	1.27	138	1.40
6	20	142	1.30	136	1.49	130	1.65	126	1.81
	30	145	1.23	139	1.40	134	1.56	130	1.71
	40	148	1.15	143	1.32	138	1.47	134	1.60
	60	155	1.02	150	1.15	146	1.29	142	1.40
7	20	152	1.21	146	1.39	141	1.55	136	1.70
	30	155	1.15	149	1.31	144	1.46	139	1.60
	40	158	1.08	153	1.24	148	1.39	143	1.51
1	0	35	2.24	32	2.46	....	....	....	....
2	−20	49	2.22	44	2.46	40	2.69	37	2.92
	−10	56	2.12	51	2.35	47	2.56	44	2.77
	0	62	1.99	58	2.23	54	2.42	51	2.62
3	−20	75	2.03	69	2.28	64	2.51	59	2.70
	−10	80	1.92	75	2.18	70	2.39	66	1.60
	0	86	1.84	81	2.08	76	2.27	72	2.46
4	−20	96	1.86	90	2.12	84	2.34	78	2.51
	−10	101	1.78	95	2.02	89	2.22	84	2.41
	0	106	1.70	100	1.92	95	2.13	90	2.31
5	−20	114	1.72	107	1.95	100	2.15	94	2.34
	−10	118	1.64	111	1.86	105	2.06	99	2.24
	0	122	1.56	115	1.77	109	1.96	104	2.14
6	−20	129	1.59	121	1.81	115	2.02	110	2.22
	−10	132	1.52	125	1.73	119	1.93	114	2.12
	0	135	1.44	129	1.65	123	1.84	118	2.02
7	−20	141	1.47	134	1.69	128	1.90	122	2.08
	−10	144	1.41	137	1.62	131	1.81	126	1.99
	0	147	1.35	140	1.54	135	1.73	130	1.90
8	−20	151	1.37	144	1.58	138	1.77	133	1.96
	−10	153	1.31	147	1.51	141	1.69	136	1.87
	0	156	1.25	150	1.44	144	1.62	139	1.78

temperature T of air entering rooms to be approximately 120° F., and of air entering heater 0° F.; steam-pressure, 5 lb., gage.

Solution.—From Table 30, the number of stacks deep required for final temperature 120° and entering air 0°, using a velocity of 1000 ft. per min., is 5. Weight of air to be circulated per min. is

$$M = H/\{0.24(T - t) \times 60\} = 1{,}420{,}000/\{0.24(120 - 70) \times 60\} = 1972.$$

Free area required is $A = 1972/(0.075 \times 1000) = 26.3$ sq. ft.

From Table 29, with a 60-in. length of unit, 5 in. on centers, free area per section is 0.92 sq. ft. Number of sections, N, required across face of heater is, $N = A/0.92$, or $26.3/0.92 = 28$.

Heating-surface per section is 16 sq. ft. Total heating-surface, therefore, is $S = 5 \times 28 \times 16 = 2240$ sq. ft. and the total condensation C per hr., to be supplied by the boiler, or exhaust-steam, at 5-lb. pressure is (Table 30)

$$C = 2240 \times 1.56 = 3494 \text{ lb.}$$

Table 31.—Final Temperatures and Condensations of Aerofin Heaters

Steam at 5 lb. per sq. in. and 227° F. temperature

Temp. of Entering Air, deg. F.	Rows of Tubes Deep	Velocity of Air through Net Face Area, ft. per min., Measured at 70° F. and 29.92 in. Barometer							
		400 ft. Face Velocity. Friction per Row = 0.0337 in.		500 ft. Face Velocity. Friction per Row = 0.052 in.		600 ft. Face Velocity. Friction per Row = 0.074 in.		700 ft. Face Velocity. Friction per Row = 0.100 in.	
		Final Temp. of Air, deg. F.	Condensation, lb. per hr., per Lineal ft. of Tube	Final Temp. of Air, deg. F.	Condensation, lb. per hr., per Lineal ft. of Tube	Final Temp. of Air, deg. F.	Condensation, lb. per hr., per Lineal ft. of Tube	Final Temp. of Air, deg. F.	Condensation, lb. per hr., per Lineal ft. of Tube
0	6	144.0	1.22	138.2	1.47	133.0	1.69	127.4	1.89
	7	155.0	1.13	149.5	1.37	144.0	1.58	139.0	1.78
	8	163.8	1.04	158.7	1.27	153.6	1.47	149.0	1.66
	9	170.5	0.96	166.0	1.17	161.5	1.37	157.2	1.56
	10	175.8	.90	171.8	1.10	168.0	1.29	164.0	1.47
+20	1	54.2	1.71	52.0	2.00	49.0	2.17	48.0	2.45
	2	82.0	1.59	78.0	1.86	74.3	2.09	71.5	2.32
	3	104.6	1.44	99.5	1.69	95.5	1.92	91.3	2.12
	4	123.0	1.32	117.8	1.57	113.2	1.80	108.5	1.99
	5	138.0	1.20	132.4	1.44	127.6	1.65	123.0	1.84
	6	150.0	1.10	144.6	1.32	139.7	1.52	135.0	1.71
	7	159.8	1.02	154.7	1.23	150.0	1.42	145.5	1.61
	8	167.4	0.94	163.0	1.14	158.5	1.32	154.3	1.50
	9	173.4	.87	169.4	1.06	165.5	1.23	161.6	1.40
	10	178.0	.81	174.7	.99	171.0	1.16	167.4	1.32
+40	1	70.4	1.52	68.3	1.77	66.5	1.99	64.5	2.14
	2	95.0	1.41	91.5	1.65	88.6	1.87	85.5	2.04
	3	115.3	1.28	111.3	1.51	107.6	1.72	103.6	1.89
	4	131.5	1.18	127.0	1.40	123.0	1.60	119.0	1.78
	5	144.7	1.07	140.0	1.28	135.8	1.47	131.4	1.63
	6	155.5	0.98	151.0	1.18	146.6	1.35	142.5	1.52
	7	164.2	.91	160.0	1.10	155.7	1.27	151.6	1.43
	8	171.0	.83	167.0	1.01	163.3	1.18	159.5	1.33
	9	176.0	.77	172.6	0.94	169.2	1.09	165.7	1.24
	10	180.3	.72	177.5	.88	174.2	1.03	171.0	1.17
+60	1	86.8	1.34	84.5	1.53	83.4	1.75	81.5	1.88
	2	108.3	1.24	105.3	1.46	103.0	1.66	100.0	1.80
	3	126.0	1.12	122.4	1.32	119.5	1.51	115.8	1.66
	4	140.5	1.03	136.3	1.23	133.0	1.41	129.0	1.55
	5	152.0	0.94	148.0	1.12	144.0	1.29	140.4	1.44
	6	161.4	.86	157.4	1.03	153.6	1.19	150.0	1.34
	7	168.6	.79	165.0	0.96	161.5	1.11	158.0	1.25
	8	174.3	.73	171.0	.89	168.0	1.03	164.6	1.17
	9	179.0	.67	176.0	.82	173.0	0.96	170.0	1.09
	10	182.6	.63	180.3	.77	177.3	.90	174.6	1.03
+70	1	94.7	1.23	93.0	1.44	91.5	1.61	90.0	1.75
	2	115.0	1.16	112.3	1.36	110.0	1.54	107.4	1.68
	3	131.5	1.04	128.0	1.23	125.0	1.40	122.0	1.54
	4	144.6	0.96	141.0	1.14	137.6	1.30	134.0	1.44
	5	155.6	.87	151.8	1.05	148.2	1.20	144.8	1.34

* Net Face Area means only that area facing the tubes and does not include the headers or casing

SELECTION OF BLAST HEATERS.—The selection of a heater for a given service is based on the final temperature desired and on the net free area of pipe-coil or Vento heaters, and gross face area of copper tube heaters, required for a certain allowable velocity. That is, for specified initial and final temperatures and a given number of net sections, a certain final temperature results when the velocity has been fixed in advance. Good practice limits velocities to those given in Table 32. High velocities are objectionable in public buildings on account of the resulting noise. Resistance through the heater increases as the square of the velocity, with an increase in the power required.

Table 32.—Allowable Velocities, Ft. per Min., of Air Through Blast Heaters *

No. of stacks deep.......					
Velocity, public buildings.	1000–1500	1000–1300	1000–1200	900–1100	800–1000
Velocity, factories.......	1200–1600	1200–1600	1200–1600	1200–1500	1200–1400

* Referred to a temperature of 70° F. Allowable velocities for gross or face area of heater is approximately one-half tabular values.

RATING OF BLAST HEATERS.—The rating of an assembled heater of several sections (pipe-coil or copper tube type) or stacks (Vento type) is based on the temperature rise of the air passing over the heating surface for certain velocities through the free or unobstructed area of the heater-face. For convenience in rating, velocity is based on the volume of air at an assumed temperature of 70° F.

Let M = weight of air to be circulated through heater, lb. per min.; A = free area of heater, sq. ft.; V = veolcity of air, ft. per min., through free area based on 70° temperature; 0.075 = density of air at 70° F. Then $A = M \div 0.075V$.

9. DESIGN OF AIR DUCTS

ALLOWABLE VELOCITY OF AIR IN DUCTS AND FLUES.—To limit the resistance or pressure-loss in the duct system, velocities should be kept within the limits of Table 33. In public buildings, air should be delivered to the room at a velocity that will insure its movement to the desired points in the room without objectionable draft or noise when passing the register-grilles.

The velocity through the fan outlet, under the ordinary conditions that obtain in heating work, varies from 1500 to 2500 ft. per min.

SHEET-METAL PIPES AND DUCTS.—The recommended gage (U. S. Std. sheet-metal gage) for various sizes of galvanized sheet-steel pipes for heating and ventilating work, blowpiping and exhaust work, is given in Table 34.

PRESSURE LOSS.—The frictional resistance of air flowing through smooth sheet-metal ducts, termed pressure-loss, measured in inches of water, for 70° F. air, and for a length of duct of 100 ft., may be estimated by the formula

$$h = 0.000136 \times (R/A) \times v^2 \quad \quad [12]$$

where R = perimeter of duct, ft.; A = area of duct, sq. ft.; v = velocity of air, ft. per sec.; h = pressure-loss, in. of water-column. For round ducts, the above formula reduces to $h = 0.00055v^2/D$, where D = diam. of duct, ft.

Fig. 28 is based on this formula, and gives the diameter of a round duct for various velocities, and the pressure-loss or resistance for various quantities of air flowing.

EXAMPLE.—To find the size of a round duct to convey 1500 cu. ft. of air per min. at a velocity of 1800 ft. per min., and also the pressure-loss per 100 ft. of duct: locate 1500 on the right-hand side of the diagram, pass horizontally to the left to the intersection of the diagonal 1800-ft. velocity line. The duct nearest the required size is 12 in. diam. At this intersection pass vertically down to the base-line and read the pressure-loss of 0.48 in. of water-column.

PRESSURE LOSS OF RECTANGULAR DUCTS.—The simplest method of determining pressure-loss for rectangular ducts is to proportion the system for round ducts throughout, and transfer to rectangular sizes giving equal pressure-losses (not equal areas) by means of Table 35.

EXAMPLE.—A duct 12 in. diam. and 120 ft. long contains two 90° elbows, whose ratio of radius of throat to pipe diameter is 3; air flowing, 1500 cu. ft. per min. at a velocity of 1800 ft. per min. The total equivalent length of duct is $120 + (2 \times 4.8) = 129.6$ ft. The pressure-loss, from Fig. 28, is 0.48 in. per 100 ft. The loss is, therefore, $0.43 \times (129.6/100) = 0.62$ in. of water-column.

The pressure-loss for square elbows is $0.85v^2/2g$ in. of water-column for round pipes, and $1.25v^2/2g$ for square pipes; v = velocity, ft. per sec. The pressure-loss through register-grilles may be taken at 0.023 in. for a velocity of 400 ft. per min. through free area. The gross area of registers is about twice the free area. The pressure-loss in air-washers and humidifiers for a velocity of 500 ft. per min. through free area is 0.25 in. of water-column. The pressure-loss through Vento heaters is given in Table 37, and through Aerofin heaters in Table 31.

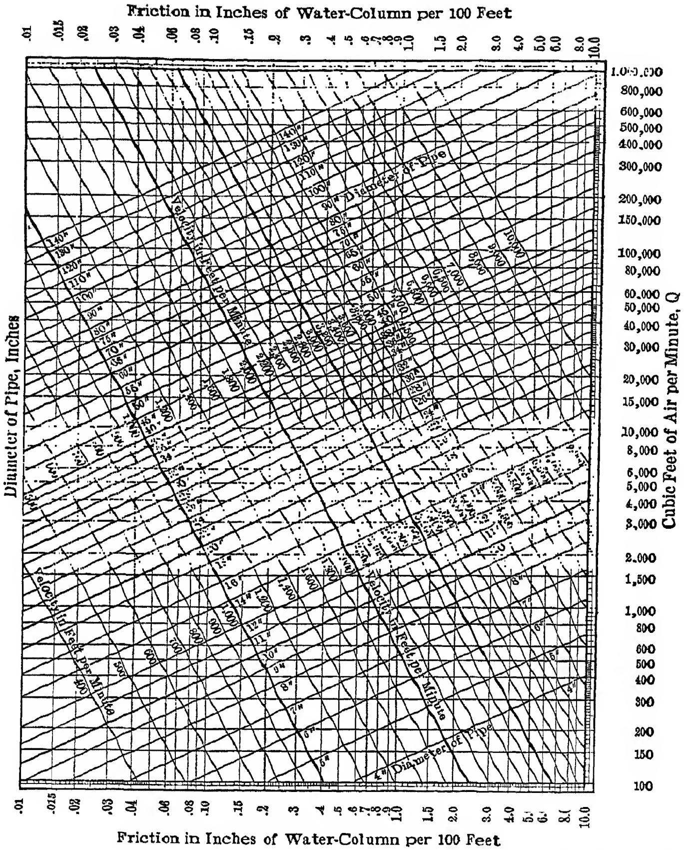


Fig. 28. Diameter of Duct for Various Velocities and Pressure Loss for Various Quantities of Air

EFFECT OF TEMPERATURE ON PRESSURE-LOSSES.—The preceding data on pressure-losses in ducts, registers and heaters are based on an air temperature of 70° F. For other temperatures, the pressure-losses are to be divided by the comparative volume of air at actual temperature to its volume at 70° F. See Table 38. For heaters use the average temperature of the air passing through the heater.

DESIGN OF DUCT SYSTEMS.—Two schemes are used to proportion air-ducts: 1. The velocity method; 2. The method of equal friction pressure-loss per ft. of length. Method 1 involves fixing the velocities (see Table 33) in the various sections, and the gradual reduction of the velocity from beginning of the duct to point of discharge. Pressure-loss is computed separately for each section having a different velocity, and the various pressure-losses are added together to obtain total pressure-loss. Method 2 is used principally in the design of duct systems for factory-heating. The velocity in the outlet farthest from the fan is fixed and the area of this branch is determined by volume of air to be delivered. Determine friction pressure-loss per 100 ft. of duct of this size by Fig. 28 and proportion remainder of the main for this same pressure-loss per 100 ft.

Example.—Method 1. In the single-duct system, Fig. 29, the risers are based on a velocity of 600 ft. per min. or 10 ft. per sec. and 400 ft. per min. or 6.6 ft. per sec., through free area of register-

Table 33.—Allowable Velocities in Fan Systems

In Public Buildings	Velocity, feet per min.	In Manufacturing Plants	Velocity, ft. per min. Employees Sitting	Employees Standing
Velocity through		Velocity through		
Free area, wall-registers.......	400– 500	Main ducts.....	1200–1500	1500–2400
Free area, floor registers.......	200– 300	Branches.......	600– 900	900–1500
Vertical flues to registers......	600– 750			
Connections to base of flues....	800–1000			
Main horizontal ducts.........	1500–2500			

Table 34.—Metal Gages for Ducts. (American Blower Corp., Detroit)

Heating and Ventilating		Thickness and Weight		Blowpiping and Exhaust Work	
Diameter, in.	U. S. Std. Gage No.	Thickness, in.	Weight, lb. per sq. ft.	Diameter, in.	U. S. Std. Gage No.
6–18	26	0.01087	0.91	3– 5	26
19–36	24	0.025	1.16	6– 8	24
38–48	22	0.0312	1.41	9–15	22
50–60	20	0.0375	1.66	16–24	20
63–72	18	0.05	2.16	26–30	18

Table 35.—Round and Rectangular Ducts of Equal Pressure Losses

Side of Rectangular Duct, in.	4	6	8	10	12	14	15	16	18	20	22	24
	Equivalent Diameters, inches											
4	4.4	.....	.....	.....	.....	.....	.....	.....	.....	.....	.....	.....
5	4.9	.....	.....	.....	.....	.....	.....	.....	.....	.....	.....	.....
6	5.4	6.6	.....	.....	.....	.....	.....	.....	.....	.....	.....	.....
7	5.8	7.0	.....	.....	.....	.....	.....	.....	.....	.....	.....	.....
8	6.1	7.6	8.8	.....	.....	.....	.....	.....	.....	.....	.....	.....
9	6.5	8.0	9.3	.....	.....	.....	.....	.....	.....	.....	.....	.....
10	6.8	8.4	9.8	11.0	.....	.....	.....	.....	.....	.....	.....	.....
11	7.1	8.8	10.2	11.5	.....	.....	.....	.....	.....	.....	.....	.....
12	7.4	9.2	10.7	12.0	13.2	.....	.....	.....	.....	.....	.....	.....
13	7.6	9.6	11.1	12.5	13.7	.....	.....	.....	.....	.....	.....	.....
14	7.6	9.9	11.5	12.9	14.3	15.4	.....	.....	.....	.....	.....	.....
15	8.2	10.2	11.9	13.4	14.7	16.0	16.5	.....	.....	.....	.....	.....
16	8.4	10.5	12.3	13.8	15.2	16.5	17.1	17.6	.....	.....	.....	.....
17	8.6	10.8	12.6	14.2	15.7	17.0	17.6	18.2	.....	.....	.....	.....
18	8.9	11.1	13.0	14.6	16.1	17.4	18.1	18.7	19.8	.....	.....	.....
19	9.1	11.4	13.3	15.0	16.5	17.9	18.6	19.2	20.4	.....	.....	.....
20	9.3	11.6	13.6	15.4	17.0	18.4	19.0	19.7	20.9	22.0	.....	.....
22	9.7	12.1	14.2	16.1	17.8	19.2	19.9	20.6	21.9	23.1	24.2	.....
24	10.0	12.6	14.8	16.8	18.5	20.0	20.8	21.5	22.8	24.0	25.2	26.4
26	10.4	13.1	15.4	17.3	19.2	20.8	21.6	22.3	23.8	25.1	26.3	27.5
28	10.8	13.5	15.9	18.0	19.8	21.5	22.4	23.1	24.6	26.0	27.3	28.5
30	11.0	13.9	16.4	18.5	20.5	22.2	23.1	23.9	25.4	26.8	28.2	29.5
32	11.3	14.3	16.9	19.1	21.1	22.9	23.8	24.6	26.2	27.7	29.1	30.5
34	11.6	14.7	17.3	19.6	21.6	23.5	24.4	26.3	26.9	28.5	30.0	31.3
36	11.9	15.1	17.7	20.1	22.2	24.2	25.1	26.0	27.7	29.3	30.8	32.2
38	12.2	15.4	18.2	20.6	22.8	24.8	25.8	26.7	28.4	30.0	31.5	33.1
40	12.5	15.7	18.6	21.1	23.3	25.4	26.4	27.3	29.1	30.8	32.4	33.9
42	12.7	16.1	19.0	21.6	23.8	25.9	26.9	27.9	29.8	31.4	33.0	34.5
44	13.0	16.4	19.4	22.0	24.3	26.5	27.5	28.5	30.3	31.2	33.7	35.3
46	13.3	16.7	19.8	22.4	24.8	27.0	28.1	29.1	31.0	32.8	34.6	36.2
48	13.5	17.0	20.1	22.8	25.2	27.5	28.6	29.6	31.6	33.4	35.2	37.0
50	13.7	17.3	20.4	23.2	25.7	28.0	29.2	30.3	32.2	34.1	35.9	37.6
52	13.9	17.6	20.8	23.6	26.2	28.5	29.6	30.7	32.9	34.7	36.5	38.3
54	14.1	17.9	21.1	24.0	26.6	29.0	30.1	31.2	33.4	35.3	37.2	38.9
56	14.3	18.2	21.5	24.4	27.0	29.5	30.6	31.7	33.9	35.9	37.8	39.6
58	14.6	18.4	21.8	24.7	27.4	30.0	31.1	32.2	34.4	36.4	38.4	40.2
60	14.7	18.7	22.1	25.1	27.8	30.5	31.6	32.7	34.9	37.1	39.1	40.9
62	15.0	19.0	22.4	25.5	28.2	30.9	32.1	33.2	35.4	37.7	39.6	41.6
64	15.1	19.2	22.7	25.9	28.6	31.3	32.6	33.7	35.9	38.2	40.2	42.2
66	15.3	19.5	23.0	26.2	29.0	31.7	33.0	34.2	36.4	38.7	40.8	42.8
68	15.5	19.7	23.3	26.5	29.4	32.1	33.4	34.7	36.9	39.2	41.4	43.4

grille. Velocity in the longest main, B, is 900 ft. per min.; volume of air to be delivered, 2000 cu. ft. per min., at 120° F. Area of riser required is 2000/600 = 3 1/3 sq. ft. = 480 sq. in. An 18 × 27-in. riser (486 sq. in. area) is used. Area of main, B, is 2000/900 = 2.22 sq. ft. = 320 sq. in. Size of duct is 12 × 27 in. The diameters of round ducts of equivalent friction-losses (Table 35) are: for riser, 24 in.; main, B, 19.5 in.

Referring to Fig. 28, pressure-losses are: Riser, 0.032 in. per 100 ft.; main, 0.09 in. per 100 ft. The main has one elbow and the riser one elbow, with ratio of radius at throat to diameter assumed as 3, in both cases. Equivalent length of main B = 200 + (4.8 × 20/12) = 208 ft.; pressure-loss = 0.09 × (208/100) = 0.187 in. Equivalent length of riser = 34 + (4.8 × 24/12) = 44 ft.; pressure-loss = 0.032 × (44/100) = 0.014 in. Pressure-loss through register-grille is assumed as 0.023 in.

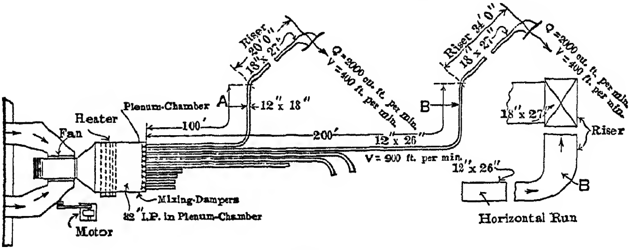


FIG. 29. Single Duct System

Total resistance of the duct system is 0.187 + 0.014 + 0.023 = 0.224 in. Assuming 5-section Vento heater to be used, with a velocity (based on free area) of 1200 ft. per min., pressure-loss through heater is 0.376 in. (Table 37.) Total resistance against which fan must operate is 0.224 in. + 0.376 in. = 0.60 in., based on 70° air. Assuming temperature of air to be 120°, resistance is (Table 38): 0.60 × (1/1.094) = 0.55 in. Method 2 is illustrated by example given under Application of Fan-blast Heating Data, below.

Table 36.—Friction Pressure Loss of 90° Elbows

(Quantities in Table Expressed in Diameters of Pipe)

Radius of throat *	1/4	1/2	3/4	1	1 1/4	1 1/2	2	3	4	5
Straight pipe of equivalent pressure loss	67	30	16	10	7.5	6	4.3	4.8	5.2	5.8

* Ratio of throat radius to diameter of pipe.

Table 37.—Friction of Air Through Vento Heaters

Friction loss, in. of water column. Vento stacks of regular section and standard 5-in. spacing of loops. Air temperature, 70° F.

Velocity, ft. per min.	One Stack	Two Stacks	Three Stacks	Four Stacks	Five Stacks	Six Stacks	Seven Stacks
800	0.037	0.070	0.103	0.135	0.167	0.200	0.232
900	0.047	0.088	0.129	0.170	0.211	0.252	0.293
1000	0.059	0.109	0.160	0.211	0.262	0.313	0.364
1100	0.071	0.132	0.193	0.255	0.316	0.377	0.438
1200	0.084	0.157	0.230	0.303	0.376	0.449	0.522
1300	0.099	0.185	0.271	0.356	0.442	0.528	0.614
1400	0.115	0.214	0.314	0.414	0.513	0.612	0.712
1500	0.132	0.246	0.360	0.474	0.588	0.702	0.816
1600	0.150	0.280	0.410	0.540	0.670	0.800	0.930
1700	0.169	0.316	0.463	0.609	0.756	0.903	1.049
1800	0.190	0.354	0.518	0.683	0.848	1.012	1.177

Table 38.—Volume of Air at Different Temperatures and Atmospheric Pressure

Deg. F.	Cu. Ft. in 1 lb.	Comparative Volume	Deg. F.	Cu. Ft. in 1 lb.	Comparative Volume	Deg F.	Cu. Ft. in 1 lb.	Comparative Volume
0	11.583	0.867	90	13.845	1.038	160	15.603	1.169
32	12.387	0.928	100	14.096	1.056	170	15.854	1.188
40	12.586	0.943	110	14.346	1.075	180	16.106	1.207
50	12.840	0.962	120	14.596	1.094	190	16.357	1.226
62	13.141	0.985	130	14.848	1.113	200	16.608	1.245
70	13.342	1.000	140	15.100	1.132	210	16.860	1.264
80	13.593	1.019	150	15.351	1.151	212	16.910	1.267

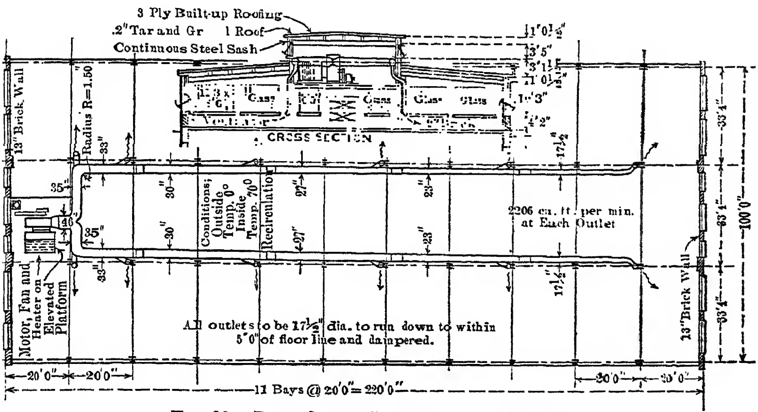


FIG. 30. Draw-through Blast System in a Factory

APPLICATION OF FAN-BLAST HEATING DATA.—The application of the preceding data may be illustrated by the design of a fan-blast draw-through heating system for the factory-building, Fig. 30; steam pressure to be 5 lb., gage. The calculated heat-loss (average inside temperature of 70° F. in zero weather) is as follows:

	Sq. Ft.		B.t.u. Transmitted per sq. ft.		Total B.t.u. per Hour.
1 5/8-in. roof	22,000	×	31.*	=	682,000
9-in. brick walls	1,830	×	25.	=	45,760
13-in. end walls	2,100	×	19.7	=	41,370
Side wall monitor	880	×	30.	=	26,400
Glass surface	7,300	×	79.	=	576,100
7/8 air-change per hr., cu. ft.	354,800	×	1.26	=	390,287
			Total	=	1,761,917 = H

Heater Calculations.—A heater made up of five 60-in. Vento stacks will be used. Air will be recirculated and will enter heater at an assumed temperature 10° F. lower than the average inside temperature, or (70 − 10) = 60° F. It will pass through free area of heater at a velocity of 1200 ft. per min., measured at 70° F. From Table 30, final temperature of air leaving heater will be 140° F. and condensation per sq. ft. of heating surface will be 1.23 lb. per hr.

M = 1,761,917 ÷ {60 × 0.24 (140 − 70)} = 1747 lb. of air to be circulated per minute, or from Table 38, 1747 × (15.1 ÷ 1.132) = 23,215 cu. ft. of air per min., measured at 70° F.

A = 23,215 ÷ 1200 = 19.34 sq. ft. of free area through heater (Table 32).
N = 19.34 ÷ 0.92 = 21 sections, width of heater (Table 29).
S = 21 × 5 × 16 = 1680 sq. ft., total heating surface (Table 29).
C = 1680 × 1.23 = 2066 steam condensed per hour (Table 30).

Design of Duct System.—The duct system in factory work ordinarily is designed on a basis of equal pressure loss per foot of length. Total air volume, measured at 140° F., is 1747 × 15.1 = 26,379 cu. ft. per min. (Table 38), and volume of air to be delivered by each of the 12 outlets is 26,379 ÷ 12 = 2198 cu. ft. per min. With an assumed discharge velocity at outlets of 1350 ft. per min. (Table 33), required area per outlet is 2198/1350 = 1.6 sq. ft., equivalent to 17 1/2 in. diam.

From Fig. 28, pressure-loss per 100 ft. of length of the last section of the 17 1/2-in. duct, discharging 2198 cu. ft. per min., is 0.20 in. of water-column. Since the duct system is to be designed for equal pressure-loss per foot of length, the remaining sizes of duct are determined by the intersection of the vertical 0.20-in. pressure-loss line with the horizontal cu. ft. lines for the quantities of air to be handled by the various sections of the duct. Each outlet should have a damper to equalize and adjust the flow of air. Total pressure-loss of system is estimated as follows:

Loss through heater	0.376 in. of water-column.	
" " 200 ft. pipe, 2 × 0.20	0.400 "	"
" " two 35-in. elbows	0.072 "	"
" " one 17 1/2-in. elbow	0.018 "	"
Total	0.866 "	"

The actual pressure-loss will be somewhat less than the above total, as the figures are based on

* Temperature of air, under side of roof a sumed to be (70 + = 82°. B.t.u. per sq. ft., U, = 0.38 × 82 = 31. (See Table 6.)

a density corresponding to an air temperature of 70° F. Usually no correction is made in practice for this difference.

The rated capacity of the fan required (see tables under Fans and Blowers, p. 1–63) for this installation is 26,379 ÷ 1.06 = 24,886 cu. ft. per min. (Table 39) at 7/8-in. static pressure or maintained resistance. The speed and horsepower as given by the fan tables should be multiplied by the factor 1.06.

RATING OF FANS.—The volume of air, cu. ft. per min., at 70° F., which a fan will deliver, varies with the resistance against which it operates. To correctly choose a fan from manufacturers' tables (see Fans and Blowers), the resistance (static pressure) must be determined by the duct-design, after the size of heater, and its resistance, have been determined. The speed and brake horsepower required to drive the fan also are stated in the tables. The temperature of air handled by the fan with draw-through apparatus is higher than 70°, except for a fan which is connected ahead of a tempering-coil, usually a heater 2 sections deep. The tabulated speed, volume and brake horsepower to maintain the pressure must be multiplied by the factors given in Table 39, for temperatures other than 70°. The factors in this table are the square roots of the ratios of the density of the air at 70° F. to its density at the temperature stated.

Table 39.—Factors for Speed, Volume and Brake Horsepower of Fans

Temperature, deg. F.....	0	100	120	130	140	150
Factor................	0.932	1.028	1.046	1.055	1.064	1.073

SELECTION OF MOTOR FOR FAN DRIVING.— Good practice will add 15% to the brake horsepower, as determined from the fan tables, for the rating of the motor. This allows for overload, due to fan being operated under somewhat different conditions of pressure and speed than those under which it was originally rated.

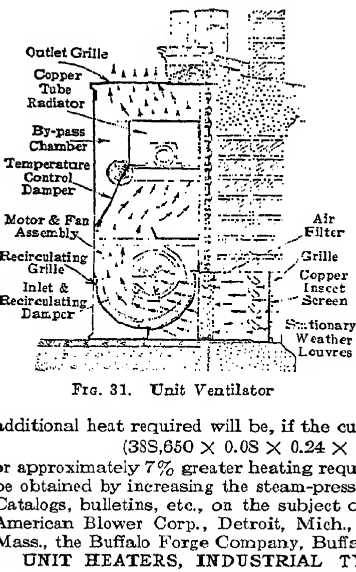


Fig. 31. Unit Ventilator

Fan Engine.—When high-pressure steam is available, an automatic high-speed engine often is used for fan-driving. The exhaust from the engine is used in the first section of the heater.

ADDITIONAL HEATING REQUIREMENT.—It may be desirable to so proportion heating-apparatus that the fan may be stopped at night and started about two hours before the shop is opened in the morning. It may be safely assumed that the temperature of the air in the building will not be below 30° F. when the fan is started, and that the air is all recirculated. Fan and heater must be of sufficient capacity to make up the heat loss from the building, including infiltration, and also to heat the contained air from 30° to 60° F. in two hours. Assuming the same data as given in the preceding example, the additional heat required will be, if the cubic contents of the building are 388,650 cu. ft.

$$(388{,}650 \times 0.08 \times 0.24 \times 30) \div 2 = 119{,}931 \text{ B.t.u. per hr.}$$

or approximately 7% greater heating requirement than previously calculated. This may be obtained by increasing the steam-pressure in the heater to approximately 10 lb., gage. Catalogs, bulletins, etc., on the subject of fan-blast heating may be obtained from the American Blower Corp., Detroit, Mich., the B. F. Sturtevant Company, Hyde Park, Mass., the Buffalo Forge Company, Buffalo, N. Y., and other manufacturers.

UNIT HEATERS, INDUSTRIAL TYPE.—Propeller-fan type unit heaters have outlet velocities ranging from 300 to 600 ft. per min. They ordinarily are suspended from the ceiling or structural framework of the building, at intervals of from 60 to 100 ft. The floor-type unit heater discharges warmed air through one or several outlets at velocities of 1000 to 1700 ft. per min. Some types will project and distribute the heating effect over a distance of approximately 200 ft. Most unit heaters are built with extended-surface copper tube heating surface. Unit heaters may be distributed through the central portion of a room, discharging toward the exposed wall surfaces, or around the walls, discharging at an angle with the walls toward the interior of the room. The discharge from the heaters should be so directed that rotational circulation of all the air in the room is set up by the discharge from the heaters.

The usual capacity rating of unit heaters is B.t.u. per hr. heating effect, based on recirculation of air in the room. Some makers state air volume delivered and temperature rise. The number of uniform-size unit heaters required in a room depends primarily

Table 40.—Capacities of Unit Heaters*

(American Blower Corp., Detroit, 1935)

Size		Motor Hp.					Size			Cu. ft.			
1–18	502.5		160	.025	120.5	124.5	2–30	2130		8,000	511.0	125.0	529
		8	720	1256	86.8	90.0				6,350	437.3	131.0	452
2–18	1002	1	1160	4050	241.0	249.0	3–30	3195		12,000	767.0	125.0	793
		1/4	720	2512	173.5	180.0				9,525	656.0	131.[illegible]	678
3–18	1505	1 1/2	1160	6068	361.5	374.0	1–38	1757		6,850	422.0	122.0	436
		1/2	720	3770	260.3	269.0				5,450	362.5	127.0	375
4–18	2008	2	1160	8100	482.0	498.0	2–38	3410		12,450	819.0	126.5	846
		1/2	720	5024	347.0	359.0			1 1/2	9,870	692.5	131.5	716
1–30	1065	1	870	4000	255.5	264.0	3–38	5280		19,275	1267.5	126.5	1310
		1/2	690	3175	218.6	226.0				14,175	1016.5	133.2	1051

* Heater fitted with Class IV convector; for Class III convector multiply B.t.u. by 0.95; for Class II convector multiply B.t.u. by 0.89.

† Basic rating, based on steam at 2 lb., entering air temp., 60° F.

‡ To obtain B.t.u. capacity for conditions other than 2 lb. steam and 60° F. entering air, multiply B.t.u. as given in Table 40 by factor from Table 41.

Table 41.—Conversion Factors for Unit Heaters

(For Application to B.t.u. Values of Table 40)

Temp. Entering Air, deg. F.	Steam Pressure, lb. per sq. in., gage													
	0	2	5	10	15	20	30	40	50	60	70	80	90	100
0	1.405	1.442	1.485	1.558	1.610	1.649	1.725	1.787	1.850	1.897	1.943	1.988	2.028	2.071
10	1.329	1.363	1.410	1.480	1.533	1.572	1.648	1.710	1.773	1.820	1.869	1.914	1.951	1.994
20	1.253	1.290	1.334	1.403	1.458	1.498	1.572	1.637	1.700	1.748	1.795	1.841	1.878	1.919
30	1.178	1.215	1.260	1.328	1.382	1.421	1.497	1.563	1.628	1.673	1.722	1.770	1.804	1.845
40	1.105	1.141	1.187	1.253	1.310	1.350	1.423	1.491	1.554	1.601	1.651	1.698	1.732	1.770
45	1.070	1.105	1.150	1.219	1.274	1.313	1.387	1.455	1.520	1.565	1.617	1.663	1.697	1.735
50	1.032	1.069	1.114	1.182	1.239	1.278	1.352	1.420	1.483	1.531	1.582	1.629	1.661	1.700
55	0.997	1.035	1.080	1.148	1.204	1.243	1.318	1.385	1.450	1.500	1.548	1.595	1.625	1.665
60	.962	1.000	1.045	1.112	1.168	1.208	1.281	1.350	1.416	1.463	1.512	1.560	1.590	1.630
65	.925	0.955	1.010	1.078	1.133	1.172	1.247	1.318	1.382	1.430	1.478	1.525	1.557	1.595
70	.892	.930	0.975	1.042	1.099	1.138	1.212	1.282	1.347	1.394	1.443	1.491	1.523	1.560
75	.858	.895	.940	1.008	1.063	1.104	1.179	1.249	1.312	1.360	1.410	1.458	1.490	1.527
80	.822	.861	.906	0.973	1.028	1.070	1.145	1.215	1.278	1.325	1.377	1.422	1.457	1.492
85	.788	.825	.872	.938	0.993	1.035	1.112	1.181	1.245	1.293	1.343	1.389	1.421	1.460
90	.754	.792	.838	.903	.960	1.002	1.078	1.148	1.211	1.260	1.310	1.354	1.387	1.425
100	.688	.728	.771	.838	.895	0.936	1.010	1.081	1.145	1.194	1.243	1.288	1.321	1.359
110	.621	.662	.706	.772	.828	.870	0.945	1.017	1.080	1.130	1.178	1.223	1.253	1.292
120	.556	.598	.641	.708	.763	.807	.882	0.952	1.015	1.065	1.113	1.158	1.189	1.227
130	.493	.531	.577	.643	.701	.742	.820	.893	0.952	1.000	1.050	1.094	1.123	1.162

on the spacing necessary to obtain uniform heating. The required B.t.u. capacity of each heater is the calculated heat loss of the room divided by number of heaters to be installed. Capacities of unit heaters as made by one manufacturer are given in Table 40. Table 42 shows dimensions of several types.

Example.—A suspended-type unit heater installation for the factory building shown in Fig. 30 conveniently could consist of ten units, in two groups of five in each side bay near the outside walls. Required capacity of each unit = 1,761,917 ÷ 10 = 176,192 B.t.u. per hr. Air is to be recirculated at 70° F. Steam pressure, 5 lb. The nearest size of unit (Table 40) is a 2–18 unit, 2512 cu. ft. per min. delivery, 173,500 B.t.u. per hr., 1/4-Hp. motor; multiplying by factor 1.01 from Table 41 for 5-lb. steam and 70° air, 173,500 × 1.01 = 175,235 B.t.u. per hr.

UNIT VENTILATORS.—A considerable number of unit heaters, suitable for use in schools and public buildings ordinarily are designed for the ventilating requirement only, with sufficient direct radiation to provide for the calculated heat loss of the walls. The combination is termed a split system. The unit ordinarily is designed to heat to approximately 80° F. outside air drawn into the apparatus.

Unit ventilators ordinarily are placed under a window. They comprise a small ventilating fan, fan motor, and indirect heating surface of either cast-iron or extended-surface

Table 42—Dimensions of Unit Heaters
(American Blower Corp., Detroit, 1935)

Size	No. of Fans and Outlets	Total Area of Outlets, sq. ft.	Overall Dimensions, in.									Pipe Sizes, in.		Overhang of Motor Bracket, in.
			Floor Type			Wall Type			Ceiling Type			Steam Supply	Drip	
			High	Wide*	Deep	High	Wide*	Deep	High	Wide*	Long			
1–18	1	1.0	89	36	18 1/8	65	36	18 1/8	18 1/8	36	79	2	2	18
2–18	2	2.0	89	54	18 1/8	65	54	18 1/8	18 1/8	54	79	2	2	20
3–18	3	3.0	89	72	18 1/8	65	72	18 1/8	18 1/8	72	79	2	2	22 1/2
4–18	4	4.0	89	90	18 1/8	65	90	18 1/8	18 1/8	90	79	3	2	24
1–30	1	1.5	110 1/2	46	30 1/2	82 1/2	46	30 1/2	30 1/2	46	103	3	2	22 1/2
2–30	2	3.0	110 1/2	70	30 1/2	82 1/2	70	30 1/2	30 1/2	70	103	3	2	26
3–30	3	4.5	110 1/2	94	30 1/2	82 1/2	94	30 1/2	30 1/2	94	103	3	2	28 1/2
1–38	1	3.05	130	53	38 3/8	99	53	38 3/8	38 3/8	53	135	3	2	......
2–38	2	6.10	130	89	38 3/8	99	89	38 3/8	38 3/8	99	135	3	2	......
3–38	3	9.15	130	124	38 3/8	99	124	38 3/8	38 3/8	124	135	3	2	......

* Exclusive of motor bracket.

copper coils, all enclosed in a finished sheet steel casing, with an outside and recirculating intake. Each intake has a damper, and all dampers usually are so connected that the full opening of one closes the other. The fan draws in either all outside air, all recirculated air, or a mixture of the two as may be desired by the manipulation of the dampers. The fan discharge may be by-passed around the heater, all passed through the heater, or a portion only by-passed by means of a damper ordinarily under thermostatic control. The temperature of the air leaving the unit is controlled by the by-pass damper, the volume of air leaving the unit remaining constant.

HEATING BY ELECTRICITY

The relations involved, and the underlying principles of electrical heating are expressed by the following equations:

$$I = E/R;\ R = kl/a;\ W = I \times E = I^2R;\ H = 3.415\,W = 3.415\,I^2R \text{ per hr.},$$

where I = current flowing, amperes; E = electromotive force or potential difference, volts; R = resistance of conductor, ohms; k = coefficient of specific resistance, ohms per sq. in. per ft. of conductor; l = length of conductor, ft.; a = area of cross-section, sq. in.; W = watts = volts × amperes; H = heating effect, B.t.u. per hr.

It is evident that the heating value of a conductor varies as the square of the current flowing and directly with the resistances. For given potential differences, the current varies inversely as the resistance; hence, increasing the resistance only, in such a case, actually would cut down the heat supplied. The resistance varies with the material, with the length of conductor, and inversely as the area of the conductor. For a conductor of given material and length, subjected to a definite voltage, there is a certain diameter and corresponding resistance which will give the maximum heating effect, which can be determined from the above equations.

The economy of electrical heating depends entirely on the cost of generating and distributing the electricity, as all the energy supplied, if the heater is of the direct type, is converted into heat at the radiator. In addition to this, account must be taken of the simplicity, additional convenience, reduced attendance and repairs, and small space occupied by the electrical equipment. The actual money value of these factors is hard to determine. Any comparison with other forms of heating usually is based on heating effect obtained per pound of coal burned, or per dollar paid for fuel and electricity. Only when electricity can be generated at a very low cost, as in certain hydro-electric plants, can it be used for heating on a basis of equal cost compared with the ordinary steam or hot-water system of heating.

Example.—A house heating boiler burns coal of 12,000 B.t.u. per lb. with 60% efficiency. The coal required to give the heating effect of 1 kw.-hr. is $3415/(0.60 \times 12{,}000) = 0.47$ lb. The cost of 1 kw.-hr. in a turbo-generating station with 90% combined mechanical efficiency is $1.00/0.90$ or 1.111 kw.-hr. for each kw. supplied for heating. With a generating set thermal efficiency of 15%, the coal required at 60% efficiency to produce 1 kw. for electrical heating is

$$(1.111/0.15) \times \{(3415/0.60 \times 12{,}000)\} = 3.5 \text{ lb.}$$

The relative cost of fuel at the same price per ton is, therefore, $3.5/0.47 = 7.5$ times as much when electrical generation by steam is employed.

VENTILATION

VENTILATION is the displacement of vitiated air from an inhabited room and its replacement with fresh air. It usually is expressed in the number of changes of air per hour. This is not strictly correct, as there is no positive change; the incoming air dilutes the foul air until it is suitable for respiration.

VENTILATION LAWS.—Many states have laws stating the amount of ventilation to be supplied to public and semi-public buildings, temperature limits, etc. These codes, and those of the various cities involved should be consulted before completing the design of any heating and ventilating system. Most states require 30 cu. ft. per min. per occupant of outside air to be introduced into school rooms to provide ventilation.

SYSTEMS OF VENTILATION.—Ventilation systems are classified as: 1. Natural ventilation in which the movement of air in flues, ducts, etc., is induced by the thermal head produced by the difference in density due to the difference in temperature between the column of air in the ducts and that of the outside atmosphere. 2. Mechanical ventilation in which the movement of air is maintained by a positively driven fan. With the mechanical system, air at any desired temperature may be positively circulated to all parts of the building, giving practically uniform ventilation, regardless of outside weather conditions. For data regarding fan capacities in mechanical ventilating systems, see Blast Heating, p. 11–33, and Fans and Blowers, p. 1–63. For data on capacity of ventilators in natural ventilation, see Table 2.

Ventilation systems also are divided into: *a*. Upward systems; *b*. Downward systems. Upward systems frequently are used for audience rooms. Air is supplied near the floor line through mushroom ventilators in the floor, through hollow pedestals in the chairs, or through floor registers. Vitiated air outlets are in the wall near the ceiling. The downward system generally is used in school rooms, hospitals, institutions, etc., where the occupants are not closely spaced. Air enters at 8 ft. or more above the floor; the vitiated air outlet usually is at or near the floor line. Inlets and outlets should be placed if possible in the same inside wall, the former being at least 7 or 8 ft. above the latter. Incoming air flows out across the room to the cold outside wall before it cools and drops to floor level. Practically uniform distribution of fresh air throughout the room thus is obtained. Theaters often are ventilated by a downward system, with the fresh air outlets located in the ceiling. See p. 11–60. Neither upward nor downward system can be characterized as superior. Each has its proper place and limitations.

REQUIREMENTS FOR GOOD VENTILATION.—Table 1 represents good ventilation practice as regards the amount of outside or new clean air to be introduced into rooms used for various purposes.

Table 1.—Amount of New Air to be Supplied per Person, with and without Air Conditioning

(A.S.H.V.E. *Guide*, 1928)

	Not Humified or Recirculated	Humidified or Dehumidified Not Recirculated	Humidified, Dehumidified and Recirculated	Number of Air Changes per Hour
	Cubic feet per minute			
Schools				
Class rooms	30	20	10 to 15	
Assembly rooms	15 to 25	10 to 15	10 to 15	
Gymnasiums	30	25	15 to 20	
Toilets				10 to 20
Locker rooms				5 to 10
Kitchens				20 to 60
Lunch rooms				10 to 20
Theaters, seating space	30 to 50	20 to 30	10 to 15	
Hospitals				
Wards	40 to 60	30 to 40		
Kitchens				20 to 60
Dining rooms				10 to 20
Toilets				10 to 20
Hotels				
Dining rooms				10 to 20
Kitchens				20 to 60
Ball rooms	20 to 30	15 to 20	10 to 15	5 to 10
Work space	....	....	....	5 to 10
Assembly rooms	20 to 30	15 to 20	10 to 15	

DISTRIBUTION OF THE AIR.—In general, with upward or downward ventilation, definite provision to remove vitiated air should be made to insure uniform distribution and prevent short circuiting between air inlets and outlets. Multiple inlets and mushroom ventilators, Fig. 1, are used to give a better mechanical distribution of air in systems of upward ventilation in audience rooms with fixed seats. Air is supplied through a false floor or plenum chamber below the main floor. The mushroom ventilator heads, located under every second or third seat, are adjusted for a uniform discharge of tempered air over the entire seating area. The heads are mounted either on an adjustable spindle as shown, or on a non-adjustable spindle with a control damper. In either case, the adjustable head or the damper must be locked in the final adjusted position.

AUTOMATIC VENTILATORS.—The chief differences in the various types of automatic ventilators for natural ventilation lie in the degree to which the wind is utilized to produce additional ventilation. Four general classes of such ventilators are: 1. Stationary non-siphoning type; 2. Stationary siphoning type; 3. Rotary plain head type; 4. Rotary siphoning type.

Type 1 does not contemplate utilizing the wind to produce additional ventilating effect. Type 2 uses the wind to produce additional ventilating effect, by an ejector action. When there is no wind, ventilation is due to natural circulation.

The rotary or revolving head types (3 and 4) have a protecting cowl revolving on a central bearing. An opening in one side is always directed in the path of the wind through the movement of the cowl. The velocity of the wind creates a region of decreased pressure at the cowl opening which in turn creates additional ventilation.

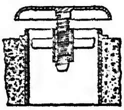


Fig. 1. Mushroom Ventilator

Average results of a series of tests conducted in 1920 by the Engineering Experiment Station of the Kansas State Agricultural College on different types of 10-in. automatic ventilators are given in Table 2. The object of the investigation was to determine the degree to which the various types utilized wind in producing additional ventilation. Approximate wind conditions were duplicated by a wind tunnel in which were produced wind velocities up to 10 miles per hr. Air velocities in tunnel and in ventilator were measured by an anemometer. The temperature within the tunnel and in the room ventilated being identical, there was no natural circulation of air, and all movement of air through the ventilator was produced by the wind currents. Actual installations of ventilators would obtain in most cases greater ventilation because of the influence of natural circulation. The amount of ventilation secured by natural circulation may be calculated, if the density of the internal and external air and the height of the ventilator above the floor are known. The tests indicated that the various types have certain well-defined advantages, but for all practical purposes little difference exists between ventilators differing slightly in design but of the same general type

EFFECT OF VITIATED AIR.—The presence of carbon dioxide, according to Dr. Ira Remsen, is not in itself dangerous to health except that it reduces the supply of oxygen by displacing it. Carbon dioxide is non-poisonous, but organic impurities exhaled with it may be a menace to health. The breathing of small quantities of decomposing organic matter and unhealthful gases are the dangers of poor ventilation. The carbon dioxide given off at the same time as the other impurities is more or less an indicator of the presence of real danger. A lowering of the oxygen supply below the point required by good ventilation causes lowered vitality and decreased production of industrial workers. Satisfactory ventilation consists in constantly supplying fresh air, free from dust and other impurities, at the proper temperature and with the proper moisture content and also in

Table 2.—Air Velocities Through Automatic Ventilators Produced by Various Velocities

Wind Velocity, miles per hr.	Velocity of Air in Ventilator, ft. per min.				Wind Velocity, miles per hr.	Velocity of Air in Ventilator, ft. per min.			
	Stationary Non-siphoning Type	Stationary Siphoning Type	Rotary Plain Head Type	Rotary Siphoning Type		Stationary Non-siphoning Type	Stationary Siphoning Type	Rotary Plain Head Type	Rotary Siphoning Type
1	27	31	46	52	7	192	215	321	361
2	55	62	92	104	8	220	246	367	413
3	83	93	137	155	9	248	277	413	465
4	110	123	183	207	10	276	307	459	516
5	137	154	229	258	11	304	338	505	570
6	165	185	275	310	12	332	368	552	621

removing vitiated air. This can be accomplished satisfactorily during the heating system only by mechanical means.

CAUSES OF AIR VITIATION.—Air is vitiated by: 1. Excess of carbon dioxide due to respiration, lights, etc. 2. Generation of heat by occupants, lights, etc., above that required for warming. 3. Water vapor in excess of or below that required for desired relative humidity. 4. Dust, either brought in by the air or carried by persons within the building. 5. Bacteria, due to respiration or otherwise. 6. Odors, due to occupants or to manufacturing processes.

The composition of pure air and respired air has been given by Prof. R. C. Carpenter as follows:

	O	N	CO_2	Water Vapor
Pure air..........	20.26%	78.00%	0.4%	1.50% (variable)
Respired air.......	16.2%	75.00%	4.0	5.0%

The respired air is at a temperature of 90° to 98° F. and is nearly saturated with water vapor.

The quantity of air required for each adult in order to maintain a CO_2 content of 7 parts, outside air containing 4 parts, per 10,000, is given by $C = (a \times b)/(n - 4)$, where C = cu. ft. per min., a and n = respectively, parts of CO_2 per 10,000 for respired air, and air of standard purity, and b = cu. ft. per min. of CO_2 per person = 0.25 ordinarily. See Table 3.

RELATION BETWEEN HUMIDITY AND TEMPERATURE.—Air, on being heated, has its capacity for absorbing moisture greatly increased, giving the sensation due to so-called "dry" heat. This causes excessive and unnatural evaporation of moisture from the skin and respiratory organs, which lowers the surface temperature of the body and causes a temperature sensation several degrees lower than the actual temperature of the room. The remedy for too low or too high humidity in a room is the introduction or decrease of moisture present in the air.

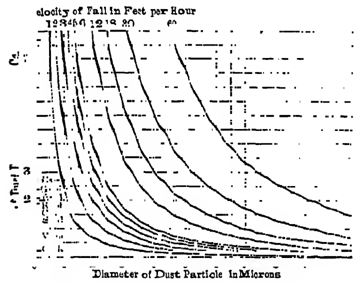


FIG. 2. Velocity of Fall of Dust

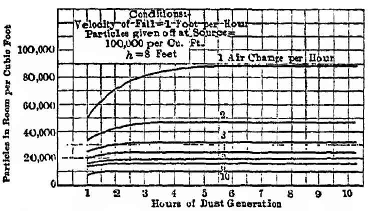


FIG. 3. Effect of Air Chambers on Dust Content

The relation of temperature and relative humidity to human comfort has been the subject of considerable research by the A.S.H.V.E. The results of this research have been published from time to time in the transactions of this society, to which the reader is referred. See A.S.H.V.E. 1932 *Guide* for comfort charts.

DUST.—Determination of dust includes both the amount actually present in a given volume of air and the examination as to its character, such as abrasive, metallic or fibrous. Soluble filters of pure sugar are used to screen out dust particles and the sugar then is dissolved.

O. W. Armspach (*Jour.* A.S.H.V.E., Dec., 1920) reports an investigation and tests carried out by the Research Bureau of the A.S.H.V.E., and the U. S. Bureau of Mines, with the following conclusions: The dust content in a room depends on: 1. Density of material, velocity of fall of particles and size of the room. The dust will accumulate until a definite content is reached, depending on the rate of fall and the number of air

Table 3.—Values of C for Various Values of n from 5 to 30

n = Vitiation allowed. Parts of CO_2 per 10,000 in the room air.
C = Cu. ft. of air containing 4 parts of CO_2 per 10,000 to be supplied per person or equivalent. A person = 0.6 cu. ft. CO_2 per hr.

n =	5.0	5.5	6.0	6.5	7.0	7.33	7.5	8.0	9.0	10.0	15.0	20.0	30.0
C =	100.0	66.6	50.0	40.0	33.3	30.0	28.6	25.0	20.0	16.6	9.1	6.2	3.8

changes. 2. The total dust given off by various machines when handling different materials can be determined from the average count of dust particles resulting in the room. 3. Dust conditions can be improved by providing the proper number of air changes. There is, however, a limit to this number, and usually 5 changes per hour are sufficient. 4. For equal conditions of air dustiness, as the density of the material increases, the weight per cu. ft. decreases. All dust determinations should be on the basis of the number of particles per cu. ft. From a formula developed in the investigation, the chart, Fig. 2, was prepared, showing the velocity of fall for various kinds of dust. This chart, for example, shows that a particle of 2 microns diameter and a density of 8 will fall at the same rate as a particle of a diameter 5.75 microns and a density of 1. Another formula developed shows the effect of change of air on dust content. The curves, Fig. 3, for dust of 2 microns diameter falling with a velocity of 1 ft. per min. and based on 100,000 particles per cu. ft. given off at the source, are plotted from this formula.

AIR FILTERS.—The use of some form of air filter is becoming a recognized necessity for the removal of dirt and dust from the air in a heating and ventilating system, or for the air supply to air compressors, internal combustion engines, etc. Air filters often are used in conjunction with air washer equipment when the removal of fine dust particles is considered important. They are distinguished from air washers in that they remove dirt and dust without using a water spray, and hence, do not change temperature or relative humidity of air passing through the filter.

All types of filters should fulfill the following requirements: 1. Efficient in dirt removal. 2. Interpose low resistance to air flow. 3. Possess large dust holding capacity. 4. Easy to clean and handle. Air filters are classified as: 1. Dry filters; 2. Viscous filters.

Dry Filters were the first type to be used. They comprise a fine mesh cloth, felt or paper screen through which the air is filtered or strained. Several types of dry air filters are available. They are cleaned by rapping, by reversing the air flow, or by supplying new filter areas.

Viscous Filters have largely superseded the dry type. Their cleaning action depends on the dirt impinging on a metal surface coated with a viscous fluid. All filters of this type operate on the principle of adhesive impingement. The viscous material employed should be odorless, fireproof and germicidal in its action. This type of filter is constructed in units of various standard sizes so that practically any area of filtering surface may be obtained by the use of one or more units.

AIR CONDITIONING

AIR CONDITIONING generally is understood to mean the simultaneous control of temperature, relative humidity, air motion, air distribution and ventilation within an enclosure. Air conditioning systems are used in theaters, churches, auditoriums, schools, restaurants, offices, homes, etc., to produce or effect comfort for occupants by maintaining a temperature and relative humidity which will lie in the so-called "comfort-zone." The comfort zone for both winter and summer has been established by the A.S.H.V.E. Research Laboratory. While the art of air conditioning properly may be regarded as a recent development, its principles are not new. A system embodying many of the principles now used was proposed for the U. S. Army hospitals by G. H. Knight in 1864.

Air conditioning systems are used in industry to maintain the temperature and relative humidity that is most desirable for the process involved. For many industrial processes the most desirable temperature and relative humidity has been fairly well established. See A.S.H.V.E. *Guide*, 1934.

The *addition* of moisture to air being circulated is humidification; the *removal* of moisture is dehumidification.

HUMIDIFICATION usually is effected by passing air through a finely-divided water spray, in an air washer. This consists of a sheet-metal housing, enclosing one or more banks of spray nozzles. The air passes through the spray and then over bent plates, called eliminators. The latter abruptly change the direction of flow and remove free water held in suspension. Larger dust particles also are removed in the washer, but for complete dust elimination an air filter must precede the washer.

In some industrial plants, a steam jet is used for humidification. When both humidification and dehumidification are required for year-round operation an air washer is used. Recirculated spray water is heated for humidification in winter, and cooled for dehumidification in summer.

Surface-type coolers are used quite generally for summer comfort cooling. Cold water is circulated through the coils of the cooler, or a refrigerant, as freon, methyl chlor-

ide, etc., is evaporated in the coils. When temperature of cooling medium is lower than temperature of air entering cooler, dehumidification results. Artesian well water, if available, may be used as a cooling agent. Artificial refrigeration, however, is usual for comfort cooling.

HUMIDITY is the water vapor (steam or moisture) mixed with air. The maximum weight of vapor which a given enclosure will contain depends only on the temperature (see the Steam Tables), regardless of the presence or absence of any other vapor or gas. That is, the weight of vapor is exactly the same whether air is present or not. The pressure of the vapor is in accordance with Dalton's Law (see p. 1-08), and depends only on the temperature, irrespective of the presence of air.

Saturated Air.—Air is saturated when it has mixed with it the maximum possible amount of vapor, which amount varies with temperature. The vapor itself under this condition also is saturated (quality $x = 1$). If the air is not saturated, the contained vapor is superheated. The *actual humidity* of the air, in meteorological work, is the number of grains (1 lb. = 7000 grains) or pounds of water vapor contained in 1 cu. ft. of a mixture of air and vapor at the observed temperature.

Relative Humidity, or degree of humidity, is the actual amount of moisture (grains or pounds) contained in 1 cu. ft. of the mixture divided by the amount which 1 cu. ft. of the mixture would hold at the same temperature if saturated. This condition is stated as percent relative humidity.

A common interpretation of this term in engineering computations, is the ratio of actual weight (lb.) of vapor mixed with 1 lb. of dry air to the weight (lb.) of vapor mixed with 1 lb. of dry air in a saturated mixture at the same temperature.

Wet- and Dry-bulb Temperatures are the observed temperatures as indicated by wet- and dry-bulb thermometers respectively.

Dew-point Temperature is the temperature corresponding to saturation (100% relative humidity) for a given weight of vapor. Any lowering of temperature produces a contraction of volume and partial condensation. Air with any amount of vapor or relative humidity has a dew-point, as temperature may be lowered to a temperature where condensation begins.

EXAMPLE.—Required the weight of vapor carried by 1 lb. of air in a saturated air-vapor mixture at 60° F. and atmospheric pressure (14.7 lb. per sq. in., or 29.92 in. Hg, absolute, at sea-level). *Solution.*—Let p and p_1 = respectively, atmospheric and absolute partial pressures of air; p_2 = absolute partial pressure of the vapor corresponding to the temperature; all pressures in lb. per sq. in. Then $p = p_1 + p_2 = 14.7$ lb. at sea level.

From the steam tables, at 60° F., $p_2 = 0.26$; density = 0.000828 lb. per cu. ft.; $p_1 = 14.70 - 0.26 = 14.44$. From the relation $PV = MRT$ (see p. 3-75), where $R = 53.35$ for air, $T = 459.6 + 60 = 519.6$, $P = 144 \times 14.44$, $M = 1$,

$$V = (53.35 \times 519.6)/(144 \times 14.44) = 13.33 \text{ cu. ft.} = \text{vol. of air.}$$

As the air and vapor occupy the same space, the volume of vapor also is 13.33 cu. ft., whence the weight of saturated vapor in the mixture is $13.33 \times 0.000828 = 0.01104$ lb. Weight of vapor per cu. ft. of mixture = $0.01104/13.33 = 0.000828$ lb. = $0.000828 \times 7000 = 5.796$ grains.

TOTAL HEAT OF DRY AND SATURATED AIR.—The total heat of dry air is $h = C_a t$, where h = total heat, B.t.u. per lb.; C_a = specific heat of air = 0.24 (usual figure); t = temperature of air, deg. F. In saturated air the "heat of the liquid" usually is neglected, the error introduced thereby being negligible in air conditioning calculations. The total heat per 1 lb. of dry air in the saturated air-vapor mixture is the heat required to raise the temperature of 1 lb. of dry air from 0° F. to saturation temperature t, and to evaporate the weight W of vapor in the mixture. Thus

$$h_m = C_a t + rW \qquad [1]$$

where h_m = total heat of mixture, B.t.u. per lb.; r = latent heat of vapor at temperature t. For saturated air at 56° F., $r = 1060.3$, $W = 0.00954$ lb. Then (see Table 10, p. 1-08)

$$h_m = (0.24 \times 56) + (1060.3 \times 0.00954) = 23.555 \text{ B.t.u.}$$

TOTAL HEAT OF PARTIALLY SATURATED AIR.—A non-saturated air-vapor mixture may become saturated by an adiabatic process (see below) at the wet-bulb temperature. The total heat of the mixture is the same as the total heat of saturated air at the wet-bulb temperature, as given by the saturated air table, p. 1-08, or the psychrometric chart, Fig. 1. This relation is used in determining the total heat of partially saturated air.

WEIGHT OF VAPOR IN MIXTURE.—Assume a saturated mixture of 1 lb. of dry air, containing W lb. of vapor at temperature t, to be heated at constant pressure to a dry-bulb temperature of t_d, with a corresponding wet-bulb temperature of t_w. If the mixture is cooled at constant pressure back to t, t is the dew-point temperature for a combination of dry- and wet-bulb temperature t_d and t_w, and weight W lb. of vapor in the mixture is the same amount as contained in saturated air at the dew-point temperature. Thus if

dry- and wet-bulb temperatures are known, the dew-point temperature can be determined from the psychrometric chart, and consequently the value of W.

Method of Mixtures.—If x lb. of air be mixed with y lb. of air at dry-bulb temperatures t_1 and t_2 respectively, the resultant dry-bulb temperature is

$$t_3 = (xt_1 + yt_2)/(x + y) \quad \text{. . . . . [2]}$$

If W_1 and W_2 = weight of moisture per 1 lb. of air at dry-bulb temperatures t_1 and t_2 respectively, then the weight of moisture per 1 lb. of air in the mixture is

$$W_3 = (x\,W_1 + y\,W_2)/(x + y) \quad \text{. . . . . [3]}$$

From the psychrometric chart, Fig. 1, the dew-point corresponding to W_3 can be found. These relations often are used in air conditioning work.

ADIABATIC SATURATION OF AIR.—Air not completely saturated, when passed through a spray or over the surface of water housed in a completely insulated enclosure, will pick up vapor evaporated from the water. Dry-bulb temperature t_d of the leaving air will be lower than that of the entering air. If leaving air is saturated, its temperature will be the same as the wet-bulb temperature t_w of the entering air, which remains constant during the process. Since no external heat is supplied, and as heat is necessary to evaporate water, evidently the heat exchange has been between the air passing through the apparatus and the water. Such an exchange without transfer of external heat is termed an adiabatic process. Let W_1 = initial vapor content, lb. per lb. of dry air, corresponding to dry- and wet-bulb temperatures t_d and t_w respectively; W_2 = final vapor content corresponding to temperature t_w; r_w = latent heat corresponding to t_w, B.t.u. per lb. of vapor; C_a = specific heat of dry air = 0.24; C_s = specific heat of vapor = $0.4423 + 0.00018t_d$. Then heat to evaporate weight of vapor added to mixture = $r_w(W_2 - W_1)$ B.t.u. per 1 lb. of dry air, and $r_w(W_2 - W_1) = (C_a + C_s\,W_1)(t_d - t_w)$;

$$W_1 = \frac{r_w\,W_2 - C_a(t_d - t_w)}{r_w + C_s(t_d - t_w)} \quad \text{. . . . . [4]}$$

This psychrometric method of determining the weight of vapor mixed with or carried per 1 lb. of dry air was devised by W. H. Carrier in 1911.

Air Washers

The ability of a washer to *cool* air depends on fineness of spray, water pressure at the nozzles, arrangement of nozzles, number of banks of nozzles through which air passes, and direction of spray discharge from the banks.

HUMIDIFYING EFFICIENCY of an air washer, considered as a cooling device, is

$$E = \{1 - (t_{dl} - t_w)/(t_{de} - t_w)\} = (t_{de} - t_{dl})/(t_{de} - t_w) \quad \text{. . . . . [5]}$$

where E = efficiency, percent; t_{de} and t_{dl} = dry-bulb temperatures of entering and leaving air, respectively; t_w = constant wet-bulb temperature. Values of E attained in practice with various arrangement of nozzles are as follows: 2 banks upstream, 1 bank downstream, E = 1.00; 2 banks upstream, E = 0.95; 1 bank upstream, 1 bank downstream, E = 0.85; 1 bank upstream, E = 0.80; 1 bank downstream, E = 0.65.

Air washers are rated at from 400 to 500 cu. ft. of air per min. per sq. ft. of cross-sectional area through the spray chamber. The pump and spray nozzles should supply from 5 to 6 gal. of water per 1000 cu. ft. of air per min., the individual spray nozzles being rated at 1 1/2 gal. per min. The friction loss of air through a washer varies with the type. For 500 ft. per min. velocity it is approximately 0.25 in.

Psychrometric Chart and Heat Exchange Diagram (Fig. 1)

EXAMPLES IN THE USE OF CHART AND DIAGRAM.—Required the relative humidity for a dry-bulb reading of 84° F. and a wet-bulb reading of 66° F.

The intersection of a horizontal line through 66° F. on the *saturation curve*, and the vertical through 84° F., dry bulb on the base line gives, approximately, 37% for the relative humidity. The *dew-point* temperature is found by tracing the corresponding *constant weight vapor line* to its intersection with the *saturation curve*, giving 56° F. for the above condition. The actual weight of vapor per pound of dry air may be read directly from the saturation curve for the *dew-point* temperature of 56° F. and is 66.8 grains or 0.00955 lb.

Humidifying.—Assuming a room temperature of 70° F. and 40% relative humidity to be maintained when outside temperature is 0° F. Locate intersection of vertical 70° F. dry-bulb temperature with 40% relative humidity curve, follow the *constant weight vapor line*, passing through the intersection, to its intersection with the saturation curve or 45° F. corresponding to 0.0063 lb. or 44.2 grains of vapor per lb. of dry air. This is then the temperature at which saturated air must leave washer, and is the temperature for which thermostat controlling spray water heater must be set. See Fig. 2.

The heat per pound of dry air required for the tempering coil and water heater is read at the right of the diagram, and is 17.5 B.t.u. to raise temperature of incoming air from 0° F. to 45° F.

and saturate it at this temperature. The additional heat required for the reheater will depend upon the final temperature desired for air entering the room.

Air Cooling.—Entering air 89° F., dry bulb, and a relative humidity of 50% corresponding to a wet-bulb temperature of 74.5° F., wet-bulb depression (89 − 74.5) or 14.5 degrees. If the *humidifying efficiency* of a washer is 60%, then temperature drop will be 14.5 × 0.60 or 8.7° F. Temperature of leaving air = (89 − 8.7) or 80.3° F. Wet-bulb temperature remains constant at 74.5°. See Fig. 3.

Drying.—Outside air temperature 80° F. and 40% relative humidity; heater to raise its temperature to 110° F., at which temperature it is introduced into drier. Air to leave drier, 70% saturated. From intersection of vertical 80° F. dry-bulb temperature line and 40% relative humidity curve, follow the diagonal *equal weight of vapor line* until it intersects vertical 110° F. line corresponding to

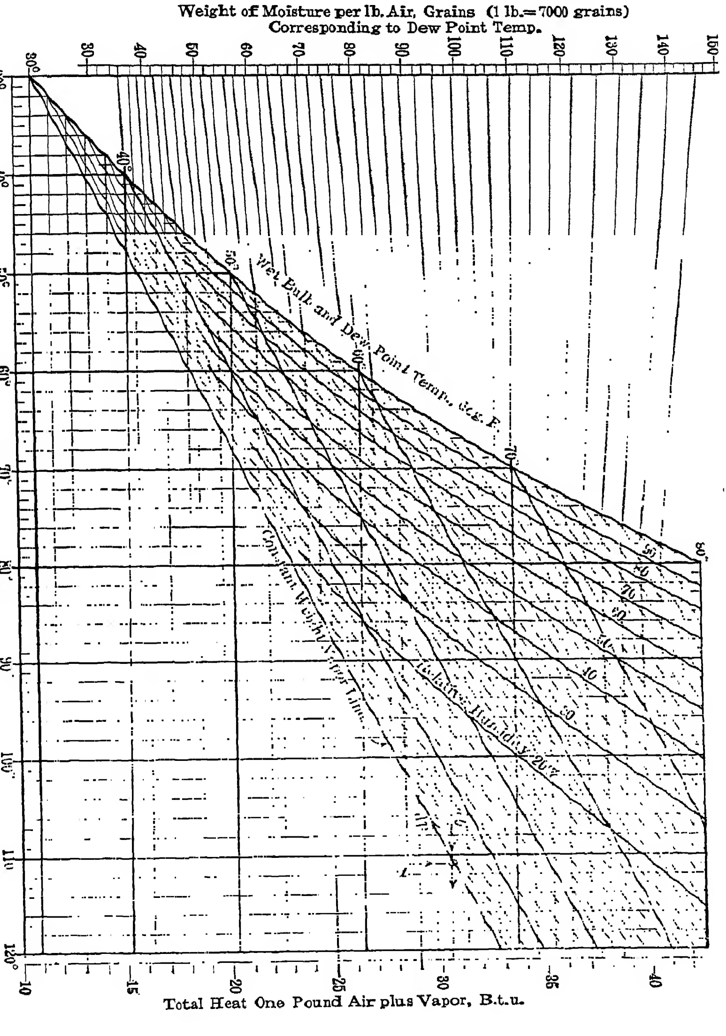


FIG. 1. Psychrometric Chart

73.5° wet-bulb. Read horizontally to the left to the intersection with the 70% curve. The corresponding dry-bulb temperature is 80.5° F., which is the required temperature of leaving air.

The weight of moisture evaporated per lb. of dry air circulated is the difference between the weight of vapor per lb. of dry air for 80.5° F. and 70% humidity and 80° F. and 40% humidity or 112.2 − 63.1 = 49.1 grains or 0.007 lb.

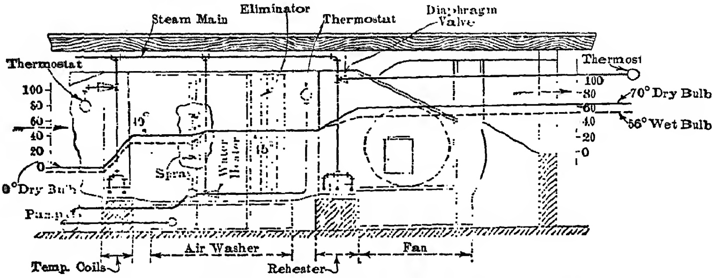


Fig. 2. Temperature Relations in Humidifying Apparatus

COMFORT COOLING.—Air is recirculated, and only enough outside air introduced to provide for ventilation requirements, approximately 10 to 15 cu. ft. per min. per occupant. Inside dry-bulb temperature to be maintained at breathing line should not exceed 10 to 15° F. below outside design dry-bulb temperature. The minimum dry-bulb temperature of air leaving cooler or washer depends largely on the distribution, which should be such as to prevent cold drafts over the occupants. Normally this temperature should be not less than dew-point temperature to be maintained in the room. A rule-of-thumb method for limiting minimum room inlet air temperature for horizontal outlets is: 2° F. per each foot of height of bottom of outlet above floor.

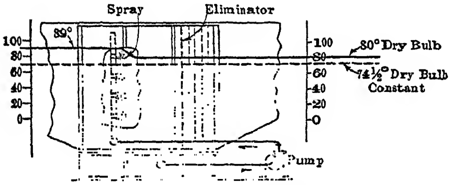


Fig. 3. Temperature Relations in Air Cooler

Weight of air to be circulated depends on sensible heat to be removed. The sensible heat load to be removed from the room is made up of: 1. Motors, electric lights and electrical heating appliances = total watts × 3.41 B.t.u. per hr. 2. Occupants at rest in 80° F. air, 225 B.t.u. per hr. per person. 3. Heat transmission of windows, walls, roofs and floors, based on the inside design temperature and outside design temperature dif-

Table 1.—Sun Effect, Solar Heat Flow Through Building Construction
(A.S.H.V.E. Research Laboratory)

Construction	Horizontal Surface	East or West Wall	South Wall
	B.t.u. per sq. ft. per hour		
Single glass *			
Maximum per hour.....	260.0	196.0	81.0
Total per day..........	1900.0	745.0	700.0
2-in. yellow pine			
Maximum per hour......	25.1	18.7	11.9
Total per day..........	203.0	80.0	75.0
4-in. gypsum			
Maximum per hour......	17.8	13.2	8.4
Total per day..........	179.0	70.0	66.0
6-in. concrete, no insulation			
Maximum per hour......	28.8	21.5	13.6
Total per day..........	249.0	98.0	92.0
3-in. concrete + 1-in. cork			
Maximum per hour......	16.7	12.4	7.9
Total per day..........	157.0	62.0	58.0

* If awnings are used, deduct 28% from tabular figures.

ference. Sun effect and heat transmission should not both be calculated for the same surface. 4. Sun effect on roofs and windows; see Table 1. 5. Heat loss from insulated steam pipes (approximately 90 B.t.u. per sq. ft. of pipe surface per hr.). 6. Infiltration of air = lb. of air per hr. × 0.24 × temperature difference between outside and inside air.

The so-called latent heat load is the heat to be removed by the washer or cooler to condense moisture that is to be removed from the air circulated through the cooler or washer. See formula [13].

Air Conditioning Calculations

Notation.—t_{od}, t_{op}, t_{ow} = respectively, dry-bulb, dew-point and wet-bulb design temperatures of outside air; t_{id}, t_{ip}, t_{iw} = respectively, dry-bulb, dew-point and wet-bulb design temperatures of inside air; t_{ed}, t_{ep}, t_{ew} = respectively, dry-bulb, dew-point and wet-bulb temperatures of air entering cooling apparatus; t_{ld}, t_{lp}, t_{lw} = respectively, dry-bulb, dew-point and wet-bulb temperatures of air leaving cooling apparatus; t_{rd}, t_{rp}, t_{rw} = respectively, dry-bulb, dew-point and wet-bulb temperature of air entering room; w_o, w_i, w_e, w_l, w_r = weight of vapor per 1 lb. of dry air, corresponding to dew-point temperatures with the same subscripts; h_1, h_2 = respectively, total heat, B.t.u. per 1 lb. of air-vapor mixture entering and leaving cooling apparatus, corresponding to wet-bulb temperatures t_{ew} and t_{lw}; H_s = computed estimate of sensible heat to be removed, B.t.u. per hr., due to occupants, lights, electrical machinery, heat transmission, sun effect, infiltration, etc.; H_v = sensible heat to be removed, B.t.u. per hr. due to air introduced for ventilation; H_l = latent heat load, B.t.u. per hr.; W_1 = weight of vapor added to room air by occupants, infiltration and other sources, lb. per hr.; each occupant at rest in 80° F. air gives off 0.17 lb. of vapor per hr.; W_2 = weight of vapor added to room return air by ventilation requirements, lb. per hr.; M = weight of air to be circulated through room and cooling apparatus, lb. per hr.; M_o = infiltration, lb. per hr.; M_v = ventilation requirements, lb. per hr. = $60n$ × cu. ft. of outside air per person × 0.075; n = number of occupants; c = cu. ft. per min. of standard air, 70° F.; 0.24 = specific heat of air at constant pressure. Then

$$W_1 = 0.17n + M_o(w_o - w_i) \qquad [6]$$

$$W_2 = M_v(w_o - w_i) \qquad [7]$$

$$M = H_s/0.24(t_{id} - t_{rd}) \qquad [8]$$

The weight of air M is based on sensible heat to be removed from room. It does not include sensible heat to be removed by cooler from outside air drawn in at the cooler for ventilation. This latter must be included in the sensible heat to be removed by the cooler.

The air leaving cooler and entering room must contain a weight of vapor of

$$w_r = w_i - (W_1/M) \qquad [9]$$

The corresponding dew-point temperature t_{rp} is found from the chart, Fig. 1.

DEHUMIDIFYING AIR WASHER (Fig. 4A).—The air leaves a dehumidifying air washer in a saturated condition at temperature t_{ld}, which is the dew-point temperature of air entering the room, or $t_{ld} = t_{rp}$. The saturated air must be warmed by a heater, or by mixing room air by-passed around the washer, to a final dry-bulb temperature of t_{rd}. Dry-bulb temperature of air entering cooling apparatus is, by method of mixtures,

$$t_{ed} = \{1 - (M_v/M)\}t_{id} + (M_v/M)t_{od} \qquad [10]$$

Weight of moisture per 1 lb. of air entering cooling apparatus is, by method of mixtures,

$$w_e = \{1 - (M_v/M)\}w_i + (M_v/M)w_o \qquad [11]$$

corresponding to a dew-point temperature of t_{ep}. Dry-bulb temperature of t_{ed} being known, wet-bulb temperature t_{ew} is obtained from the psychrometric chart, and h_1 thus is determined. The heat to be extracted from air circulated by the cooling apparatus is $M(h_1 - h_2)$ B.t.u. per hr., or

$$\text{Tons of refrigeration} = M(h_1 - h_2)/12{,}000 \qquad [12]$$

The cooler must condense the weight of vapor $(W_1 + W_2)$, and also must remove the sensible heat load $(H_s + H_v)$.

$$H_v = 0.24(t_{od} - t_{id})M_v \qquad [13]$$

Latent heat load may be approximated as

$$H_l = 1060(W_1 + W_2) \qquad [14]$$

in which 1060 is the assumed latent heat of vapor at the temperature at which it is condensed. This assumption is not strictly correct. For most practical purposes

$$\text{Tons of refrigeration} = (H_s + H_v) + H_l/12{,}000 \qquad [15]$$

This approximation may be used for a by-pass dehumidifying washer or a surface cooling

unit, but not for a dehumidifying washer without by-pass. In the latter, the heat to be removed by the washer is somewhat greater.

DEHUMIDIFYING WASHER WITH BY-PASS (Fig. 4B).—To avoid necessity of reheating to dry-bulb temperature t_{rd}, the saturated air leaving the washer at dew-point temperature t_{lp}, and also to reduce materially the refrigeration required, a return air by-pass around the washer is essential. Let x, y and z represent the fractional part of M that is outside air for ventilation, recirculated air through washer, and by-passed air around washer, respectively. Then $x + y + z = 1$.

By the method of mixtures we may write the equations

and $$z\,t_{ip} + (x + y)t_{lp} = t_{rp} \qquad [17]$$

Equation [17] is based on the fact that for comparatively small temperature ranges, dew-point temperatures may be substituted, without appreciable error, for the weight of vapor per lb. of dry air.

Subtracting [17] from [16], $z = (t_{rd} - t_{rp})/(t_{id} - t_{ip})$; substituting value of z in [17], $t_{lp} = (t_{rd} - z\,t_{id})/(x + y)$. By method of mixtures, $t_{ed} = (x\,t_{od} + y\,t_{id})/(x + y)$, and $w_e = (x\,w_o + y\,w_i)/(x + y)$, which is the weight of vapor per lb. of air entering washer. The corresponding dew-point, t_{ep} of air entering washer thus is ascertained. Corresponding wet-bulb temperature t_{ew} is given by the psychrometric chart, and h_1 is found in the saturated air table, p. 1–08. The value of h_2, from the saturated air table, corresponds to temperature t_{lp}. Tons of refrigeration required is found by the equation [12].

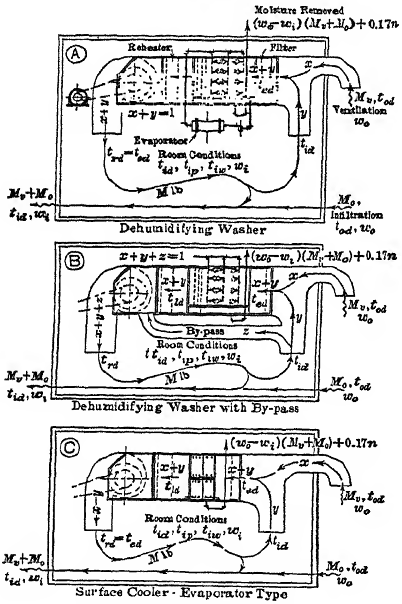


FIG. 4. Dehumidifying Washers and Surface Cooler

EVAPORATOR-TYPE COOLING SURFACE (Fig. 4C).—The heat transfer rate of any type of air cooling is a function of the velocity of air through free area of coils, and of the temperature of coil surface. The following data apply to a common type of evaporator cooling surface made of 5/8-in. copper tubing wound with a continuous copper fin, 8 fins per inch, coils being placed 1 7/16 in. centers. Velocity of air through gross face area of cooling coil enclosure should not be less than 400 ft. per min., to avoid carrying over free moisture. These coils are largely used for direct expansion cooling surface with freon and methyl chloride refrigerants.

One manufacturer furnishes assembled unit coils in sheet-metal casings. Unit casings may be obtained with overall heights of 20 9/16 in. with 12 rows of tubes across face, or 29 in. with 18 rows of tubes across face. Net face area, sq. ft., of unit is $A = \{(\text{casing height, in.} - 3)/12\} \times \{\text{tube length, ft.} - (8.5/12)\}$. Tube lengths obtainable vary from 1 ft. 6 in. to 6 ft. 6 in., in multiples of 6 in. Units are obtainable 1 to 3 rows of tubes deep in direction of air flow. Usually 3 to 5 tube rows depth of cooler is required, assembled from units available. An automatic refrigerant expansion valve is used for each row of tubes in the cooler. Capacity of valves required must be proportioned for the cooling load per row as shown in Table 2.

$$\text{Face area of cooler, sq. ft., } A = M/(0.075 \times 400) \qquad [18]$$

The relation between temperature of coil surface and temperature of refrigerant is $(t_x - t_s) = (H_s - H_v) + H_l$, or

$$t_s = t_x(h_1 - h_2)M/AH \qquad [19]$$

where t_s = temperature of refrigerant in evaporating coils, deg. F.; t_x = temperature of coil surface, deg. F.; H = heat absorbed, B.t.u. per hr. per sq. ft. of face area per deg.

Table 2.—Capacity of Refrigerant Expansion Valve Required in Evaporator Coolers

	No. of Rows Depth in Direction of Air Flow		
	3	4	5
	Proportion of Total Load		
1st Row..	0.417	0.346	0.305
2nd Row.	.319	.265	.234
3rd Row.	.264	.220	.194
4th Row.	....	.168	.148
5th Row.	....	....	.119

temperature difference between t_s and t_x; other notation as before. Values of H are given in Table 3.

Determination of Temperature t_S.—1. Locate on psychrometric chart, Fig. 1, the conditions of the air entering and leaving cooler, wet- and dry-bulb temperature being known. 2. Between the points so located draw a straight line to intersect the saturation line. See Fig. 5. Temperature corresponding to this intersection is required temperature of coil surface t_x. 3. Determine temperature of refrigerant by equation [19].

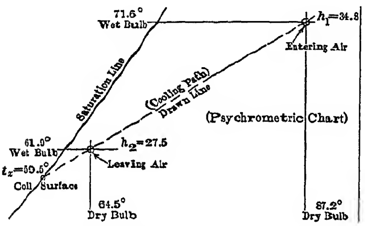


Fig. 5. Determination of Temperature of Refrigerant

Number of Tube Rows, Depth of Cooler depends on the ratio R of the sensible heat to the total heat to be extracted.

where h_x = total heat, B.t.u. per lb., of saturated air corresponding to temperature t_x of coil surface; other notation as before. Table 3 shows the maximum value of R for a given number of tube rows depth.

Reference to the psychrometric chart shows the relation

$$R = (h_1 - h_2)/(h_1 - h_x) = (t_{ed} - t_{ld})/(t_{ed} - t_x) \qquad [21]$$

[22]

Equation [22] enables surface temperatures for various tube loads to be determined.

Table 3.—Relation of Tube Row Depth and Heat Absorbed in Cooler

No. of Tube Rows in Depth...	2	3	4	5	6
R (max.).....................	0.544	0.682	0.770	0.839	0.900
Heat absorbed, H *..........	800	1200	1600	2000	2400

* B.t.u. per hr. per sq. ft. of face area per deg. F. temperature difference between refrigerant and coil surface, based on a heat transfer of 26.21 B.t.u. per hr. per sq. ft. of cooling surface per deg. temperature difference. 1 sq. ft. of face area = 15.9 sq. ft. of surface per tube row.

Example.—Required the face area of evaporative-type surface cooler, number of tube rows depth and temperature of refrigerant required for the following conditions: 10° F. cooling; occupancy, n = 100 persons; outside design conditions, t_{od} = 95° F., t_{ow} = 74.6° F. (38% relative humidity), t_{op} = 66° F., w_o = 0.01370 lb.; inside design conditions, t_{id} = 85° F., t_{iw} = 71° F., t_{ip} = 64° F., w_i = 0.01276 lb.

Solution.—Assume temperature of air entering the room $t_{rd} = t_{ld}$ = 64.5° F., and sensible heat to be removed H_s = 91,363 B.t.u. per hr. Weight of air to be circulated through room and cooler, equation [8],

$$M = 91{,}363/0.24\ (85 - 64.5) = 18{,}570 \text{ lb. per hr.} = 4127 \text{ cu. ft. per min.}$$

Air required for ventilation, M_v = 4050 lb. per hr.

Assume infiltration M_o = 11,600 lb. per hr.

Weight of vapor added to room air by occupants, etc., equation [6],

$$W_1 = (0.17 \times 100) + 11{,}600\ (0.01370 - 0.01276) = 27.9 \text{ lb. per hr.}$$

Weight of vapor to be added to room by ventilation air, equation [7],

$$W_2 = 4050\ (0.01370 - 0.01276) = 3.81 \text{ lb. per hr.}$$

$$W_1 + W_2 = 27.9 + 3.81 = 31.7 \text{ lb. per hr.}$$

Air must leave cooler and enter room with a vapor content of, equation [9],

$w_2 = w_3 = 0.01276 - (27.9/18{,}570) = 0.0112$ lb. per lb. of dry air, which corresponds to a dew-point of 60.5° F., a wet-bulb temperature of 61.9° F. and a heat content h_2 of 27.5 B.t.u. per lb.

Dry-bulb temperature of air entering cooler is, equation [10],

$$t_{ed} = \{(1 - (4050/18{,}570)\}85 + (4050/18{,}570)\ 95 = 87.2° \text{ F.}$$

Air entering cooler carries a weight of moisture, equation [11],

$$w_1 = \{1 - (4050/18{,}570)\}0.01276 + (4050/18{,}570)\ 0.01370 = 0.0129 \text{ lb. per lb. of air}$$

which corresponds to a dew-point of 64.5° F., a wet-bulb temperature of 71.6° F. and a heat content h_1 of 34.8 B.t.u. per lb.

Tons of refrigeration, equation [12], = 18,570 (34.8 − 27.5)/12,000 = 11.3.

By the approximate method, equation [15],

$$\text{Tons of refrigeration} = \{91{,}363 + 9720 + (1060 \times 31.7)\}/12{,}000 = 11.2$$

Face area of cooler, at 400 ft. per min. face velocity is, equation [18], A = 4127/400 = 10.3 sq. ft.

Locate on psychrometric chart, Fig. 1, initial and final conditions of air entering cooler. A line between the points so found intersects saturation line at t_x = 59.5° F. See Fig. 5.

Determine tube row depth by equation [21] and Table 3. R = (87.2 − 64.5)/(87.2 − 59.5) = 0.82, which from Table 3, calls for a 5 row depth of cooler with a value of H = 2000.

Temperature of refrigerant is, equation [19],

$$t_s = 59.5 - \{(34.8 - 27.5)\ 18{,}570/(10 \times 2000)\} = 52.9°\ \text{F.}$$

For weights of refrigerants to be handled by the compressor, compressor displacement, cu. ft. per min., and horsepower required, see section on Refrigeration.

UNIT COOLERS consisting of an assembly of fan, various numbers of rows depth of coils and a dry type air filter are obtainable from several manufacturers. These assembled units are used largely for comfort cooling of restaurants, stores, offices, etc. Unit coolers, consisting of a direct expansion cooler and fan, similar to unit heaters, are used

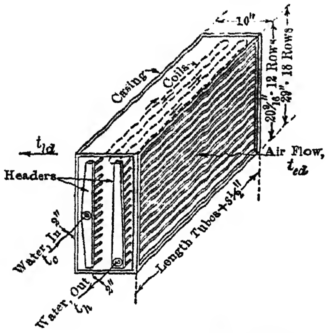


Fig. 6. Unit of Continuous Tube Water Cooler

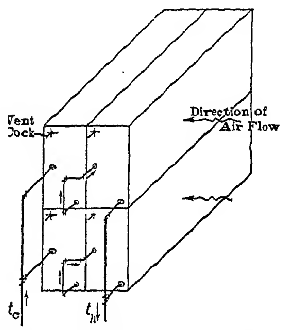


Fig. 7. Assembled Cooler of Four Units

in cold storage rooms in which the air simply is recirculated. Units in which ammonia is the refrigerant are obtainable from a number of manufacturers.

SURFACE TYPE COOLERS WITH CHILLED WATER SUPPLY.—A type of air cooler used in air conditioning is made of copper tubes 5/8 in. diam., with 8 copper ribbon tinned fins per inch of extended surface. The continuous coils are connected, at the beginning and end of run, into a bronze header. See Figs. 6 and 7. Header inlet and discharge connections are 2 in. diam. The coils are assembled in a sheet-metal casing, obtainable with face heights of 12 tubes (20 9/16 in. height of casings), and 18 tubes (29 in. height of casing). Assembled units may be obtained varying in length of finned tube from 2 ft. to 10 ft. in multiples of 6 in. Overall length of casing is (tube length + 8 1/2 in.). Depth of casing in all cases is 10 in.

The net tube area, sq. ft., is

$$(\text{length of tube across face, ft.}) \times \{(\text{casing height, in.} - 3)/12\}.$$

Rated heat transfer surface of these coils is 15.4 sq. ft. per row of tube depth per sq. ft. of face area. The number of tube rows depth also is the number of passes the water makes across the unit. The quantity of water flowing through each unit of the cooler is

$$\text{Gal. per min. (g.p.m.)} = vN/1.235 \quad . \ . \ . \ . \ . \ . \ . \ . \quad [23]$$

where N = number of tube rows high; v = velocity, ft. per sec.

Face area of cooler A = cu. ft. per min. ÷ allowable face velocity V, ft. per min.

Number of tube rows depth N required is

$$N = S/(A \times K \times F \times T_m) \quad . \ . \ . \ . \ . \ . \ . \ . \ . \quad [24]$$

where $S = HR$ = sensible heat to be transferred; H = total heat to be transferred, B.t.u. per hr.; R = ratio of sensible heat to total heat to be abstracted; A = face area

of cooler, sq. ft.; K = sensible heat transfer from air to water per sq. ft. of face area per deg. mean effective temperature difference between air and water per row depth per hr. (see Table 4); T_m = mean effective temperature difference between air and water $= (d_1 - d_2)/\log_e(d_1/d_2)$, where d_1 and d_2 are the greatest and least differences of temperature between entering and leaving air and water. Values of H and R are as follows:

$$H = 0.75 \times \text{cu. ft. per min.} \times (h_1 - h_2);$$

where h_1 and h_2 = respectively initial and final heat content of air passing through cooler, t_{ed}, t_{ld} = respectively, entering and leaving dry-bulb air temperatures, deg. F.

Initial and final water temperatures obviously must be lower than dew-point temperature of leaving and entering air if condensation is to occur. Entering water temperature may be assumed 8 to 10° F. lower than dew-point of entering air.

Table 4.—Values of F and K in Equation 24

Face Velocity of Air ft. per min. V	Water Velocity, v, ft. per sec.*					R	Water Velocity, v, ft. per sec.*				
	1	2	3	4	8		1	2	3	4	8
	Values of K						Values of F				
300	97	112	117	121	129	0.90	0.954	0.965	0.970	0.974	0.980
400	113	133	141	148	158	.80	.901	.924	.935	.943	.957
500	126	150	162	170	183	.70	.842	.876	.893	.907	.928
600	136	166	179	190	206	.60	.774	.820	.844	.860	.894
						.50	.696	.752	.782	.806	.847
						.40	.605	.670	.706	.735	.787

* v = (g.p.m. per unit)/1.235 × No. tube rows high. Units may be either 12 or 18 tube rows high.

Table 5.—Friction of Dry Air Through Cooler

Face Velocity of Air, ft. per min. V	No. Rows Depth of Cooler									
	2	4	5	6	8	10	12	16	18	20
	Friction loss, in. of water *									
300	0.08	0.10	0.12	0.15	0.19	0.24	0.28	0.37	0.42	0.47
400	.10	.18	.22	.25	.34	.42	.50	0.65	0.74	0.82
500	.15	.27	.33	.39	.52	.64	.76	1.10	1.14	1.26
600	.21	.38	.47	.56	.73	.91	1.08	1.44	1.61	1.80

* For wet coil surfaces, with initial water temperature above 50° F., multiply tabular values by 1.35; for water temperatures below 50° F., by 1.50.

Table 6.—Water Friction Loss per Row of Tubes Depth

Length of Tube Unit Across Face, ft.	Water Velocity through Tubes, ft. per sec.				
	1	2	3	4	8
	Friction, ft. head of water				
2	0.10	0.50	1.20	2.20	8.40
4	.15	0.75	1.80	2.80	10.60
6	.20	1.00	2.00	3.45	13.00
8	.22	1.10	2.40	4.02	15.30
10	.25	1.13	2.80	4.70	17.60

Example.—Air entering cooler: Dry-bulb, 95° F.; wet-bulb, 75° F.; dew-point, 66.5° F.; h_1 37.81 B.t.u. per lb.

Air leaving cooler: Dry-bulb, 63.7° F.; wet-bulb, 60.0° F.; dew-point, 58.0° F.; h 26.18 B.t.u. per lb.

H = 1,046,700 B.t.u. per hr. = 17,445 B.t.u. per min. Equivalent volume of standard air to be circulated through cooler = 20,000 cu. ft. per min. = 90,000 lb. per hr.

Face velocity V = 580 ft. per min. Water velocity v = 5 ft. per sec. Initial temperature t_e of water = 50° F., or 8° F. below dew-point of entering air.

Required the number of rows of tubes in cooler.

Solution.—Assume each cooler unit to be 18 tubes high, in parallel. Water required per unit, equation [23], is g.p.m. = 5 × 18/1.235 = 73 gal. per min.

Face area of complete cooler = A = 20,000/580 = 34.5 sq. ft., indicating a cooler 2 units high with 8 ft. length of tubes. Total g.p.m. for both units = 2 × 73 = 146, or 146 × 8.33 = 1213.2 lb. per min.

Temperature rise of water = 17,445/1213.2 = 14.4° F.

Final temperature of water = 50 + 14.4 = 64.4° F., which is lower than the dew-point of entering air.

Maximum temperature difference between air and water at air inlet of cooler is (95 − 64.4) = 30.6° F.

Maximum temperature difference between air and water at air outlet of cooler is (63.7 − 50) = 13.7° F.

Mean effective temperature difference is $(30.6 - 13.7)/\log_e (30.6/13.7) = 20.06$.

$$\text{Arithmetical mean} = \{(95 + 63.7)/2\} - \{(64.4 + 50)/2\} = 22.15$$

Ratio of sensible heat to total heat removed by cooler is

$$R = 0.24\ (95.0 - 63.7)/(37.81 - 26.18) = 0.646$$

Sensible heat S = 1,046,700 × 0.646 = 676,168 B.t.u. per hr.

By interpolation in Table 4, for $V = 580$ and $v = 5$, $K = 190$; for $R = 0.646$ and $v = 5$, $F = 0.889$.

Then, equation [24], $N = 676{,}168/(34.5 \times 190 \times 0.889 \times 20.06) = 5.78$, indicating the use of a 6-row cooler.

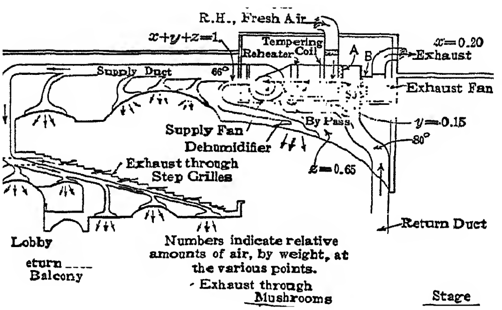


FIG. 8. Air Conditioning of Theater

AIR CONDITIONING A THEATER.—Applying the formulas and data above given for the sensible and latent heat loads and usual outside design dry-bulb temperatures (90° to 95° F.) and 60% relative humidity, the relative weight of fresh outside air, recirculated air, and by-passed air are about as indicated in Fig. 8. That is, $x = 0.20$, $z = 0.65$, $y = 0.15$. Usually the refrigeration requirement is 13.5 seats per ton, or 74.3 tons per 1000 seats.

Section 12

INTERNAL COMBUSTION ENGINES

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DIESEL ENGINES

By Edgar J. Kates

A Diesel engine is a prime mover actuated by the gases resulting from the combustion of a liquid or pulverized fuel, injected in a fine state of subdivision into the engine cylinder at or about the conclusion of a compression stroke. The heat generated by the compression to a high temperature of air within the cylinder is the sole means of igniting the charge. After being ignited, the charge burns and expands, thus converting the heat energy of the fuel into work.

The thermal efficiency of Diesel engines has advanced from 13.5% efficiency of the Hornsby-Akroyd hot-surface oil engine in 1890 to the 41% efficiency (brake Hp. basis) of a 5500-Hp. two-cycle marine engine tested in 1934.

High-compression, self-ignition engines, now being built in the U.S. to which the generic name Diesel is applied, are the outcome of two parallel developments.

The first practical petroleum engine was produced by Julius Hock in Vienna in 1870, and the first compression oil engine of Brayton was built in the United States about 1873. The Hornsby-Akroyd oil engine was the first to meet with undoubted success in 1886-1890. The chief feature of this design was its extreme simplicity of construction. For a complete history of Petroleum Engines, see Gas and Petroleum Engines by W. Robinson; American Diesel Engines by L. H. Morrison.

The Diesel engine, patented in 1892, was designed originally by Dr. Rudolf Diesel to operate on the Carnot or isothermal cycle, and to use pulverized coal as fuel. This engine was unworkable. It was modified to use oil as fuel, and to maintain a nearly constant pressure in the cylinder during the injection of fuel. See Fig. 1. This was the progenitor of the air-injection Diesel engine, which uses a blast of compressed air to inject highly atomized fuel into the cylinder.

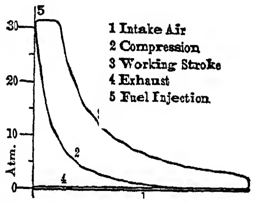


Fig. 1. Diesel Engine Cycle

The solid-injection Diesel engine atomizes the fuel by means of its own pressure, without the use of blast air. It is a development of the low compression, hot-bulb engine. It operates either on the cycle shown in Fig. 2, where the combustion takes place almost instantaneously, or on the so-called dual cycle, Fig. 3. In this cycle the first part of the combustion is at constant volume and the latter part at nearly constant pressure.

1. TYPES OF DIESEL ENGINES

CLASSIFICATION.—Diesel engines may be classified as: 1. *a*, 4-cycle; *b*, 2-cycle. 2. *a*, Single-acting; *b*, double-acting. 3. *a*, Vertical; *b*, horizontal; *c*, radial. 4. By speed. In 1935, rotative speeds ranged from 95 to 2500 r.p.m., and average piston speeds from 690 to 2330 ft. per min. 5. *a*, Air injection; *b*, solid injection; the latter may be subdivided into, b_1, direct injection; b_2, antechamber; b_3, auxiliary air chamber. 6. By size (horsepower). In 1935, Diesel engines were built in sizes of 5 Hp. to 22,500 Hp.

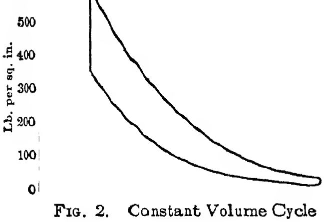


Fig. 2. Constant Volume Cycle

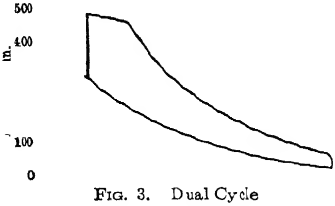


Fig. 3. Dual Cycle

CYCLES OF OPERATION.—All commercially used Diesel engines operate on one of the following cycles: *a*. The 4-cycle (4-stroke cycle, Otto cycle, or Beau de Rochas cycle) completed in four strokes of the piston, *viz.*, 1. Air inlet. 2. Compression. 3. Expansion. 4. Exhaust. See Fig. 4B. *b*. 2-cycle (2-stroke-cycle) where the processes are completed in two strokes of the piston, *viz.*, 1. Expansion (compression stroke having just been completed); at the end of the expansion stroke, exhaust ports are uncovered by the piston, the burnt gases exhausted and pure air at a slight pressure enters the cylinder; 2. Compression, or inward stroke of the piston, which closes the exhaust ports; fuel injection occurs as compression is completed. See Fig. 4A.

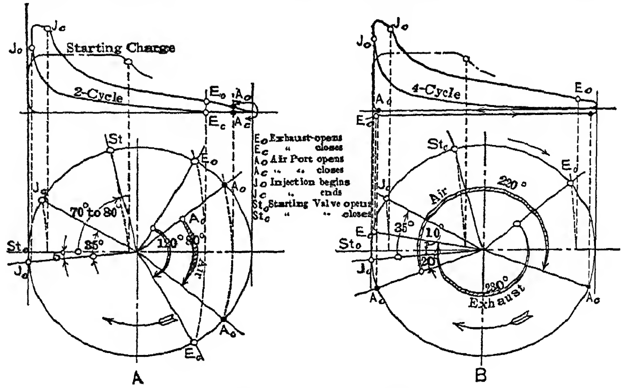


FIG. 4. Operation Cycles

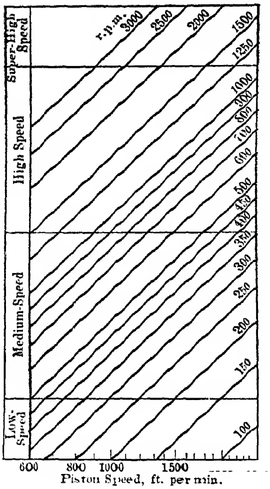


FIG. 5. Speed Classification Chart

Goldingham gives relative advantages and disadvantages of 2-cycle and 4-cycle engines as:

Advantages.—4-Cycle: 1. More complete combustion of fuel. 2. Lower fuel consumption. 3. Greater accessibility where the crank-case is open and moving parts in view. 4. Use of lower grades of fuel without a tendency to carbonize on piston, rings and exhaust passages.

2-Cycle: 1. No exhaust valves, air inlet valves or valve motion. 2. Power developed by piston displacement per unit of volume, 75% to 90% greater than with a 4-cycle type. 3. More even crank effort. 4. Lighter fly-wheel than with 4-cycle type.

Disadvantages.—4-Cycle: *a*. Variable crank-pin effort, requiring heavy fly-wheel. *b*. Greater total weight per horsepower. *c*. Exhaust and air-inlet valves and valve motion necessary.

2-Cycle: *a*. Inferior combustion of fuel and smoky exhaust. *b*. More lubrication and cooling water required. *c*. Increased fuel consumption. *d*. Possible leakage of air with crank-case compression.

SPEED.—Design and construction of Diesel engines are influenced by speed. Increase in rotative speed causes: 1. Increased number of reversals of reciprocating parts. 2. Shortened periods for the various cycle events, as intake of air, exhaust, and particularly injection of fuel. 3. Increase in acceleration of reciprocating parts, accompanied by increased inertia forces on rotating parts. 4. Increased noise due to increased inertia forces and velocities. 5. Larger stresses from torsional vibrations and critical speeds. 6. Increased peripheral speeds of bearings, gears, chain drives. 7. Increased linear speeds of reciprocating wearing surfaces. 8. Increased piston displacement per unit of time, with increased gas velocities through intake and exhaust-valve openings.

Engines commonly are classified as to speed by rotative speed and average piston speed. V. L. Maleev and E. C. Magdeburger (*Trans.* A.S.M.E., OGP–54–5, 1932) show that

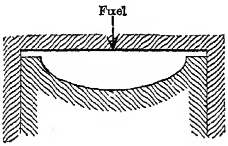


FIG. 6. Combustion Space of Air Injection Engine

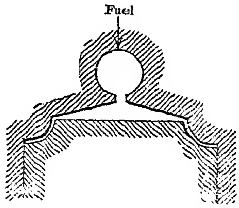


FIG. 7. Antechamber or Precombustion System

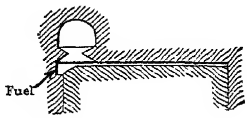


FIG. 8. Auxiliary Air Chamber System

neither of these by itself is suitable. They proposed the characteristic (R.p.m. × piston speed, in ft. per min.) ÷ 100,000. The division lines are:

Speed Class.	Low	Medium	High	Super-high
Speed Characteristic.	Below 3	3–9	9–27	over 27

Any engine may be quickly classified as to speed by means of Fig. 5.

COMBUSTION SYSTEMS.—The principal combustion systems are: 1. Air-injection (blast-injection); fuel is blown into the cylinder and atomized by air at 800 to 1200 lb. per sq. in. Fig. 6 shows shape of combustion space in an air-injection engine. 2. Solid-injection (mechanical injection, airless injection, pump-injection); fuel is precompressed to 800 to 6000 lb. per sq. in., and is atomized by passing through one or more small orifices.

Solid-injection system may be classed as: 1. *Direct injection*, or *single-chamber*. Fuel is injected directly into an undivided combustion space of shape similar to that of Fig. 6. 2. *Antechamber*, or *Precombustion Chamber*, Fig. 7. The combustion space comprises two parts, the antechamber and the main space, joined by a constricted passage. Fuel is injected into the antechamber and is partly burned. The resulting rise in pressure in the antechamber ejects the remaining fuel and hot gases into the main chamber, where the fuel mixes with the air and burns. 3. *Auxiliary Air Chamber*, Fig. 8, often is confused with the antechamber type, because the combustion space similarly is in two parts joined by a constricted passage. The important difference is that the fuel does not burn in the auxiliary air chamber. During the compression stroke, air is forced into the auxiliary

Table 1.—Relative Advantages and Disadvantages of Various Combustion Chamber Types

Direct injection or single chamber See Fig. 6	Antechamber or precombustion chamber See Fig. 7	Auxiliary air chamber See Fig. 8
	Advantages	
Best thermal efficiency.	Low fuel injection pressure.	Low fuel injection pressure.
High mean effective pressures.	Inexpensive fuel injection system.	Inexpensive fuel injection system.
Start from cold without electric heating coils.	Uses fairly viscous fuels.	Uses fairly viscous fuels.
Simple cylinder head design.		Better thermal efficiency and mean effective pressures than antechamber.
Maximum space for valves.		
	Disadvantages	
High fuel injection pressure.	Greater surface-volume ratio than in direct injection. Greater heat loss through jackets.	Lower thermal efficiency and mean effective pressure than direct injection.
High cylinder pressure.		
Sensitive to more viscous grades of fuel.	Lower thermal efficiency; lower mean effective pressures.	Requires heating coils or higher compression for quick starting.
	More difficult to start; requires heating coils or higher compression.	

air chamber, which acts as an air reservoir. Just before top dead center, fuel is injected into the passage between the air reservoir and main combustion space, and combustion starts in the passage. As the piston moves down, air flows out of the air reservoir, both by pressure difference and by inertia effect, meeting the injected fuel in the passage. Pure air for combustion thus is continually supplied to the burning fuel, and the products of combustion pass into the cylinder.

Table 1 summarizes the advantages and disadvantages of the various types of combustion chambers used in solid-injection engines.

FIELDS OF APPLICATION.—The principal fields of application and the types of Diesel engines generally used are: 1. Oil pipe line pumping. Formerly horizontal, now (1935) generally vertical. Two- or four-cycle, slow speed, air or solid injection. 2. Marine propulsion. Vertical, 2- or 4-cycle, air or solid injection. Slow speed for direct drive, medium speed for gear or electric drive. 3. Electric central stations. Vertical, 2- or 4-cycle, slow or medium speed, air or solid injection. 4. Ice-making and refrigeration. Vertical or horizontal, 2- or 4-cycle, slow speed, air or solid injection. 5. Industrial plants, cotton-gins, etc. Vertical or horizontal, 2- or 4-cycle, slow or medium speed, air or solid injection. 6. Construction equipment. Vertical, generally 4-cycle, medium speed, solid injection. 7. Locomotives and rail cars. Vertical, 4-cycle, medium or high speed, solid injection. 8. Auto trucks and buses. Vertical, 4-cycle, high speed, solid injection. 9. Airplane propulsion. Radial, 4-cycle, high or super-high speed, solid injection.

2. THERMODYNAMICS OF DIESEL ENGINES

For general subject of thermodynamics, see Section 4. The Diesel engine is a heat engine whose purpose is to convert the combustion energy of its fuel into mechanical energy. The air which unites with the fuel always is precompressed to raise its temperature so that it will ignite the fuel injected near the end of the compression stroke, and to increase efficiency by subsequent expansion of the burned charge. In any internal combustion engine, compression before ignition extends the range of effective expansion.

Fig. 9 from *Diesel Power*, Dec., 1933, shows the heat balance throughout the range of load. Engines of different makes will show considerable variation.

Power and Efficiency Formulas

DISPLACEMENT of an engine is the volume, cu. ft. per min., swept by the piston or pistons during the power strokes. It is equal to (Number of cylinders × area of each piston, sq. ft. × stroke, ft., × number of power strokes per minute).

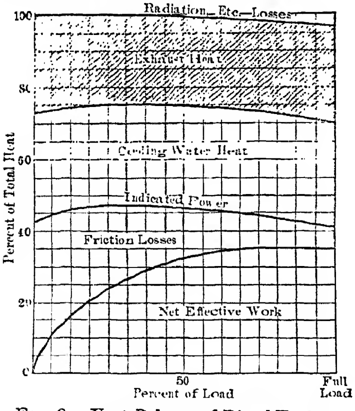


FIG. 9. Heat Balance of Diesel Engine

VOLUMETRIC EFFICIENCY is the ratio V_a/D, where V_a = volume of air, cu. ft. per min., at intake temperature and pressure, induced and compressed, and D = displacement.

INDICATED HORSEPOWER of an engine cylinder is the horsepower developed in the cylinder. The formula is

$$\text{I.Hp.} = (\text{M.i.p.} \times L \times A \times N) \div 33{,}000 \quad [1]$$

where M.i.p. = mean indicated pressure, lb. per sq. in.; L = stroke of piston, ft.; A = net piston area, sq. in.; N = number of power strokes per min.

MEAN INDICATED PRESSURE (M.i.p.) of an engine cylinder is average net pressure, lb. per sq. in., acting on piston throughout one cycle.

BRAKE HORSEPOWER (B.Hp.) is the horsepower delivered by the shaft at the output end. The formula is

$$\text{B.Hp.} = 2\pi l n W/33{,}000 \quad . \quad . \quad . \quad [2]$$

where l = distance between shaft center and bearing point of brake arm, ft.; n = revolutions per minute of brake shaft; W = net weight on brake arm, lb.

BRAKE MEAN EFFECTIVE PRESSURE (B.m.e.p.) is

$$\text{B.m.e.p.} = (\text{B.Hp.} \times 33{,}000)/(L \times \quad [3]$$

PISTON SPEED is the total feet of travel made by each piston in one minute. The formula is

$$\text{Piston Speed} = 2 \times \text{stroke in feet} \times \text{r.p.m.} \qquad [4]$$

INDICATED THERMAL EFFICIENCY is the ratio of the heat equivalent of 1 Hp.-hr. (2545 B.t.u.) to the heat units actually supplied per I.Hp.-hr., based on the higher heating value of the fuel.

BRAKE THERMAL EFFICIENCY is the ratio of the heat equivalent of 1 Hp.-hr. to the heat units actually supplied per B.Hp.-hr., based on the higher heating value of the fuel.

MECHANICAL EFFICIENCY is the ratio of brake horsepower to indicated horsepower.

HEAT LOSSES IN INTERNAL COMBUSTION ENGINES are: *a.* Loss through externally cooled walls, when gases are at maximum temperature and during compression; *b.* Throttling of air inlet and back pressure during exhaust; *c.* Incomplete combustion at maximum temperature and loss through exhaust. The distribution of heat energy in a typical Diesel engine at full load is:

	B.t.u. per B.Hp.-hr.	Percent of heat in fuel
Brake work	2545	33
Friction	680	9
Heating of jacket water	2100	27
Heat in exhaust gases	2200	28
Radiation, etc.	255	3
Total	7780	100

THERMODYNAMIC ANALYSIS OF INTERNAL COMBUSTION ENGINE CYCLES (Goodenough and Baker, *Bull.* No. 160, Univ. of Ill., 1927).—In a theoretical analysis of the cycle of an internal combustion engine three degrees of approximation may be observed. 1. The simplest and crudest system of analysis gives the so-called air standard, which is still used to estimate the engine efficiencies. This analysis assumes that the medium throughout the cycle is air or a gas with the same properties. During the combustion phase the air is supposed to receive an amount of heat equal to the heat of combustion of the fuel. The specific heat of air usually is taken as constant. The air standard efficiency deduced from this analysis always is from 10 to 25% higher than the efficiency obtained from more accurate analyses.

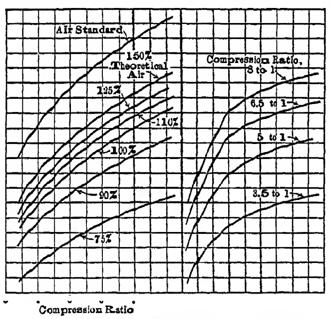


Fig. 10. Variation of Efficiency with Compression Ratio and with Mixture Strength for Otto Cycle

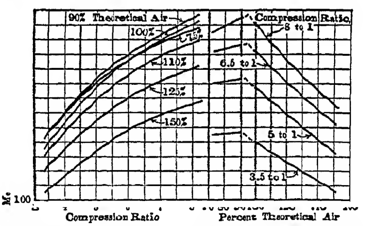


Fig. 11. Variation of M.E.P. with Compression Ratio and Mixture Strength for Otto Cycle

2. The properties of the actual gas mixtures are used. The medium compressed is a mixture of fuel and air; the medium expanding adiabatically after combustion is an entirely different mixture of different properties. In this analysis, it is assumed that combustion is complete before adiabatic expansion begins.

3. It is well known that at the maximum pressure and temperature attained in the cycle, combustion is incomplete, and that, due to dissociation of CO_2 and H_2O at temperatures above 2500° F., the mixture will contain unburned CO and H_2 at the beginning of adiabatic expansion. As the temperature falls during expansion, combustion continues, and at the end of expansion it is practically complete. The third system of analysis takes account of these phenomena.

For the Otto cycle the effect of compression ratio and mixture strength on thermal efficiency is shown in Fig. 10. The effect of these factors on mean effective pressure is

shown in Fig. 11. Corresponding relations for the Diesel cycle are shown in Figs. 12 and 13.

As a result of studies based on the third system of analysis, Goodenough and Baker concluded: 1. Efficiency increases with compression ratio, *i.e.*, the higher the compression the higher the efficiency, other conditions being unchanged. 2. For the same compression, efficiency increases with amount of air used. A lean mixture gives higher efficiency than a rich mixture. 3. Mean effective pressure is a maximum when air supply is somewhat less than 100% of the theoretical amount (Fig. 11). The mixture for maximum power is a mixture of relatively low efficiency. 4. Ideal efficiencies obtained from various liquid fuels are practically the same. 5. Efficiencies of the Diesel cycle, as a group, range higher than efficiencies of the Otto cycle. However, a comparison of the two efficiencies at the same compression ratio (r = 8) shows the Otto cycle inherently to be more efficient than the Diesel cycle. The superior efficiency of the Diesel cycle is due to the high compression ratio permitted by the system of operation.

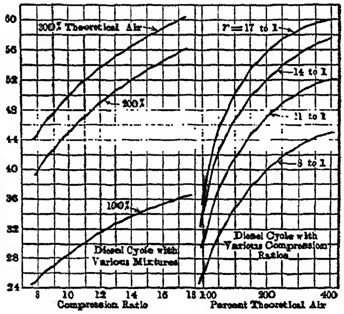


FIG. 12.—Effect of Compression Ratio and Mixture Strength on Efficiency of Diesel Cycle

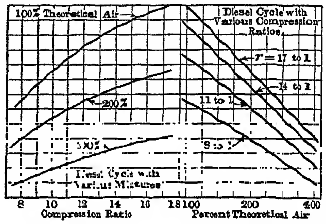


FIG. 13. Effect of Compression Ratio and Mixture Strength on M.E.P. of Diesel Cycle

Ellenwood, Evans and Chwang (*Trans*. A.S.M.E., OGP–50–5, 1928) present Fig. 14, showing in convenient form the efficiency of an ideal Diesel engine using petroleum. A useful temperature-entropy diagram for air and the diatomic gases O_2, N_2 and CO is given by H. A. Everett in *Mech. Engg.*, Mid-Nov. 1926, p. 1329.

COMPRESSION PRESSURES AND TEMPERATURES.—The temperature of the air charge at the end of compression must be high enough to ignite the fuel. The relations of pressures and temperatures at beginning and end of the compression stroke are expressed by $(T_2/T_1) = (P_2/P_1)^{n-1/n}$, where T_1, T_2 = respectively, temperatures

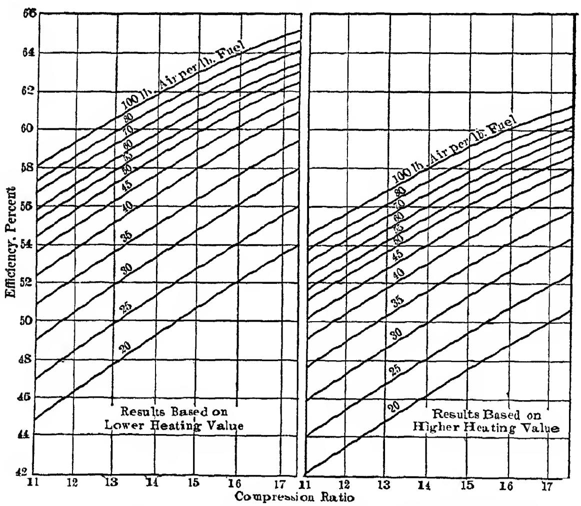


FIG. 14. Efficiency of Ideal Diesel Engine Using Petroleum Oil

(absolute) at beginning and end of compression; P_1, P_2 = respectively, initial and final pressures (absolute). For adiabatic compression, $n = 1.408$. In an actual engine, however, loss of heat occurs, and the value of n is about 1.35. These relations are shown graphically in Fig. 15.

3. RESEARCH ON INJECTION PHENOMENA

The development of light-weight, high-speed engines for automotive use has led to considerable useful research on injection and combustion phenomena. Outstanding work has been done by the National Advisory Committee for Aeronautics at Langley Field, Va., Pennsylvania State College, State College, Pa., H. R. Ricardo, A. L. Bird, and the Royal Aircraft Establishment in England, and F. Sass, K. Neumann and R. Kuehn in Germany.

A few of the significant findings are abstracted below. See original reports mentioned for full details.

FACTORS IN THE DESIGN OF CENTRIFUGAL TYPE INJECTION VALVES FOR OIL ENGINES (Nat. Advis. Comm. for Aeronautics, Report No. 268, 1927).—The general method was to record the development of single sprays by high-speed photography, taking 25 consecutive pictures of the moving spray at a rate of 4000 per second. Investigations were made of effects on spray characteristics of the helix angle of helical grooves, ratio of cross-sectional area of orifice to that of the grooves, ratio of orifice length to diameter, and position of the seat. Sprays were injected at 6000, 8000 and 10,000 lb. per sq. in. pressure into air at atmospheric pressure, and into nitrogen at 200, 400, and 600 lb. per sq. in. pressure. Orifice diameters were from 0.012 to 0.040 in.

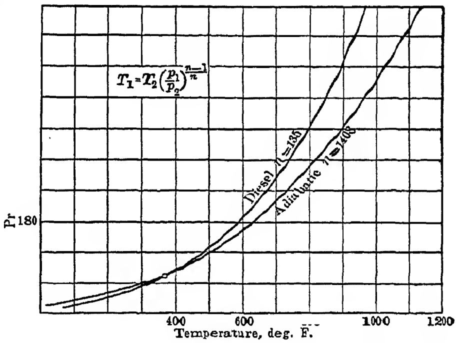


FIG. 15. Relation of Pressure and Temperature of Air

Decreasing the pitch of the helical grooves, thus increasing the centrifugal force applied to the spray, increased the spray cone angle considerably; percentage increase was much less in dense air than in atmosphere. Spray penetration decreased as centrifugal force applied increased. About twice the spray volume per unit oil volume was obtained with a high centrifugal spray as with a non-centrifugal spray. Spray cone angle increased, and spray volume-oil volume ratio, and spray penetration decreased with increase in ratio of orifice area to groove area. Maximum spray penetration occurred at a ratio of orifice length to diameter of about 1.5. Slightly greater penetration was obtained with the seat directly before the orifice.

EFFECT OF ORIFICE LENGTH-DIAMETER RATIO ON FUEL SPRAYS FOR COMPRESSION-IGNITION ENGINES (N.A.C.A. Report No. 402, 1931).—With a plain stem in the injection valve and an increase in length-diameter ratio of the orifice from 0.5 to 10, rate of spray-tip penetration decreased to a minimum between ratios of 1.5 and 2.5, reached a maximum between ratios of 4 and 6, and again decreased as ratio was increased to 10. Exact position of maximum and minimum points depended upon orifice diameter. Spray cone angle was affected but little by variation either of diameter of orifice or length-diameter ratios greater than 4.

With a helically grooved stem in injection valve, ratios at which highest penetration occurred ranged between 5 and 7. Spray cone angle increased with increased ratio of orifice area to groove area.

N.A.C.A. APPARATUS FOR STUDYING FORMATION AND COMBUSTION OF FUEL SPRAYS. (N.A.C.A. Report No. 429, 1932).—The report includes result of tests to determine effect of air temperature, air flow, and nozzle design on spray formation. Compression temperature has little effect on penetration of fuel spray, but does affect dispersion. Air velocities of about 300 ft. per sec. are necessary to destroy the spray core. Effect of air flow on spray is controlled to a certain extent by design of injection nozzle. Results of tests on combustion of spray show that when ignition occurs after spray cut-off, ignition may start almost simultaneously throughout the combustion chamber or at different points in the chamber. When ignition occurs before spray cut-off, combustion starts around the edge of the spray, and spreads throughout the chamber.

FUEL VAPORIZATION AND ITS EFFECT ON COMBUSTION IN A HIGH-SPEED COMPRESSION-IGNITION ENGINE (N.A.C.A. Report No. 435, 1932).—Tests were made to determine whether or not fuel injected into a high-speed compression-ignition engine vaporizes appreciably during the time available for injection and combustion. Effects of injection advance angle, fuel boiling temperatures, fuel quantity, engine speed, and engine temperature were investigated. The results show that an appreciable amount of fuel is vaporized during injection even

though temperature and pressure in engine are too low to cause ignition either during or after injection. When conditions are such as to cause ignition, the vaporization process affects combustion.

INJECTION LAG IN A COMMON-RAIL FUEL INJECTION SYSTEM (N.A.C.A. Technical Note No. 332, 1930).—Injection lag increased lineally with increase in injection tube length, but was not affected by the bore of the tube. Lag increased slightly with increase of valve opening pressure, and decreased materially with increase of injection pressure. Initial pressure in injection valve tube before timing valve opened did not affect injection lag, if injection pressure was considerably in excess of valve opening pressure.

INVESTIGATION OF THE DISCHARGE RATE OF A FUEL-INJECTION SYSTEM (N.A.C.A. Technical Note No. 373, 1931).—Fuel discharged by a cam-operated pump and automatic injection valve was collected in a rotating receiver. Ninety-eight percent of the fuel was discharged during an interval of 25 deg., whereas the period from start to stop of the fuel spray was 68 deg.

EFFECT OF HIGH VELOCITIES ON DISTRIBUTION AND PENETRATION OF A FUEL SPRAY (N.A.C.A. Technical Note No. 376, 1931).—High-speed moving pictures were taken of the formation and development of fuel sprays from an automatic injection valve. Sprays were injected normal to, and counter to, air at velocities from 0 to 800 ft. per sec. Air was at atmospheric temperature and pressure. Results were: 1. Air velocities of approximately 400 ft. per sec. are necessary to break up the central core of a fuel spray from a single, round-hole orifice during the injection period in a high-speed compression-ignition engine. 2. Lower velocities have little effect on the main body of the spray during injection, but break up the spray at the end of injection.

PHOTOMICROGRAPHIC STUDIES OF FUEL SPRAYS (N.A.C.A. Technical Note No. 424, 1932).—Photomicrographs were taken of fuel sprays injected into air at various densities in order to study spray structure and the stages in the atomization of fuel. Fuel leaves the nozzle as a solid column, is ruffled and then torn into small, irregular ligaments by action of the air. The ligaments quickly are broken into drops by the surface tension of the fuel. The photomicrographs also show that dispersion of a fuel spray at a given distance from the nozzle increases with increase in jet velocity, or increase in air density.

4. SUPERCHARGING

SUPERCHARGING.—The amount of energy that can be produced in a given size of Diesel cylinder, charged with air in the ordinary way, is limited by the amount of fuel that can be burned efficiently in the air available. By artificially increasing the amount of air, more fuel can be burned efficiently and more energy produced. The artificial increase of combustion air supplied to the cylinder before fuel injection is called supercharging.

The best known systems of supercharging are the Rateau and the Buchi. Both provide the supercharging air by means of turbine-driven centrifugal blowers, using exhaust gases of the engine as motive fluid. By proper adjustment of compression ratio and air excess, supercharged engines are not subjected to higher temperatures or higher maximum pressures than exist in similar normally-charged engines, although average cylinder pressures are higher. Pressure of the supercharging air is about equal to pressure of the exhaust gases, generally 18 to 20 lb. per sq. in., absolute. Engine capacity is increased 30 to 40%.

According to Max Rotter (*Trans.* A.S.M.E. OGP-53-9, 1931), in the Buchi system the speed of the blower and, therefore, the pressure of supercharging air automatically adjust themselves to the existing engine load, due to variation of exhaust pressure with load. High efficiency results, with a flat fuel-consumption curve. The Buchi system is suitable for continuous operation, and has been applied exclusively to four-cycle engines.

The Sulzer system primarily is intended to increase overload capacity of a two-cycle Diesel engine. Supercharging air is furnished by a compressor, attached to the engine, whose action is so controlled by the engine governor that supercharging occurs only at the higher engine loads. Automatic valves in the upper tier of Sulzer scavenging ports for normal scavenging and charging are supplemented by timed valves, controlling delivery of supercharging air to the cylinder. Air furnished by the supercharging compressor is admitted only toward the end of the charging period. Engine capacity temporarily may be increased as much as with the Buchi system, but fuel efficiency of the engine is affected adversely about 5%, due to the power consumed by the supercharging pump, even when not in action.

Another system of supercharging has been applied to German 4-cycle engines to accomplish the same purposes as the Sulzer system. Supercharging air is furnished by a compressor attached to the engine. In addition to the normal admission valve, a timed supplementary piston valve delivers excess atmospheric air to the admission valve during the earlier part, and of supercharging air during the latter part, of the suction stroke. Dimensions and power consumption of the charging compressor are less than they would be if all charging air were furnished by it, but fuel consumption at all loads is increased.

Complete tests of the Buchi system were made by Prof. A. Stodola, Zurich, Switzerland. (See *The Shipbuilder*, June, 1928, p. 413.) The tests were made on a 4-cycle, 6-cylinder engine, standard

except for the addition of the exhaust-gas turbo-blower supercharger. The normal rating of the engine without supercharger was 850 Hp. at 167 r.p.m. With the turbo-blower, output was increased to 1275 Hp., with combustion and exhaust temperatures the same as at normal rating. Even at 1652 B.Hp. output, engine exhaust was absolutely clear. At 1275 B.Hp., fuel consumption was 0.391 lb. per B.Hp.-hr. The lower calorific value of the fuel used was 18,240 B.t.u. per lb. The same engine, without supercharging, consumed 0.408 lb. of fuel per B.Hp. per hr. The turbo-supercharging improved fuel consumption of the engine tested by about 4%, due to increase in mechanical efficiency from 72 to 80.2% referred to normal rating of the engine.

Another feature of the Buchi system is the reduced amount of heat passing through walls of piston, cylinder, and cylinder head, *i.e.*, 1620 B.t.u. per B.Hp. per hr. at full load. An engine of a similar speed, with uncooled pistons and a fuel consumption of 0.41 lb. per B.Hp.-hr., showed a loss through the cooling water of 28.7% of the heat value, equal to 2130 B.t.u. per B.Hp. per hr.

5. HEAT RECOVERY

The heat energy of the fuel supplied to the Diesel engine finally is distributed as actual work at the engine crank-shaft, frictional losses, heating of cooling water in engine jackets, heat in the exhaust gases, and miscellaneous heat radiation to atmosphere. Table 2 gives a typical heat balance. The jacket water heat, 27% of the heat in the fuel, almost equals the heat utilized as work. Jacket water heat can be utilized directly in the form of warm water. About 40% of the exhaust, or 11% of the heat of the fuel, can be recovered by passing it through a simple heat exchanger with a moderate amount of surface. By thus utilizing jacket and exhaust heat, an additional recovery of about 38% may be obtained, corresponding to about 4500 B.t.u. per kw.-hr.

Although the exchanger reclaims less than one-third of the total heat recovered, it is important in that it raises the temperature of the water higher than would be safe in the engine jackets. Thus if water enters the engine jackets at 40° F. and leaves them at 140°, it will leave the exhaust heat reclaimer at about 180° F. This final temperature rise makes the water useful where a temperature of 140° would be too low. Exhaust heat also can be used to make steam, but only part of the jacket water can be evaporated. However, if the quantity of steam needed is small, and the rest of the jacket water can be used as warm water, all the heat can be recovered.

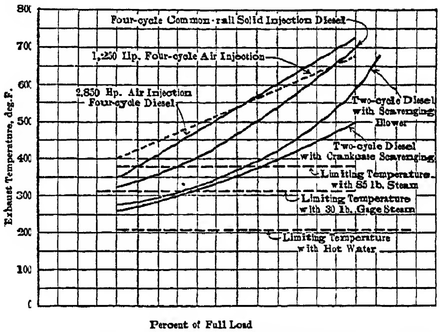


Fig. Representative Exhaust Temperatures of Various Diesel Engines

EXHAUST HEAT RECOVERY.—At full load (at the point a heater could be installed) the exhaust of a heavy-duty 4-cycle engine will be at 650° to 800° F. A 2-cycle engine exhaust will be at 450° to 600° F. Heat recovered will vary in proportion to temperature drop in exhaust gas as it passes through the boiler. More heat can be recovered in heating water than in generating steam. In steam generation, heat recovery is greater with low-pressure than with high-pressure steam, as the temperature limits are lower in generating low-pressure as compared with high-pressure steam. For practical purposes, gas temperature leaving an exhaust-heat boiler should be above 250° F., although a temperature as low as 200° F. will not cause the water vapor to condense. Fig. 16 (*Diesel Power*, Dec., 1933) shows representative exhaust temperatures of different types of engines at various loadings.

Table 2.—Heat Balance of Diesel Engine

	B.t.u. per B.Hp.-hr.	Percent	Useful Percent	Heat Recoverable, B.t.u. per B.Hp.-hr.
Brake work	2545	33	33	....
Friction	680	9	....	....
Jacket water heat	2100	27	27	2100
Exhaust gas heat	2200	28	11	880
Radiation, etc.	255	3	....	....
Total	7780	100	71	2980 = 4500 B.t.u. per kw-hr.

The quantity of heat recoverable from the exhaust gas at any load may be estimated roughly from the formula

$$Q = (\text{B.Hp.} \times C \times D)/4 \quad . \quad . \quad . \quad [5]$$

where Q = recoverable heat, B.t.u.; B.Hp. = brake horsepower at the load in question; C = a constant (approximately 12 for 4-cycle engines, 20 for 2-cycle engines); D = temperature drop of exhaust gases through heater, deg. F.

Exhaust heaters may be fire-tube, water-tube or thimble-type, and in general, are similar to the corresponding types of fuel-fired boilers, except that heating surfaces are larger. They are compact and inexpensive. When used with modern high-grade Diesel engines, they are quite easy to keep clean, as little or no soot is deposited. They also are effective silencers.

The recovery of waste heat from Diesel engines is especially valuable in mild weather, if it permits shutting down a fuel-burning boiler or discontinuing purchased steam. A good use for this heat is in a combination steam and Diesel plant, where the Diesel jacket water at, say, 140° F. goes directly to the feedwater heater, thus recovering about 27% of the heat of the Diesel fuel. The utilization of heat in the jacket water serves a double purpose. It effects a reduction in heat required from other sources, and also eliminates equipment for cooling jacket water, or the expense of wasting it.

Table 3 gives the results of tests of a Foster-Wheeler water-tube muffler-boiler, with 1400 sq. ft. of extended heating surface, connected to a 4000-Hp. Hooven-Owens-Rentschler double-acting, 2-cycle, 4-cylinder, air-injection Diesel engine.

Table 3.—Test of Boiler Utilizing Diesel Engine Exhaust Gases

Steam pressure, lb. per sq. in., gage	43.5
Back pressure on engine exhaust, in. of water	6.00
Temperature gases entering boiler, deg. F.	538
Temperature of gases leaving boiler, deg. F.	346
Temperature of water entering boiler economizer, deg. F.	168
Indicated horsepower	4,812
Brake horsepower	4,019.5
Fuel consumption, lb. per hr.	1,669.5
" " lb. per I.Hp.-hr.	0.346
" " lb. B.Hp.-hr.	0.438
Main engine speed, r.p.m.	110
Water fed to boiler, lb. per hr.	2,820
Exhaust gases, lb. per hr.	85,000
Stroke of main engine, in.	47.25
Diameter of cylinders, main engine, in.	27.5
Thermal value of fuel, B.t.u. per lb.	19,000

6. COMBINED DIESEL-STEAM POWER PLANTS

Diesel engines have been used in conjunction with steam engines to obtain greater overall economy in production of both power and heat. When all the exhaust of a steam power unit can be used for processes requiring heat, power is produced as a by-product. The boiler plant practically would be as large and burn as much fuel if it produced steam solely for heating. If the demand for heat is not large enough or uniform enough to utilize all the exhaust from engines or turbines, some exhaust heat is wasted. This waste may be reduced by the addition of Diesel engine capacity to carry the load when there is no demand for heat, or to carry that portion of the load that exceeds the demand for exhaust steam.

The advantages of such a combination plant are: 1. The Diesel engine is compact and self-contained. It easily can be installed in an existing plant. 2. It need not interfere with operation of the steam plant. 3. Investment is moderate. 4. The same staff can operate and maintain steam and Diesel equipment. 5. The Diesel equipment is an emergency reserve for the steam plant. 6. Heat in Diesel jacket water generally can be used to advantage. If desired, an exhaust gas boiler may be used to recover the Diesel exhaust heat. 7. The foregoing combine with the fuel efficiency of the Diesel engine to give low operating costs.

Exhaust-use Factor

The economy of a steam plant supplying power and heat depends on the exhaust-use factor, *i.e.*, a ratio representing the proportion of the total steam power that is by-product. Before adding a Diesel engine to an existing steam power plant, the exhaust-use factor must be determined accurately.

In determining the amount of by-product power, figures for power and heat requirement should be measured over short periods for each different operating condition. Daily

averages may lead to large errors; longer periods, as a week or a month, are practically worthless, as long periods disregard the diversity of demands for power and heat. Even if daily average process or heating steam demand exceeds average power demand, there always are periods when conditions are reversed. When power demand exceeds heating demand, some or all of the exhaust steam is wasted. Hence, the exhaust-use factor should be determined by readings for periods of one hour or less, taken at various times of the day, week and year.

The addition of a Diesel engine to a steam-power plant may improve overall economy by: 1. Increasing exhaust-use factor of steam plant. 2. Replacing expensive *prime steam power* (steam engine power corresponding to wasted exhaust steam) with less expensive Diesel power. In such case, steam units are loaded to the point of best overall economy for the entire plant, and the Diesel generates the remainder of the power.

In a study of the joint use of the Diesel and steam engines in a definite project, it is best to select arbitrarily several different combinations of commercial sizes of Diesel and steam engines. With existing steam plant, only the amount of Diesel power to be added is determined. For each combination a series of charts is made for all seasons of the year, based on curves of hourly demands for power and heat. These charts should show: 1. Amount of steam needed to supply heating load. 2. Amount of by-product power that can be produced from that steam. 3. Amount of additional power that Diesel equipment can generate. 4. Amount of prime steam power, if any, that must be produced if Diesel plant cannot generate all power required in excess of by-product steam power. From these charts may be computed the quantities of coal and the Diesel fuel that will be used, thus determining for each combination the annual cost of boiler and Diesel fuel. To the fuel costs must be added cost of attendance, repairs, ash-disposal, water and supplies. Costs will vary with each combination, depending on relative capacity and kw-hr. output of the steam and Diesel parts of the combination. The sum of these items is total annual operating cost, to which are added fixed charges, as taxes, interest, amortization and insurance. The resulting overall annual costs of the several combinations then can be compared.

7. AUTOMATIC DIESEL PLANTS

Automatic electric-generating units driven by gasoline engines, as farm-lighting outfits, have been used for twenty years; the automatic multiple-unit Diesel-electric power plant is a recent development. Up to August, 1935, four such plants, based upon patents of C. F. Strong, had been installed. All supply direct current.

This type of plant consists of several Diesel-electric generating units, operating in parallel with a storage battery and a battery booster, all controlled automatically. The generating units start and stop in accordance with load requirements; line voltage is maintained constant, regardless of load surges; all supply services to engines are fully controlled and safeguarded. Engines are started by supplying battery current to their

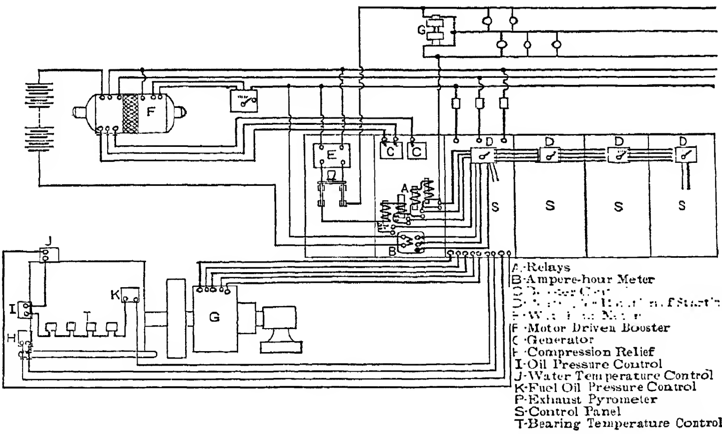


Fig. 17. Diagram of Automatic Control System

generators, the battery-booster regulation maintaining constant voltage on the line during starting. Battery and booster also absorb line load surges, and prevent flicker of lights.

To insure reliability, a spare unit is installed, with controls so arranged that any unit failing to start, or developing trouble while running, automatically is stopped and disconnected, and the spare unit started in its place in less than 30 seconds. The storage battery meanwhile supplies any deficiency in power. A unit disconnected from the circuit, due to trouble, automatically transmits a signal to a central office.

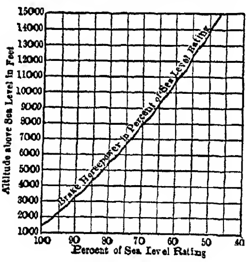
A diagram of the Strong automatic control system for a 4-unit plant is shown in Fig. 17. The control panel of the switchboard has, for each engine, a spring-loaded relay in series with the line. The relays are adjusted to act with an increase of load on the building feeder lines, and cause the engines to start. The switchboard relays are set to keep but one engine in operation if the current flowing to the building lines does not exceed a predetermined value. When current increases beyond the capacity of the first engine, second, third and fourth units are started successively in accordance with the load. As load decreases the engines are stopped in sequence. The operating units thus have an economical load factor. While idle, exhaust valves are held open by a spring opposing a solenoid, to prevent compression in the cylinders. Also, the governor mechanism holds open the by-pass valves of the fuel pumps, preventing injection of fuel into the cylinders. When the load requires starting an engine, the generator acts as a motor, using battery current, bringing the engine up to about half speed. The solenoid operating the compression relief mechanism then is energized and full compression thrown on the engine.

To insure continuity of service, each generating unit is entirely independent, the only common points being the main fuel storage tank and the switchboard bus-bars. Each generating unit is controlled by its own automatic switching panel. Safeguards are provided for jacket water temperature, lubricating oil pressure and main bearing temperature. In case of derangement the engine affected is stopped automatically, and the spare unit started in its place.

8. STANDARD DIESEL ENGINE PRACTICE

The following is an abstract of tentative (1935) standard practices of the Diesel Engine Manufacturers' Assoc., as given in the book of standards of the association.

STANDARD SEA-LEVEL RATING of a Diesel engine is the net brake horsepower that engine will deliver continuously when in good operating condition and located at not over 1500 ft. above sea level, with atmospheric temperature not over 90° F. and barometric pressure not less than 28.25 in. of mercury. The standard rating must be such that the engine will deliver an output of 110% of rating, with safe operating temperatures for two out of any 24 hours.



rsepowe
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Altitudes

Net Brake Horsepower is horsepower delivered to the engine crank-shaft coupling, less any power consumed by certain separately-driven auxiliaries, if used, *i.e.*, injection-air compressor (for air-injection type engines), scavenging air pump or blower (for supercharged engines), and pumps for circulating lubricating or piston-cooling oil through engine or oil cooler. No deductions are made for power to drive such auxiliaries when mechanically driven by the engine. No deductions are made for auxiliaries intermittently operated, or governed in size or operation by special conditions of plant apart from the engine, *viz.*, centrifuges, compressors for starting-air, and pumps for oil-transfer, circulating water and raw water.

RATINGS AT HIGHER ALTITUDES.—The net brake horsepower capacities of a normal supercharged Diesel engine at various altitudes, in percent of sea-level rating, are given in Fig. 18. The power that a Diesel engine can deliver decreases with increase in altitude. The resultant decrease in atmospheric density decreases horsepower by: 1. Decreasing amount of oxygen available for combustion. 2. Decreasing possible rate of heat absorption. The reduction in rating due to an increase in altitude is not exactly in proportion to the decrease in barometric pressure because horsepower required to overcome internal friction remains nearly constant for all altitudes.

CAPACITY AND FUEL-CONSUMPTION GUARANTEES.—Guarantees of brake horsepower capacity and fuel consumption are based on tests conducted by the engine builder on his shop test floor. Brake horsepower capacity guarantees for non-super-

charged engines are contingent on an air-intake pressure of not less than 28.25 in. of mercury, and air-intake temperature of not over 90° F. Sea-level fuel-consumption guarantees are made in fractions of a pound per net brake horsepower at half, three-quarters and full loads, when operating at rated revolutions per minute. Such guarantees are referred to a fuel oil of a high heat value of 19,000 B.t.u. per lb. All sea-level fuel-consumption guarantees are contingent on the following conditions: 1. Intake-air temperature between 40° and 90° F., inclusive. 2. Barometric pressure of intake air between 28.25 and 30 in. of mercury, inclusive. 3. Fuel oil to conform to engine builder's specifications for type of engine involved. In adopting these standard practices, the engine builder absorbs the differences in performance caused by three variables. Since these variables may operate adversely in a cumulative way, fuel-consumption guarantees are subject to a tolerance of 5%.

Standard fuel consumption guarantees are not made for quarter load, because tests at this load are inaccurate and not reproducible. The fuel consumption rate at the guarantee points increases with increase in altitude, principally because of the different full-load ratings. Fuel consumption for altitude operation is guaranteed, therefore, on a basis of the rate at corresponding *horsepower* load under sea-level conditions.

FIELD TESTS FOR HORSEPOWER CAPACITY AND FUEL CONSUMPTION.—Horsepower capacity, fuel consumption or other tests made after the engine is installed in its ultimate location, are made in accordance with the Field-test Code.

LUBRICATING OIL CONSUMPTION.—The same Diesel engine if installed in several plants in succession, would show different rates of lubricating oil consumption, so material is the influence of plant conditions and operation on lubricating requirements. These conditions are beyond the control of the engine builder, and it is not feasible to guarantee lubricating oil consumption.

STANDARD EQUIPMENT FOR STATIONARY ENGINES.—The standard of minimum equipment furnished with stationary engines is as follows:

Engine fly-wheel, unless weighted rotor (fly-wheel-type) generator is to be driven; extension shaft and outboard bearing; engine piping to common inlet and outlet nozzles; exhaust manifold or individual exhaust pipes to conduit below floor; air-inlet manifold; strainer-type fuel oil filter; strainer-type lubricating oil filter; lubricating oil sump tank; force-feed lubricator (if required by design); all necessary lubricating oil pumps and coolers; all necessary piston-cooling oil pumps and coolers (for oil-cooled pistons); lubricating oil pressure gage; thermometers for engine circulating water; thermometers for piston-cooling oil or water (if pistons are cooled); piston-oil cooler (for oil-cooled pistons); safety, or relief, valves; synchronizing device, hand or pneumatically operated; a set of tools, but not including any tools that can be purchased in the open market (these are not duplicated for a plant where two or more engines are installed); a minimum set of spare parts; such platform and steps or stairs as may be required; drilled and tapped holes for exhaust-temperature measuring devices, but not including such devices; drilled and tapped holes for attachment of indicator, but not including indicator, indicator cocks or reducing motion; complete foundation bolts, including generator and exciter bolts if these units are to be driven; the services of an erection superintendent, from the time the building and foundations are ready and the machinery is at the site, until 7 days after the equipment is ready for commercial operation.

STANDARD GOVERNOR PERFORMANCE.—Two types of governors commonly are furnished: 1. Centrifugally-powered type. 2. Relay-powered type. The choice depends on the class of service.

Centrifugally-powered governors will conform to the following standards: 1. Governor will control engine speed to within 3% above or below mean speed, at all loads between no-load and full-load for gradual changes in load. 2. Governor will control the momentary engine speed change to within 5% above or below mean speed, at all loads between no-load and full-load, for sudden changes in load, where engines are intended for parallel operation. 3. Stabilization period will not exceed 20 seconds. 4. Sensitiveness will not exceed $^1/_4$ of 1%. 5. Under constant load there will be no hunting; with the load changing there will be no persistent false governor movements.

Relay-powered governors will conform to the following standards: 1. Governor will control the engine speed to within 3% above or below the mean speed, at all loads between no-load and full-load for gradual changes in load. 2. Governor will control the momentary engine speed change to within 5% above or below the mean speed, at all loads between no-load and full-load, for sudden changes in load, where engines are intended for parallel operation. 3. Stabilization period will not exceed 10 seconds. 4. Sensitiveness will not exceed $^1/_{100}$ of 1%. 5. Under constant load there will be no hunting; with the load changing there will be no persistent false governor movements. 6. The speed-droop will be adjustable during operation from zero to 6%. 7. Average frequency control, commercially accurate, may be specified when the governor is adjusted to zero speed-droop.

STANDARD OF PARALLEL OPERATION.—It is standard practice to build engine-generator units capable of running in parallel with other generating units, provided that

such other generating units are: 1. Capable of operating in parallel with each other. 2. Equipped with proper damper windings. 3. Equipped with voltage regulators with adequate cross-current compensation.

TORSIONAL VIBRATION AND CRITICAL SPEEDS.—Standard practice is so to co-ordinate the assembly of each engine and driven equipment that the peak of any deleterious critical speed for a constant-speed unit will not fall within a range of speed of 10% of the rated speed of the unit.

9. SELECTION AND INSTALLATION OF STATIONARY DIESEL ENGINES

(Abstracted from Standard Practices of Diesel Engine Manufacturers' Assoc., 1935.)

SELECTION OF SIZES.—The general principles to be followed in the selecting of Diesel engines are: 1. The most efficient load range, from the standpoint of fuel economy, is from half load to full load. 2. A small engine is practically as efficient as a large one. 3. Dependence should not be placed on overload capacity to handle regularly recurring peak loads. 4. Additional engines easily can be added at any future date.

LOAD VARIATION is presented most effectively in curve form, using successive power demands as ordinates and hours of the day as abscissæ. Number and size of engines should be selected to conform with load conditions.

OVERLOAD CAPACITY.—While Diesel engines are rated to allow some overload capacity for short periods, this should not be utilized to handle routine peak loads. Any predictable load should be within the combined ratings of the engines.

POWER TRANSMISSION.—Diesel engines may be direct-connected to driven machinery if starting torque can be limited to less than 50% of full-load torque.

CLUTCHES.—Torque delivered by a Diesel engine reaches full-load value at 25% to 35% of full load speed. Machinery requiring a high starting torque, but requiring only normal torque immediately after starting, may be driven by Diesel engines through clutches. A flour mill line-shaft is a typical instance.

CLUTCH PULLEYS.—For belt drives clutch pulleys should be on the driven, rather than on the driving, shaft. If driving shaft location is essential for the clutch, driving pulley should be on a quill.

ELECTRIC, HYDRAULIC AND PNEUMATIC DRIVES.—Where prolonged high starting torque is required, electrical, hydraulic, pneumatic, or a combination clutch and sliding gear drive is necessary. Such transmissions are used in Diesel locomotives, shovels and drag lines.

MACHINERY LAYOUT.—In designing a Diesel plant, machinery should be laid out first and buildings designed around it. Good natural light contributes to lower operating costs. Ample space should be allowed around each engine. When engines and auxiliaries have been worked into the floor plan, a work room and store room should be added. All floors should drain to one or more drains. The engine builder's specifications of minimum headroom from top of engine to bottom of crane hook should be observed.

SPACE FOR FUTURE UNITS.—If more horsepower possibly will be needed at a future date, space should be allowed for it in the building, or the building so designed that it can be extended. Doorways should permit the introduction of future units and parts for present units.

CRANES.—The best crane layout provides runways on either side of the building for a traveling crane that can serve the entire floor area. The crane may be hand operated, but with two or more engines, cost and time of erection can be reduced if the hoist is motor-driven. Individual cranes may be mounted on I-beam tracks over each engine center line, extending sufficiently beyond units to allow easy landing of parts.

ARRANGEMENT OF TWO OR MORE ENGINES.—The most compact arrangement for two or more engines is side by side. This permits locating engine auxiliaries at one side of the room and driven apparatus with appurtenances at the opposite side.

FOUNDATIONS.—Manufacturers supply foundation drawings with each engine sent out. These drawings usually apply for a bearing of hard pan, confined gravel, hard clay, or rock. If such firm sub-soil lies considerably below the minimum depth required, the foundation may be designed in the regular way, but supported on piling. An alternative construction is a reinforced slab supported by reinforced concrete pillars. An expert foundation engineer should be consulted where special conditions exist.

Minimum Depth for Foundations should be sufficient to prevent settling from frost, vibration, or influence of loads borne by adjacent ground, irrespective of firmness of sub-soil. If solid rock exists at less than the recommended depth, excavation should be

carried a foot or two into the rock to secure firm anchorage; in no event should absurdly shallow foundation be allowed.

Foundation Over Underground Water.—An engine foundation never should be located over underground water without the advice of an experienced foundation expert, nor without full details being given to the engine builder.

Foundation Bolt Template.—The bolts should be supported by a template, carefully leveled, resting on the foundation forms. Each bolt should be surrounded with iron pipe or stove pipe, supported by the bottom washer, and stopped with waste at the upper end to prevent entrance of concrete.

Pouring the Concrete.—The foundation should be of concrete, one part cement, two parts sand, and four parts broken stone or gravel (1 in. maximum). The entire foundation should be poured at one time, with no more interruptions than are required for proper spading and ramming. The top should be level, and left rough and clean for grouting. After pouring, the top should be covered and wet down twice daily until the forms are removed at the end of the third or fourth day. The engine should not be placed on the foundation until ten days have elapsed, nor operated until another ten days have passed.

Leveling the Engine.—The engine should be leveled by wedges resting on steel plates on the top of the foundation, and grouted in with a thin mixture of one part cement and two parts sand. The grout should fill the pipes around the foundation bolts. The leveling wedges should be removed after grout has thoroughly set.

PIPING AND WIRING should be located in trenches with side walls and floors of concrete at least 4 in. thick, and covered by removable steel plates. Inlet, exhaust, fuel oil, lubricating oil, circulating water, and air piping trenches should be drained. Generator lead trenches should drain to their respective fly-wheel and generator pits through 3-in. drains. All piping should be supported in trenches by racks. To avoid fire hazard, fuel and lubricating oil piping should not be located in the same trench with exhaust piping.

INSULATING FOUNDATIONS AGAINST VIBRATION.—Engines in residential districts, hotels, office buildings, or department stores should be set on cork-insulated foundations to minimize vibration. All piping from engine to off-foundation locations should have flexible joints to prevent transmission of vibration. (See Section on Vibration, vol. 3 of this series.)

FUEL STORAGE.—Fuel oil storage tanks may be of steel or concrete. Concrete tanks should be painted inside to prevent absorption of oil. If near a railroad siding, the minimum aggregate fuel storage should be 1 1/4 to 1 1/2 the capacity of an average tank car. If the siding is some distance from the fuel storage, an unloading pump at the siding may be advisable.

Heating Fuel Oils.—The trend is toward heavier fuel oils. Even at present (1935) it often is economical to burn an oil which may need heating in cold weather before it can be pumped from an outside tank. Heating coils, therefore, are recommended for storage tanks. They should be arranged around the suction pipe and have no joints within the tank.

Fuel Level Indicators.—If records are to be kept, a sounding pipe and float-operated indicator are essential. If tank is located outside, the indicator should be enclosed.

Locating Fuel Tanks.—Storage tanks are best located above ground. Where local conditions call for a buried tank it should be set on a firm foundation and surrounded with soft earth, well-tamped sand, or encased in concrete. With a concrete casing, the tank manhole should be accessible in the bottom of a well. Before locating a fuel storage or day tank, local laws and ordinances applying should be consulted.

FUEL HANDLING SYSTEMS.—A common system of handling fuel oil is by means of individual day tanks for each engine. Oil is pumped from the main storage to the day tank, whence it flows by gravity to the engine pump. Engines sometimes are equipped with a supply pump which draws fuel directly from the main storage tanks.

Day Tanks.—Day tanks above the floor line should have gage glasses to permit measurement of fuel consumed. Day tanks below the floor should have float or pressure-operated gages.

Fuel Oil Piping and connections should be made up tighter than is customary for water piping. Wrought-iron, steel or preferably brass or copper pipe may be used for lines between fuel strainer and engine. Unions should be of brass without gaskets. Piping should be located so that oil cannot come in contact with hot exhaust piping.

Fuel Oil Filtering or straining usually is sufficient purification for fuel oils; poorer grades often must be centrifuged. Filtering elements may be metal screens, cloth bags, or thin washers held tightly together by king bolts. Filters should be of duplex construction so that one half may be cleaned while the other half is in operation.

Care of Fuel Pumps, spray nozzle valves, air injection compressors and scavenging compressors should be carried on according to a regular schedule. See p. 12–32.

AIR FILTERS.—If air drawn into an engine contains much dirt and dust, liner wear results. A good air filter will remove practically all dust and dirt from intake air without materially lowering air pressure at intake valves.

COMPRESSION PRESSURE.—A Diesel engine will not function properly unless compression pressure is maintained at or near the point specified by its builder. Compression pressure may decrease because of valves seating poorly, too great piston clearance, or poor piston rings. Users should learn from the engine builder the proper method of indicating engine cylinders for compression pressure, and of maintaining valves, piston clearance and piston rings.

EXHAUST TEMPERATURES.—Relative cylinder loading can be checked by observing exhaust temperatures of the several cylinders, which should be not more than 50° F. apart. Temperatures can be measured either by exhaust pyrometers or suitable thermometers.

EXHAUST PIPING should be as direct as possible and free from 90 deg. short-turn elbows. The pipe to the muffler never should be smaller than the size of connection on exhaust manifold on the engine. Exhaust lines preferably should be standard steel pipe with screwed connections up to 6-in. pipe, and flanged connections for 6-in. pipe and over. Cast-iron pipe may be used. The line must be designed to allow for the expansion due to a 1000° F. variation in temperature.

MUFFLERS usually are of cast iron or of sheet steel, but can be any kind of a chamber which promotes gradual dispersion of exhaust gases. Concrete chambers filled with stone or with coke sprinkled with water are effective mufflers. Cast-iron or steel mufflers, if connected to a straight length of exhaust piping, should be on rollers or provided with expansion joints.

STARTING AIR SYSTEM.—On low-pressure starting systems, such as are used for mechanical injection engines, standard lap-welded steel pipe, with globe valves of the best materials and workmanship, should be used. On high-pressure (500–1000 lb. per sq. in.) starting systems, seamless drawn steel tubing or copper tubing should be used, with needle valves.

Tanks for starting air should meet the requirements of the insurance companies and comply with state and local codes.

LUBRICATING OIL SYSTEMS.—Lubricating oil systems for Diesel engines vary somewhat, but the main connecting-rod, and cam-shaft bearings generally are force-feed lubricated; cylinders, including air-compressor cylinders, usually are drop-feed lubricated. Pure mineral oil should be used for power cylinder lubrication and mineral oil, or specially compounded compressor oil should be used, sparingly, in injection and scavenging compressor cylinders. Physical specifications for crank-case lubricating oil do not furnish a sure indication of its suitability. Users must rely on the experience with various oils of the engine builder, and should choose from oils recommended by him.

Clarification.—Lubricating oil may be clarified by straining through wire gauze or filtering cloth; by settling, with or without chemicals; or by centrifuging. Centrifuging may be: 1. By the batch, all oil of the engine system being run through a centrifuge periodically. 2. By the continuous method, wherein a small amount of the oil continuously is by-passed through a centrifuge which operates whenever the engine operates.

Lubricating Oil Piping including centrifuge system pipe, may be black iron pipe; copper or brass pipe is preferable.

Acidity of Lubricating Oil.—When high-sulphur fuel oil is used, the crank-case oil should be examined frequently for acidity. Manufacturers can recommend treatment to neutralize acid oil.

Lubricating Oil Coolers.—The efficiency of lubrication is not impaired if crank-case oil remains below 135° F. With a tendency toward higher temperatures, a lubricating oil cooler must be used, and the oil cooled to 90 to 100° F. A rising temperature of oil as it drains to the sump indicates a hot bearing, and oil pressure should be raised until the engine can be shut down and investigated.

COOLING WATER.—In the simplest cooling water system, water passes through the engine system once and then is rejected to waste. This system is feasible only where a large and inexpensive supply of suitable soft water or sea water exists. The sodium salts of sea water do not tend to give trouble, while salts of calcium, magnesium, aluminum and iron frequently do. Surface water, water in lakes, and water in rivers frequently hold salts in solution by virtue of dissolved CO_2. In a circulating water system, the CO_2 may become liberated and the salts deposited. Such deposits defeat the purpose of the cooling system by insulating the metal surfaces and retarding the flow of heat. Water should be analyzed if the least doubt exists about its suitability for cooling system use.

Treatment of Hard Water.—Hard water may be softened by passing through a layer of Zeolite, which absorbs the calcium, magnesium, aluminum and iron, in exchange for sodium. The Zeolite is rejuvenated by passing a solution of common salt through it. This system is satisfactory if properly supervised. The water should be tested frequently for hardness, and engine jackets should be inspected frequently for scale. The temperature of water from cylinders and heads usually should not exceed 120° F. maximum, as the tendency of salts to precipitate increases at higher temperatures.

Scale Removal from engine jackets usually can be effected by a treatment with an acid solution. Manufacturers can recommend the exact procedure in such cases.

Raw Water Cooling Systems.—With a limited supply of fairly good water, engine circulating water can be sent over a cooling tower or through a spray pond. See p. 9–16. Such systems should be completely drained and replenished with fresh water at regular intervals, as hardness in the water, even if moderate at the start, continually concentrates due to additions of make-up water and evaporation of pure water.

Double Circuit Cooling Systems.—Engine cooling water passes through a heat exchanger, where it is cooled by raw water which is recooled in a cooling tower or spray pond. As the loss of engine circulating water in a properly set-up system is slight, the best grade of water can be used, *viz.*, rain water, distilled water, or water thoroughly treated. The heat exchanger may be a series of pipe coils with raw water flowing outside, or it may be a shell-and-tube cooler, with raw water passing through the tubes.

Radiator Cooling is a variation of the double circulating system. The soft water passes through a radiator, similar to an automobile radiator, and is cooled by air, circulated through the radiator by a fan. The radiator system can be thoroughly satisfactory.

CIRCULATING WATER PIPING AND TEMPERATURES.—Circulating water should be under a static head at the circulating water header on the engine. If the circulating water is at all hard, discharge from the cylinder head and piston cooling system should not exceed 120° F. No bad effect is traceable to somewhat higher discharge temperatures with soft water. Air compressor cylinders should be kept as cool as possible. With equal loads in power cylinders, outlet water temperatures should be within 5° F. of the average.

The spread between the inlet and outlet circulating water temperatures should be held within 30 to 40° F. depending on the engine. A greater spread will result in strains in the cylinders. If double-circuit circulation is used, the temperature rise through the engine cylinders and heads should be held to one-half the above, since the temperature of the raw water is not lower than can be produced in a cooling tower.

Amount of Water which should be circulated in a single-circuit system, with inlet water at 90° F. and outlet water at 120° F., is about 0.2 gal. per min. per rated Hp. This amount should be increased to about 0.4 gal. per min. per rated Hp. for double circuit systems, with the temperature rise limited to 15° F.

Circulating Water Piping should be of wrought iron or steel, galvanized. The best gaskets are rubber with cloth insertions.

Stopping and Starting.—The circulating water system should operate for 20 minutes after stopping an engine. The flow should not be reduced suddenly, as, for instance, when the engine is running too cold. Cold water never should be turned into a hot engine. If the installation is in a freezing climate, circulating water should be drained from all parts of the engine, whenever it is shut down for more than short periods.

TYPICAL LAYOUT OF DIESEL POWER PLANT.—Figs. 19 and 20 from Hammond, Diesel Power Plant Layout, show a plant layout, designed for economical construction and operation, with provision for plant extension.

SPECIFICATION FOR DIESEL-ELECTRIC POWER PLANTS.—A form of specification in considerable detail, for Diesel-electric generating plants, will be found in "Standard Practices," published (1935) by the Diesel Engine Manufacturers' Assoc., New York.

10. VARIATIONS IN DIESEL ENGINE DETAILS *

ATTACHED PUMPS.—Smaller engines usually have built-in circulating water pumps. Larger engines usually require the installation of motor-driven pumps. Some 2-cycle engines have built-in scavenging pumps or blowers; others require motor-driven equipment.

FLY-WHEELS.—The dimensions and weight of fly-wheels for Diesel engines depend on the equipment to be driven by the engine; its engineering data must be analyzed and evaluated with respect to parallel operation of alternating-current generators and torsional vibration. Fly-wheels may be of one- or two-piece construction, and must have suitable

* Abstracted from Tentative Standard Practices of Diesel Engine Manufacturers Association, 1935.

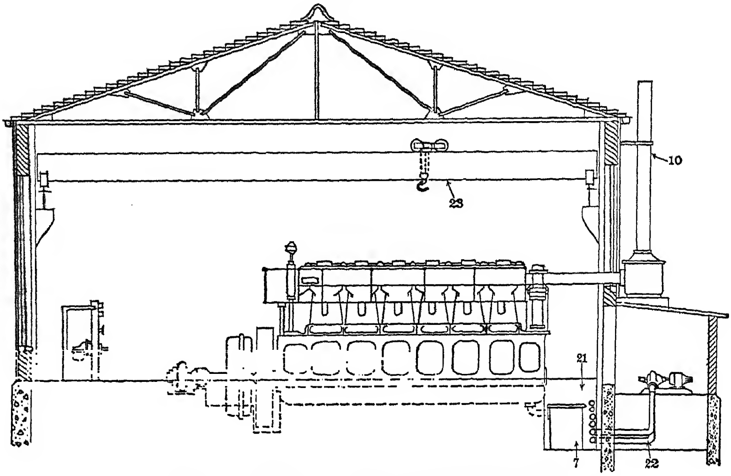
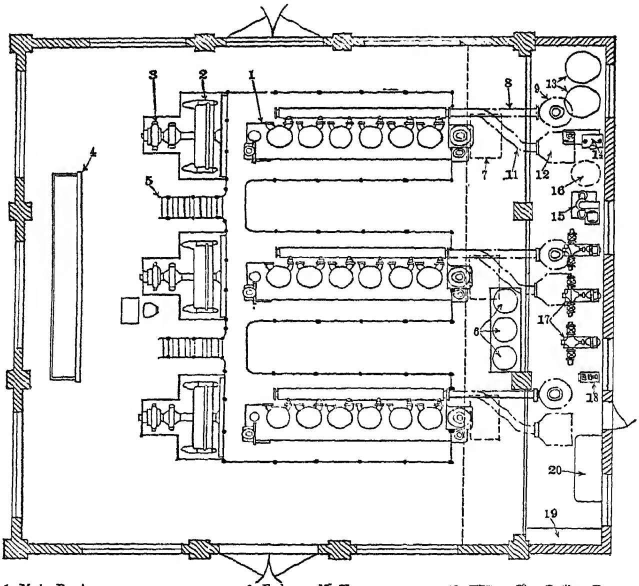


FIG. 19. Elevation of Typical Diesel Engine Plant



1. Main Engine
2. Generator
3. Exciter
4. Switchboard
5. Interconnecting Platform
6. Daily Use Fuel Tanks
7. Lubr. Oil Sump Tanks
8. Exhaust Line
9. Exhaust Muffler
10. Exhaust Stack
11. Air Intake
12. Air Filter
13. Air Starting Tanks
14. Air Starting Outfit
15. Centrifuge
16. Centrifuge Sump Tank
17. Water Circulating Pumps
18. Fuel Oil Transfer Pump
19. Lockers
20. Work Bench
21. Pipe Trenches
22. Piping
23. Overhead Crane

G. 20. Plan of Typical Diesel Engine Plant

fly-wheel effect for the purpose intended. Some types of engines require but little fly-wheel effect, which may be incorporated in the generator rotor. Generators with weighted rotors are specified as fly-wheel type, or weighted-rotor type, as distinguished from standard engine type.

ENGINE EXTENSION SHAFTS.—The length and diameter of extension shafts of Diesel engines vary with type and speed of engine, type of generator or other driven equipment, and critical speed or torsional vibration conditions. Fly-wheel type generators usually are mounted on short extension shafts; standard engine type generators require longer extension shafts. Exact dimensions cannot be determined until the driven equipment is selected and its engineering data analyzed.

FLY-WHEEL-BARRING DEVICES are required for rotating the engine to its starting position or for adjustment and repair. The three types of barring devices used are manually, pneumatically or electrically operated. The first two types usually are supplied by engine builders. Slow-speed engines up to about 750 Hp. generally have manually-operated fly-wheel barring devices.

PYROMETERS, if used, may be attached to and made a part of the engine, or may be mounted on the switchboard or engine-room wall.

OIL COOLERS for lubricating oil are standard equipment, where climatic conditions require them. Sizes and types are selected by the engine builder.

BEDPLATES.—When engine-generator sets are mounted on concrete foundations, general practice is not to use an extension of the engine base or separate sub-base to support the generator, exciter and outboard bearing.

RIGHT- AND LEFT-HAND ENGINES.—An engine is right-hand if the controls are on the right-hand side when the engine is viewed from the fly-wheel end. There is little, if any, advantage in a stationary plant with two engines, in having one engine right-hand and the other left-hand.

11. ECONOMICS OF DIESEL POWER

Stationary Plants

The most common application of Diesel engines in stationary plants is the driving of electric generators. The annual reports of the Oil Engine Power Cost Sub-committee of the A.S.M.E. contain many data on the operating costs of over 100 Diesel-driven electric generating plants of all ages and types located in various parts of the U. S. In a study of operating costs of a well-run plant with modern engines, L. H. Morrison used these data and plotted separately fuel consumption, lubricating oil consumption, engine and other repairs, and supplies. These were plotted against plant kilowatt-hour output per kilowatt capacity, which is a measure of the use of the plant. Fuel and lubricating oil costs were adjusted to prices of 4 cts. and 50 cts. per gal., respectively. Repairs include additional labor used on repairs. Median lines then were established with as many plants above as below the lines. Since many of the plants plotted were not of modern design, and some had inferior attendance, Morrison took for his criterion for modern, well-run plants the average costs of those above the median lines. Fig. 21 (*Diesel Power*, Nov., 1932) shows costs with fuel at 4 cts. per gal., and also at 5 cts. per gal.

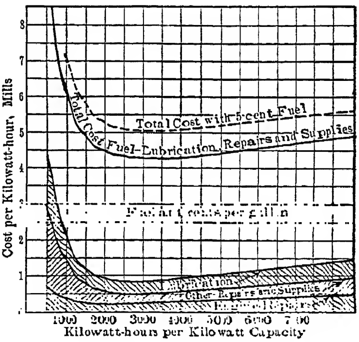


Fig. 21. Operating Cost Excepting Labor of Diesel Plants

ATTENDANCE.—Fig. 21 does not include labor, *i.e.*, attendance cost. The A.S.M.E. report shows that the better plants employ, as a rule, only one man per shift. Plants over 1500 kw. may need a second man on one shift, especially in lighting plants where the evening load varies rapidly. Wages vary widely and plant executives must establish this item for themselves.

OPERATING COST.—On the basis of one-shift operation, engineer's wages of $2250 yearly, and 4-cent fuel oil, Morrison gives total operating costs, including labor, as shown in Table 4. In the smaller Diesel plants, especially factory power plants, arrangements often can be made for part-time attendance, thus reducing the cost to less than that shown in Table 4.

Table 4.—Operating Costs of One-shift Diesel Plants

Based on fuel oil at 4 cents per gal., and attendant's wage of $2250 per year.

Plant Capacity, kw.	Yearly Output, kw.-hr. per kw. Capacity			
	1000	2000	3000	4000
	Operating Cost, Cents per kw.-hr.			
250	1.518	0.899	0.730	0.654
500	1.068	.674	.580	.542
1000	0.842	.561	.505	.486
2000	.730	.505	.467	.458

FIXED CHARGES per kilowatt-hour generated in a Diesel power plant depend on many factors, among which are: *a*. Amount of investment, which depends on size and type of engines and completeness of accessory equipment; *b*. Probable useful life of plant; *c*. Annual output of plant; *d*. The rate of interest.

Diesel engines cost (1935) $25 to $45 per rated horsepower. To this must be added cost of electric generators, switchboards, wiring, foundations, fuel oil storage tank, piping, measuring instruments, etc., which vary widely. A complete Diesel-electric plant, installed, may cost from $60 to $250 per kilowatt capacity. The useful life of most Diesel engines in stationary service is determined not by the actual wearing out of the engine, but by changes in need for the engine. The wearing parts of a Diesel engine plant comprise only a small proportion of the total cost; the life of an engine kept in repair is indefinite. The management of a Diesel plant must decide its probable useful service life. Funding periods are taken as high as 20 years, and as low as 10 years. From 12 to 15 years are usual periods.

STATIONARY DIESEL ENGINES, NON-ELECTRICAL DRIVE.—The foregoing figures are based on electric drive and the costs are on a kilowatt-hour basis. Diesel engines also are used for mechanical drive, for example, to drive ammonia compressors and oil pipe-line pumps. To obtain mechanical cost from electrical cost, an approximately correct factor is $^2/_3$, *i.e.*, cost per B. Hp.-hr. at the engine shaft equals $^2/_3 \times$ cost per kw-hr. generated. This relation is exactly true for an electrical efficiency of 89.4%.

Typical Operating Records of Stationary Diesel Engine Plants

MUNICIPAL ELECTRIC PLANT.—Table 5 shows costs of 1933 for the municipal Diesel-electric plant of Hudson, Mass. (L. D. Wood, *Diesel Power*, Mar., 1934). The plant consists of two 615-kw., one 460-kw. and one 835-kw. units. Current generated was 5,135,900 kw.-hr., gross, 4,543,900 kw.-hr., net. Overhead charges include both depreciation and bond retirement, as required by state regulations. If the smaller item, depreciation, be disregarded, the total cost per kilowatt-hour distributed becomes 9.93 mills.

Table 5.—Operating Costs of Hudson, Mass. Municipal Diesel Engine Plant

	Cost for Year 1933		Cost per kw.-hr. Generated, Mills	Cost per kw.-hr. Distributed, Mills
Production Expenses				
Superintendence and Labor	$14,846.61		2.87	3.27
Fuel	14,929.91		2.9	3.29
Water	269.18		0.05	0.06
Lubricants	1,844.17		0.36	0.41
Maintenance				
Station Structures	1,709.38		0.33	0.37
Diesel Plant Equipment	523.89		0.11	0.12
Diesel Engines	1,132.49		0.22	0.25
Accessory Electric Equipment	45.27		.0	.0
A. Total Production Expenses		$35,300.90	6.84	7.77
Distribution Expenses		11,953.11	2.3	2.63
Utilization Expenses		1,542.60	0.3	0.34
New Business Expenses		1,511.59	0.29	0.33
General Expenses		31,250.28	6.1	6.88
B. Total Operating Expenses		$81,558.48	15.83	17.95
Overhead Costs Chargeable to Production				
Depreciation	$5,000.00		0.97	1.10
Bond Interest	1,800.00		0.35	0.4
Bond Retirement	7,500.00		1.46	1.65
Insurance and Additional Labor	500.00		0.09	0.11
C. Total Overhead		$14,800.00	2.87	3.26
Total chargeable to production (A+C)		$50,100.90	9.71	11.03

FACTORY ELECTRIC PLANT.—Many factories have good load factors, better than the ordinary municipal electric plant, and Diesel engines have been found economical even in plants that are quite small. Table 6 gives the performance and costs of a 40-kw. Diesel unit supplying power to a factory, with auxiliary electric service furnished by a public utility.

Table 6.—Operating Record and Costs for 4 Months' Operation of 40-kw. Diesel-Electric Plant with Public Utility Auxiliary Service

Diesel Unit Data. Current generated, 40,327 kw.-hr.; hours run, 1532 1/4; average running load, 26.3 kw.; average running load factor, 65.7%; fuel oil consumed, 4734 gal.; lubricating oil consumed, 84 3/4 gal.

Diesel Operating Cost	Total	Per kw.-hr.	Public Utility Service
Fuel oil (6.55 c. per gal.)	$309.50	$0.0077	Current purchased, 8879 kw.-hr.
Lubricating oil (46 c. per gal.)	39.00	.0010	Max. demand (30 min.) 27 kw.
Maintenance reserve	60.50	.0015	Price per kw.-hr. 0.0478
Attendance and supervision	176.00	.0043	Total cost $424.04
Total!	$585.00	.0145	
Credit for coal saved by using Diesel heat	100.00		
Net cost	$485.00	$0.0120	

Totals, Diesel Plus Public Utility

Current generated and purchased	49,206 kw. hr.
Maximum demand	67 kw.
Load factor	24.4%
Total cost, Diesel plus public utility	$909.04
Total cost, Diesel plus public utility, per kw.-hr.	$0.0185
Comparative cost, if all purchased	$1980.00
Saving effected by Diesel plant	$1071.00
Annual saving, assuming Diesel heat used 6 mo. per year	$2990.00

OFFICE BUILDING ELECTRIC PLANT.—One of the first Diesel-electric plants for supplying the entire requirements of an office building was installed in 1932 in the 18-story building, No. 1 Park Ave., New York. The plant is fully automatic, and consists of four 180-kw. Diesel direct-current generating units operating in conjunction with a storage battery. The latter starts the engines, and provides good voltage regulation despite an irregular load. The jacket water, discharged at about 140° F., is pumped to the building's hot water service tank. Operating costs (*Diesel Power*, Mar., 1935) during 1934 were:

Total power generated	1,500,000 kw.-hr.
Fuel oil, at 6.4 cents per gallon	$9,000
Lubricating oil, at 50 cents per gallon	750
Maintenance parts, labor, supervision	5,550
	$15,300
Less credit for hot water	900
Total generating cost	$14,400
Generating cost per kw.-hr.	0.96 cents

ICE PLANTS.—Fuel and lubricating oil consumption and costs of eight Diesel-driven ice plants, in different parts of the U. S., are given in *Diesel Power*, Mar., 1935. Plant capacity ranged from 12 to 98 tons of ice per day. Ammonia discharge pressure ranged from 120 to 190 lb. per sq. in. Average costs, covering a week or more during the summer were: Fuel oil, per ton of ice, 4.79 gal.; cost per gal., delivered, 5.41 c.; cost per ton of ice, 25.9 c.; lubricating oil, per ton of ice, 0.0533 gal.; cost per gal. delivered, 50 c.; cost per ton of ice, 2.66 c. Combined cost of fuel and lubricating oil per ton of ice produced, 28.56 c. Although the fuel oil per ton of ice varied in the eight plants from 3.62 to 6.25 gal., the total cost of fuel and lubricating oil per ton of ice produced varied only from 23.9 to 34.2 cents.

Performance Data, Stationary Plants

A.S.M.E. REPORT ON OIL-ENGINE POWER COST.—The outstanding tabulation of records of performance and operating costs of Diesel engines in stationary electric generating plants is the annual report on oil-engine power cost, compiled from original sources by the A.S.M.E. Subcommittee on Oil-Engine Power Cost.

The 1933 report includes information from 156 oil-engine generating plants, containing 398 engines, totaling 216,010.5 rated B. Hp., with a total net output of 259,209,519 kw.-hr. The information on production cost includes type of plant, load, size, hours operated, load factors, costs of fuel

lubricating oil, attendance, supplies and repairs. The data on engine details and operating information comprises the following items for each engine in each plant:

Engine Data
- Engine desig
- Engine cycle
- Injection system
- Scavenging system
- Trunk piston or crosshead
- Rated B.Hp.
- Equivalent kw. (90% generating efficiency)
- Number of cylinders
- Cylinder dimensions, bore × stroke, in.
- Rated r.p.m.
- Generator rating, kv.-a
- Year in which engine started work
- Engine-hours operated in reported period

Lubrication
- Total new lubricating oil used, gal.
- New lubricating oil for cylinder lubrication only, gal.
- Unfit lubricating oil discarded, gal.
- Rated Hp.-hr. per gal. of new lubricating oil
- Lubricating oil treatment

Fuel
- Fuel oil used, gal.
- Nature of fuel oil used
- Is fuel centrifuged?

Output
- Gross output, kw.-hr.
- Gross kw.-hr. per gal. of new lubricating oil
- Gross kw.-hr. per gal. of fuel oil

Loading
- Running engine capacity factor
- Running plant capacity factor
- Peak load during reported period, gross kw.
- B.M.E.P. at rated B.Hp., lb. per sq. in.
- B.M.E.P. at peak load (90% generating efficiency)

Miscellaneous
- Piston cooling
- Air filters
- Type of cooling system
- Average temperature of incoming cooling water, deg. F.
- Average temperature of outgoing cooling water, deg. F.
- Purpose for which jacket water heat is utilized
- Purpose for which exhaust heat is utilized
- Plant altitude, ft. above sea level

Maintenance and Repairs
- Cost of engine regular upkeep, dollars
  - Material
  - Extra labor
- Cost of repairs for engine accidents, dollars
  - Material
  - Extra labor
- Total engine maintenance in dollars per rated B.Hp. per yr.
- Major engine parts renewed during reported period
- Number of enforced engine shut-downs
- Total duration of enforced engine shut-downs, hr.
- Total engine maintenance time not included in enforced shut-down, time, hr.

Attendance
- Number of shifts in period
- Number of hours per shift
- Number of attendants per shift
- Output per man-hour, net kw.-hr.

Plant Altitude
- Feet above sea level

In addition to the tabulated data, the following charts on which the plants are separately plotted, are shown: 1. Lubricating oil economy *vs.* plant running capacity factor. 2. Fuel economy *vs.* plant running capacity factor. 3. Total production cost *vs.* plant yearly output. Two charts, in simplified form, are reproduced in Figs. 22 and 23.

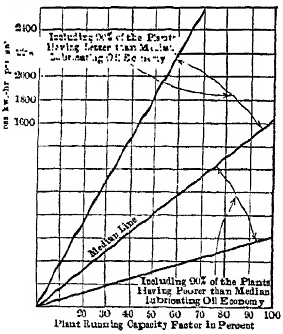


FIG. 22. Lubricating Oil Economies of 120 Full Diesel Plants

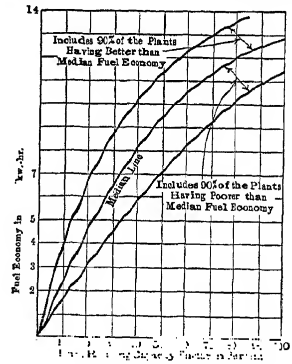


FIG. 23. Fuel Economies of 122 Full Diesel Plants

Economics of Diesel Power, Marine Plants

The wide diversity in conditions of marine service make detailed cost figures of limited value. The economics of Diesel application are better indicated by the extent of use. L. R. Ford (at the A.S.M.E. annual meeting, Dec., 1934) gave the status of Diesel engine propulsion in various classes of marine service as follows:

1. River and Harbor Craft. Includes tugs, work boats, ferries and dredges. Progress

in Diesel propulsion of tugs is such that in some localities the steam tug probably will disappear. Work boats and ferries are being powered increasingly with Diesel engines. The number of Diesel-operated dredges is growing.

2. Lake, Bay and Sound Craft. Includes ferries, passenger vessels, freight boats, fishing vessels, tugs, and self-propelled barges. Diesel propulsion is progressing in this class, with the exception of the larger passenger vessels in the East and the large cargo and passenger vessels on the Great Lakes. In 1934, about 125,000 Hp. of Diesel engines were installed in vessels of this class in the U. S.

3. Ocean Going Vessels. Includes coastwise and intercoastal types of passenger ships, combination passenger and cargo ships, cargo ships and tankers. The world's motorship tonnage increased from 1,540,463 tons in 1922 to 10,604,526 in 1934. Steamship tonnage decreased 10.1% in the same period. Of 1,217,366 tons of ships under construction throughout the world on Sept. 30, 1934, 729,131 tons was motorship tonnage. Of 324,712 tons of tankers under construction, 87.8% were being fitted with Diesel engines.

POWER TRANSMISSION FROM DIESEL ENGINE TO PROPELLER.—In many vessels, desirable propeller speed is much lower than desirable engine speed. Direct-drive, which is simplest, demands a compromise, the Diesel engine being of slower speed and larger dimensions than otherwise would be necessary. Alternatives are: 1. Diesel-electric drive; one or more engines drive generators to supply electric energy to a propulsion motor on each propeller shaft. 2. Diesel-gear drive: engine power is transmitted through reduction gears, with or without a clutch, to the propeller shaft or shafts.

L. B. Jackson gives advantages of Diesel-electric drive as: 1. Control is easy and highly refined. Pilot has complete control. 2. More reliable. With several generating units, one or more can be stopped without stopping propulsion. 3. Flexible as to engine room layout. 4. Overload torque can be developed at low propeller speeds without overloading engines. 5. Power can be measured accurately and easily. 6. Electric energy for auxiliaries can be obtained from main engines. 7. Low maintenance cost, because engines run at constant speed, and electric drive absorbs all shocks. 8. Engine sizes can be standardized.

Advantages of the Diesel-gear drive are: 1. Lower cost than electric drive. 2. Less complication. 3. Less space. 4. Smaller losses. Gear drives, however, may be noisy and suffer from vibration unless well designed and built with extreme precision.

Diesel engines in motorboats and aircraft eliminate the fire hazard that exists with gasoline fuel.

Economics of Automotive Diesel Engines

The first applications of Diesel engines to automotive services were in railroad switching locomotives (in 1925 in the U. S.), and in contractors' machinery such as shovels and drag-lines. With the development of engines of lighter weight, there has been since 1931 wide application of Diesel engines to trucks, tractors and buses, railcars and high-speed streamlined railroad trains.

The principal advantage of the Diesel engine over the gasoline engine in these services is lower fuel cost, due to its ability to use a cheaper fuel and also its smaller consumption of fuel. The reduced fuel consumption results from: *a.* Inherently better fuel efficiency at full load, due to higher gas pressures; *b.* Diesel part-load efficiency more closely approaches full-load efficiency than does that of the gasoline engine. *c.* Torque characteristics are more favorable. Fig. 24 (*Auto. Ind.*, Aug. 5, 1935) shows variation of fuel consumption with load factor of a Diesel engine and a gasoline engine built by the same manufacturer, and of approximately equal power. Both engines developed the same maximum torque, 390 lb.-ft., at rated speed, 850 r.p.m. When slowed by increasing load, maximum Diesel engine torque was 440 lb.-ft. and maximum gasoline engine torque was 413 lb.-ft., both at 600 r.p.m.

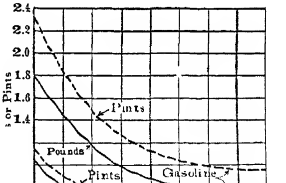


FIG. 24. Comparison of Specific Fuel Consumption of Diesel and Gasoline Tractor Engines

DIESEL ENGINES IN TRUCKS.—The data in Table 7 for Diesel and gasoline engines, obtained from truck owners, were reported by Ben J. Selbst in *Diesel Power*, May, 1934. The vehicles in each comparison operated under identical conditions.

DIESEL LOCOMOTIVES.—The Diesel engine develops little power while starting, and direct drive is impractical. Mechanical transmission, *i.e.*, gearing, generally is used on Diesel locomotives of the sizes used for factory switching and general hauling. A manufacturer gives the following operating record of two 13-ton locomotives, equipped

Table 7.—Comparative Operating Records of Diesel Engine and Gasoline Engine Trucks

	Tank Wagon Milk Truck		Truck and Trailer	
	Gasoline	Diesel	Gasoline	Diesel
Total trip mileage	220	220	1015	1015
Payload, lb.	39,322	39,322	37,000	37,000
Payload, tons	19.661	19.661	18.5	18.5
Total ton-miles hauled	4325	4325	18,800	18,800
Fuel consumption, gal.	75	35	360	175
Fuel consumption, miles per gal.	2.93	6.3	2.82	5.8
Ton-miles per gal.	57.6	123.5	52.3	107.2
Fuel cost, cents per gal.	11.625	4.75	11.5	4.0
Total fuel cost	$8.72	$1.66	$41.40	$7.00
Cost of fuel per 100 ton-miles	$0.202	$0.0344	$0.22	$0.037
Ratio of fuel cost	5.9	1	6.0	1
Ratio of fuel price	2.45	1	2.9	1

with 90-Hp. Diesel engines, used in tropical plantation service. The record covers a three-month period during which the locomotive traveled 8206 miles in 1409 hours.

	Per mile	Per service hour
Fuel consumed, gal.	0.1960	1.191
Lubricating oil consumed, gal.	0.0043	0.053
Cost figures:		
Operator	$0.0303	$0.176
Lubricating oil	.0116	.067
Service cost	.0075	.044
Repair cost	.0036	.021
Fuel cost	.0213	.124
Total operating cost	$0.0743	$0.432

Diesel-Electric Locomotives.—Up to 1935 the larger Diesel locomotives, for switching and light passenger service, almost universally have used electric transmission, *i.e.*, a Diesel engine-driven electric generator supplies energy to traction motors driving the wheels. This system involves considerable equipment and investment cost but gives ideally simple control.

Operating data of Diesel and steam locomotives gathered by Am. Elect. Ry. Assoc. and Am. Ry. Assoc., and tabulated in their 1932 committee reports are summarized by A. H. Candee (*Diesel Power*, May, 1934) as follows: 1. Diesel fuel costs per 100 ton-hr. approximate 25% of steam fuel cost, with oil at 5c. per gal. and coal at $3.00 per ton. 2. One gallon of fuel oil in switching service is equivalent to 140 lb. of coal. 3. Diesel engine lubrication approximates 8.76c. per 100 ton-hr. of locomotive switching service. 4. "Other lubrication" cost is 35% of steam locomotive lubricating cost. 5. Total cost of fuel, engine lubrication, locomotive lubrication and "other expenses" of Diesel locomotives is 33% of corresponding steam locomotive costs. 6. Diesel locomotive repair costs for industrial switching service are lower than are indicated for trunk line railroads, and lower than steam locomotive repair costs.

Diesel Streamlined Trains.—Many railroads in the U. S. are (1935) experimenting with complete light-weight streamlined trains of articulated cars, and equipped with one or more light-weight, high-speed Diesel-electric units. These trains are capable of exceedingly high speeds and low operating cost.

The Chicago, Burlington and Quincy R.R. operates twin "Zephyrs" 440 miles daily between Chicago, Minneapolis and St. Paul. These are three-car trains, designed for 110 miles per hour. The motive power is a 600-Hp., 750-r.p.m., 8-cylinder, 8 × 10-in. uniflow 2-cycle, solid injection Diesel engine, scavenged by a rotary blower. The operating expenses per mile of the "Zephyrs" during May, 1935, and also of a corresponding 5-car steam train are given as follows:

	Zephyr	5-car Steam Train
Fuel	$0.0141	$0.1190
Water		.0068
Lubricants	.0108	.0034
Wages of crew	.1709	.2360
Train supplies and expenses	.0311	.0468
Terminal expenses	.0351	.0375
Maintenance of power plant	.0188	
Maintenance of train	.0356	
Locomotive repairs		.1466
Passenger car repairs		.0800
Locomotive supplies		.0019
Engine house expenses		.0263
Lounge car attendant		.0077
Total	$0.3164	

11. TESTS OF DIESEL ENGINES

THE A.S.M.E. TEST CODE for Internal Combustion Engines is applicable to Diesel engine tests. See p. 16–44.

The Society of Automotive Engineers has developed Diesel engine Testing Forms, which primarily are adapted to the testing of automotive-type engines. These are abstracted below. Complete codes may be obtained from the S.A.E.

S.A.E. CODE.—A complete engine test includes the determination, at different speeds, of: Maximum horsepower; fuel economy at rated load, and at 125%, 75%, 50% and 25% of rated load at each speed; friction horsepower. These data are plotted against r.p.m. to give the following curves: Maximum brake horsepower; maximum torque; maximum brake M.E.P.; friction horsepower; mechanical efficiency; fuel per B.Hp.-hr. at rated loads, and at 125%, 75%, 50% and 25% of rated load; brake thermal efficiency at rated load, and at 125%, 75%, 50% and 25% of rated load.

No readjustment may be made during a test, except as required for actual changes of speed and load. The engine should be run-in sufficiently to show no appreciable change in friction before and after a 30 min. run at normal rated speed and load. Before a test run, the engine must be brought to a condition of sustained operation under the conditions of the run, and all loads, temperatures, etc. must remain substantially constant during the run.

Number of Runs.—For horsepower and fuel economy tests runs should be made at intervals - 20% of rated r.p.m., and also at the lowest steady operating speed.

Duration of Runs.—Prior to any test, engine should run at least 1 hr. at maximum load at desired r.p.m. The test runs at 125%, 100%, 75%, 50% and 25% of rated load, should be made in the sequence given, and should continue at least 30 min. before readings are taken. Duration of fuel consumption tests should be at least 5 min.; a longer period is desirable. Friction-horsepower tests should be at least 1 min. duration. Maximum speed variation during a run, ±10% of stated r.p.m. of test run.

Dynamometer and Brake Loads.—Dynamometer should be carefully balanced before brake-load readings are recorded. Brake-loads should be measured with accurately calibrated platform or beam scales. Connection of dynamometer arm to scales should be by knife-edges, calibrated spring-balance and tripod, or suitable linkage. Accurate counterbalance or tare loads must be provided. The platform or beam scales are kept balanced during a run, and load registered must be substantially constant.

Revolutions per Minute should be taken by positively-driven counters, which engage and disengage at beginning and end of runs. Difference between readings divided by duration of run in minutes is average r.p.m. Tachometers may be used only as an approximate check on average speed and to indicate speed variations.

Fuel Consumption should be determined by noting decrease in weight of tank from which fuel is fed. The tank, mounted on sensitive platform scales, is connected to fuel supply pipe by a tube fed by siphoning or pump suction. Difference in weight of tank at beginning and end of run, divided by duration of run in minutes, permits calculation of the consumption per hour. By repeatedly setting the scale counterpoise back a definite small amount and noting the times at which it falls, an indication of steadiness of fuel consumption may be obtained.

Temperatures are given in deg. F. Readings should be made of temperatures of entering air, entering water, outgoing water, fuel; and also, in friction horsepower tests, of mean temperature of jacket water. With pump-circulated jacket water, mean temperature may be taken as average of inlet and outlet temperatures. With thermosyphon circulation, thermometers should be inserted in the jacket, and average of readings taken. At least three readings should be taken during each run, one each being at beginning and end of run. Outlet water temperature should be kept within ±5° of that recommended by engine builder.

Friction Horsepower may be determined approximately by using an electric dynamometer, preferably of the cradle type, to drive the engine and measure the torque reaction. As reaction is in opposite direction to that obtaining when engine drives the dynamometer, provision must be made to measure torque on both sides of dynamometer, or direction of pull must be changed by linkage. Friction horsepower test should immediately follow brake horsepower test, before engine has cooled. Throttle remains in same position as for corresponding brake horsepower test. Compression relief cocks should remain closed, and all accessories used during brake horsepower test should remain in operation.

If an electric dynamometer is not available, friction horsepower can be determined approximately by testing at various speeds throughout load range, plotting a fuel consumption-brake horsepower curve, and then repeating tests with 1, 2, 3, etc. cylinders cut out. By replotting the curve, friction horsepower can be calculated by interpolation.

Indicated Horsepower can be determined approximately by adding friction horsepower at a given speed to brake horsepower at same speed. Results can be corrected to standard barometric pressure of 29.92 in. Hg and standard temperature of 60° F. (520° abs.) by formula

$$\text{B.Hp.}_c = \text{B.Hp.}_o \times (P_S / P_O) \times \sqrt{T_O / T_S}$$

where B.Hp._c and B.Hp._o = corrected and observed brake horsepower, respectively; P_S and P_O = standard and observed barometric pressures; T_S and T_O = standard and observed absolute temperatures, deg. F.

Dimensions and Engine Data.—The code includes a specification sheet for reporting dimensions and other details concerning the engine and its accessories, including the fuel injection system.

COMPLETE TEST OF LARGE MARINE DIESEL ENGINES.—Unusually complete tests of the twin-screw motor-ship *Polyphemus* are given in *Proc.* Inst. M.E. vol. 121, p. 183, 1931. The ship was driven by two 6-cylinder, 4-stroke cycle, air-injection engines rated at 2750 B.Hp. at 138 r.p.m., supercharged on the Buchi system (see p. 12–10). Particulars of the engines follow:

Bore, 24.41 in.; stroke, 51.18 in.; clearance volume, 2115 cu. in; compression ratio, 12.32; inlet valve opens 70° before top center, closes 35° after bottom center; exhaust valve opens 42.5° before bottom center, closes 60° after top center; fuel valve opens 11° before top center; fuel pump capacity per cylinder, 1.125 cu. in.; fuel pump pressure, 1000 lb. per sq. in.; starting air valve opens at top center, closes 135° after top center; starting air pressure, 450 lb. per sq. in., maximum. The total running weight of engine room machinery was 1098.4 tons (2240 lb.) divided as follows: Main engines, without flywheels, but including all pipes, fittings, thrust block and shafts, 492.2 tons; flywheels, 5.5 tons; shafting, stern gear and propellers, 101.2 tons; auxiliaries, pipes and fittings, 197.7 tons; boiler, steam and exhaust pipes, 12.5 tons; air reservoirs, 33.2 tons; miscellaneous fittings, tools, spare parts, etc., 151.9 tons; water in system and lubricating oil, 104.2 tons.

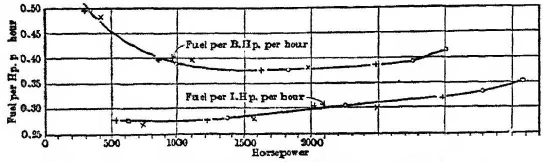


Fig. 25. Fuel Consumption of Motorship *Polyphemus*

A typical analysis of the fuel oil used is: C, 86.24; H, 13.24; S, 0.34; N, nil; ash, 0.0022; carbon residue and free carbon, 0.03; soft asphaltum, 0.08; water, trace; calorific value, 19,500 B.t.u. per lb.; sp. gr., 0.863; flash point, closed, 177° F., open, 162° F.; viscosity (Redwood No. 1) at 70° F., 39 sec.; at 150° F., 30 sec.; at 200° F., 27.8 sec.

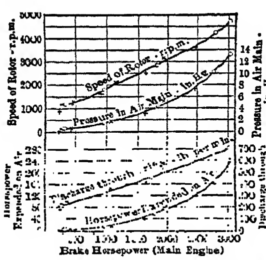


Fig. 26. Performance of Scavenger Air Blower

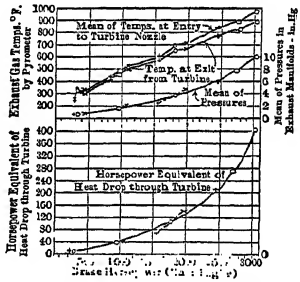


Fig. 27. Temperature, Pressure and Equivalent Horsepower of Exhaust Gases Passing through Exhaust Gas Turbine

Table 8 gives analyses of the exhaust gases at different loads. Hydrogen in the gases was 0.04%, and CO ranged from 0.02 to 0.07%. Engine performance is given in Table 9.

Table 8.—Analysis of Exhaust Gases of Motorship *Polyphemus*

Mean indicated pressure, lb. per sq. in.	131	132	102	71	49
Oxygen, percent	13.0	12.9	13.9	16.0	16.6
Carbon dioxide, percent	5.7	5.3	4.5	2.3	2.2

Table 9.—Performance of Engines of Motorship *Polyphemus*

	Test No.				
	12	13			
Brake horsepower	3020	2760	1810		340
Brake M.E.P., lb. per sq. in.	117	110	82.4	55.1	27.6
Average speed, r.p.m.	142.3	138.4	121.2	98.0	68.6
Friction horsepower	414	368	295	275	192
Friction M.E.P., lb. per sq. in.	16.0	14.7	13.4	15.5	15.5
Mechanical efficiency (B.Hp./I.Hp.), percent	84.2	83.9	80.8	71.5	55.8
Fuel per hr., lb.	1264	1094	683	384	169
Fuel per B.Hp. per hr., lb.	0.419	0.397	0.377	0.39[illegible]	0.497
Fuel per I.Hp. per hr., lb.	0.35	0.33	0.31	0.28	0.28
Thermal efficiency, B.Hp. basis	0.312	0.328	0.346	0.32[illegible]	0.262
Thermal efficiency, I.Hp. basis	0.37	0.39	0.43	0.47	0.47

Table 10.—Heat Balance at Various Loads of Engine of Motorship *Polyphemus*

All quantities are in 1000 B.t.u. per min.

	Test No.				
	12	13			
Heat in fuel	411.0	357.0	222.0	125.0	55.0
Distribution of heat					
Thermal equivalent of B.Hp.	128.0	117.0	76.8	41.5	14.4
Air-compressor Hp.	6.62	6.87	5.73	4.88	3.31
Rejected through jackets and heads.	56.10	46.20	32.50	20.90	12.94
Rejected through pistons	13.60	11.40	8.40	4.83	2.46
Heat drop through turbine	17.11	11.61	4.54	1.67	0.30
Rejected in exhaust gases	157.85	137.53	74.90	36.45	15.63
Radiation, etc.	31.72	25.39	19.13	14.77	5.96
Total Heat	411.00	357.00	222.00	125.00	55.00

Fuel consumption is plotted in Fig. 25, which illustrates the Diesel engine characteristic that while fuel consumption per B.Hp.-hr. is at a minimum between 3/4 load and full load, fuel consumption per I.Hp.-hr. decreases with decrease of load. Fig. 26 shows performance of the scavenger air blower. Fig. 27 gives data of exhaust gases. Table 10 gives the heat balance at various speeds and loads. A summary of the maneuvering trials is: 1. Number of engine starts possible with an available initial pressure of 455 lb. per sq. in., 65. 2. Average time, *a*, for all cylinders to fire, 11.5 sec.; *b*, to attain prescribed speed, 1.2 sec.; *c*, to stop turning, 7.5 sec. 3. Lowest pressure at which engine started was 120 lb. per sq. in. Engine refused to start with a pressure of 115 lb. per sq. in., on the 66th attempt. 4. Average volume of free air used per start, 249 cu. ft.

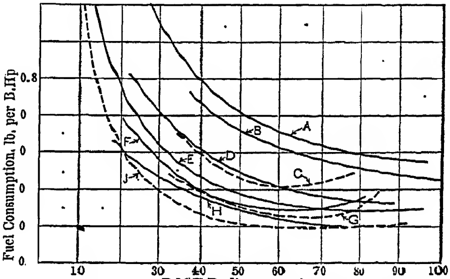


FIG. 28. Fuel Consumption of Truck and Bus Engines. *A*—6-cyl., 4.33 × 6 in.; *B*—4-cyl., 5.90 × 7.09 in.; *C*—4-cyl., 4.33 × 7.09 in.; *D*—4-cyl., 5.31 × 7.87 in.; *E*—2-cyl., 3.15 × 11.00 in.; *F*—6-cyl., 4.33 × 5.59 in.; *G*—4-cyl., 4.72 × 7.09 in.; *H*—2-cyl., 7.5 × 12.0 in.; *J*—4-cyl., 4.5 × 5.5 in.

TEST OF HIGH-SPEED OIL ENGINE.—Table 11 gives results of a test of a Paxman-Ricardo 4-cylinder heavy oil engine rated at 40 Hp. at 1000 r.p.m. and 60 Hp. at 1500 r.p.m., made by W. A. Tookey (*Engg.*, April 5, 1935). The cylinders were 4 5/8 × 5 7/8 in. (394 cu. in. total displacement). Rating is based on a B.M.E.P. of 80 lb. per sq. in. At 1500 r.p.m. the engine developed 125% of its rated power with clear exhaust and 135.8% of its rated power with black exhaust.

FUEL CONSUMPTION OF HIGH-SPEED DIESEL ENGINES.—S. J. Davies (*Jour.* Royal Aeronautical Soc., April, 1931) gives the curves of Fig. 28 showing fuel consumption per B.Hp.-hr. on a base of B.M.E.P. of different types of truck and bus engines, all about 1200 r.p.m. and developing about 50 B.Hp. Maximum values given of B.M.E.P. are the highest consistent with a clear exhaust.

DATA ON HIGH-SPEED DIESEL ENGINES.—Tables 12 to 14 (E. F. Ruehl, *Jour.* S.A.E., 1931) compare weight, speed and power of typical commercial Diesel and

Table 11.—Test of Paxman-Ricardo 4-Cylinder Heavy Oil Engine

	1000 r.p.m.			1500 r.p.m.
Load rating, percent	100	125*		100
Brake horsepower	40	50	58	60
Brake M.E.P., lb. per sq. in.	80	100	115.	80
Indicated M.E.P., lb. per sq. in.	104			108
Mechanical efficiency, percent	77			
Indicated horsepower	52			
Fuel consumption				
Lb. per B.Hp.-hr.	0.40	0.417	0.462	0.397
Lb. per I.Hp.-hr.	0.308			0.295
Heat value of fuel, B.t.u.	19,728			
Indicated thermal efficiency, percent	41.8			
Exhaust manifold temp., deg. F.		806	1112	

* Maximum load with clear exhaust. † Maximum load with smoky exhaust.

Table 12.—Dimensions, Output and Fuel Economy of Heavy-duty, High-speed Internal Combustion Engines of 5-in. and up to 7-in. Bore

	Diesel Engines										Gasoline Engines			
	Single Combustion Chambers, Individual Fuel Pumps						Antechambers							
	Treiber	Krupp	Treiber	Buda	Buda	Cummins	Hill	Koerting	Oberhaensli	Hill	Sterling	Buffalo	Waukesha	Waukesha
No. of cylinders	6	6	6	6	6	6	6	6	4	6	6	6	4	4
Bore, in.	5	5.3	6	6	6 1/2	6 3/4	5	5.1	5.12	6	5 3/4	5 7/8	5 3/4	6 3/4
Stroke, in.	7	7.85	8	8	8 3/4	9	7	7.1	7.48	10	6 3/4	7	8	8
Rev. per min.	1200	1000	1000	1000	1000	800	1000	1200	1250	800	1000	1500	950	950
Brake horsepower	100	100	150	138	180	170	75	90	60	120	150	160	80	120
Total cyl. displacement, cu. in.	825	1040	1360	1360	1742	1930	825	880	615	1696	1050	1138	830	1145
Engine weight, lb.	2200	1760	....	4550	5425	5000	3920	3300	....	7500	2750	3400	5025	5125
M.E.P., lb. per sq. in.	80.0	74.5	87.5	80.6	81.85	87.5	72	67.5	71.3	70	113	70	80.5	87
Piston speed, ft. per min.	1400	1305	1333	1333	1458	1200	1166	1420	1560	1333	1125	1750	1270	1270
Weight: Displacement, lb. per cu. in.	2.67	1.69	....	3.33	3.12	2.59	4.75	3.75	....	4.43	2.62	2.98	6.05	4.47
Specific weight, lb. per B.Hp.	22.0	17.6	....	33.0	30.3	29.4	52.5	36.7	....	58.6	18.3	19.4	62.8	42.6
Specific output, B.Hp. per cu. ft.	210	166	191	174	179	152	157	177	194	122	247	271	166	181
B.Hp. per cylinder	16.7	16.7	25	23	30	28.3	12.5	15	17.2	20	25	26.7	20	30
Fuel at rated M.E.P., lb. per B.Hp.-hr.	0.43	0.45	0.43	0.435	0.435	....	0.46	....	0.46	0.46	....	....	....	....

Table 13.—Dimensions, Output and Fuel Economy of Heavy-duty, High-speed Internal Combustion Engines of 7-in. to 10-in. Bore

	Diesel Engines												Gasoline Engines		
	Single Combustion Chambers, Individual Fuel Pumps							Single Combustion Chambers, Common Rail							
	Ricardo	Foos	De La Vergne	Westinghouse	McIntosh-Seymour	Cummins	Ingersoll-Rand	Atlas-Imperial	Bessemer	Winton	Speedway	Superior	Sterling	Winton	Brill
No. of cylinders	6	6	6	6	6	6	6	6	6	6	6	6	6	6	6
Bore, in.	7 1/2	8 1/2	9	9	9 1/2	10	10	7	7 1/2	8	8 1/2	8 1/2	8	8	8 3/4
Stroke, in.	12	11	11	12	10 1/2	12	12	8 1/2	10 1/2	10	11	11	9	10	10 1/2
Rev. per min.	900	700	750	900	800	800	550	650	650	750	700	600	900	900	950
Brake horsepower	300	230	300	400	360	500	300	120	200	250	300	210	300	300	400
Total cyl. displacement, cu. in.	3180	3750	4180	4575	4460	5655	5655	1962	2782	3020	3750	3750	2720	3020	3780
Engine weight, lb.	11,000	10,500	10,000	12,900	14,400	14,500	19,000	6600	8200	7400	7600	....	5670	7400	8000
M.E.P., lb. per sq. in.	83	70	75.5	77	80	87.6	76.4	74.5	87.6	87.5	90.5	73.8	97.5	88	88
Piston speed, ft. per min.	1800	1280	1375	1800	1400	1600	1100	921	1138	1250	1285	1100	1350	1500	1660
Weight: Displacement, lb. per cu. in.	3.46	2.80	2.39	2.82	3.23	2.56	3.36	3.36	2.95	2.46	2.05	....	2.08	2.45	2.11
Specific weight, lb. per B.Hp.	36.7	45.5	33.3	32.3	40.0	29	63.3	55	41	29.6	25.3	....	18.9	24.7	20.0
Specific output, B.Hp. per cu. ft.	163	106	124	151	140	153	92	106	124.5	143	138	97	190	173	183
B.Hp. per cylinder	50	38.3	50	66.7	60	83.3	50	20	33.3	41.7	50	35	50	50	66.7
Fuel at rated M.E.P., lb. per B.Hp.-hr.	0.38	....	0.39	0.40	....	....	0.44	0.42	0.43	0.44	0.43	0.45	....	....	....

Table 14.—Dimensions, Output and Fuel Economy of Heavy-duty High-speed Internal Combustion Engines of Less than 5-in. Bore

	Diesel Engines									
	Single Combustion Chambers, Individual Fuel Pumps				Antechambers			Gasoline Engines		
	C	Lin H				D				
No. of cylinders	6	4	6	9	6	6	6	4	6	6
Bore, in.	4 1/2	4 1/2	4.7	4 13/16	4.13	4.53	4.33	3 3/4	4 1/8	4.32
Stroke, in.	6	6 1/2	7.1	6	6 1/2	6.7	5.9	5 1/8	4 3/4	6.5
Rev. per min	1000	1200	1000	1950	1300	1250	1600	1800	1600	1800
Brake horsepower	60	50	68	225	70	80	80	36.5	68	100
Total cyl. displacement, cu. in.	572.5	410	738	982	523	646	520	226	381	570
Engine weight, lb.	2200	1100	1980	510	1600	1650	....	660	890	1280
M.E.P., lb. per sq. in.	83	80	71	92.7	81.6	78.5	74.6	71	88.5	77.5
Piston speed, ft. per min.	1000	1300	1185	1950	1410	1394	1575	1540	1270	1950
Weight: Displacement, lb. per cu. in.	3.84	2.7	2.68	0.52	3.06	2.53		2.92	2.34	2.25
Specific weight, lb. per B.Hp.	36.6	22.0	29.2	2.27	22.9	20.6		18.1	13.1	12.8
Specific output, B.Hp. per cu. ft.	182	212	159	398	230	214	267	279	308	302
B.Hp. per cylinder	10	12.5	11.3	25	11.7	13.3	13.3	9.1	11.3	16.7
Fuel at rated M.E.P., lb. per B.Hp.-hr.		0.42	0.43	0.46	0.46	0.46	0.49			

gasoline engines. The Diesel engines all are of the 4-cycle, solid injection type. In later designs, particularly of the larger engines, specific weight has been greatly reduced.

12. DIESEL ENGINE MAINTENANCE

Inspection and maintenance should cover both engines and auxiliary plant. Table 15 (Diesel Engineering Handbook, 7th ed., L. H. Morrison and T. A. Burdick) is a typical schedule form. Opposite each item in the schedule should be entered maximum interval that may elapse between inspections. These intervals will vary in different plants and must be determined individually. Frequency of maintenance operations is controlled by: 1. Cost. 2. Loss of efficiency due to condition of equipment. 3. Necessity of maintaining equipment in service.

A monthly maintenance log on which are listed operations to be performed is advisable. Table 16 (*Power Plant Engg.*, April 15, 1930) is typical. A specific instruction sheet, showing desired clearances and settings, will enable definite standards, initially determined by the manufacturer, and modified by experience, to be maintained. Table 17 (Diesel Engineering Handbook, 7th ed.) is a typical form.

MAINTENANCE EQUIPMENT.—The principal work involved in maintenance is disassembling and replacing parts. Maintenance processes may be classed as: 1. Cleaning. 2. Adjusting. 3. Reconditioning. 4. Renewal of parts. The nature and amount of equipment required to carry out these processes depends on whether the system of maintenance is that of renewal or of reconditioning. In general, renewal is most economical in small plants and with small engines. With larger equipment, reconditioning usually saves time and money. Certain machine tools and precision gages then are essential, as well as one or more skilled mechanics.

COMPRESSION AND COMBUSTION PRESSURES.—Compression pressure should be maintained within 20 lb. of recommended pressure. Probable causes for variation in compression are leaky valves, leaky piston rings, low scavenging pressure, obstructions in suction lines, and increased piston end clearance.

Combustion pressure, *i.e.*, maximum pressure at full load, should be within 50 lb. of manufacturer's recommendation. It is affected by compression pressure, fuel or spray valve conditions, injection timing, injection air pressure, and fuel oil characteristics. The quality of combustion should be good at all times. Incomplete combustion is a direct fuel loss, and causes wear. A poor exhaust shows smoke and high temperature; a good exhaust is clear and the temperature is low.

ROCKER ARM ROLLER CLEARANCE.—Too much rocker arm roller clearance is better than too little. Inadequate clearance may cause blocking of valves when engine is at operating temperature, resulting in loss of compression, overheating, and cutting of

Table 15.—Inspection and Maintenance Schedule for Diesel Engines

Make................Hp..................Type.................Date..............

Enter in this schedule the maximum allowable operating hours between inspections of the listed parts. At the time of inspection all cleaning and adjusting consistent with good engine maintenance is to be done.

Part	Hours	Part	Hours
Crank-shaft		**Fuel Pumps**	
Main bearings	......	Pistons	......
Outboard bearing	......	Packing	......
Thrust bearing	......	Valves	......
Crossheads, Rods and Auxiliary Shafts		**Fuel Oil System**	
Crank-pins and bearings	......	Filters or strainers	......
Piston-pins and bearings	......	Auxiliary storage tanks	......
Crosshead-pins and bearings	......	Supply lines	......
Crosshead shoes and slides	......	Heaters	......
Compressor piston-pin and bearing	......	**Gears or Chain Drives**	
Compressor crosshead-pin and bearing	......	Governor drive	......
Compressor crosshead shoes and slides	......	Cam-shaft drive	......
Vertical shaft bearings	......	Fuel pump drive	......
Vertical shaft thrust bearings	......	Lubricating pump drive	......
Cam-shaft bearing	......	**Lubricating System**	
Cam-shaft thrust bearing	......	Lubricating oil pump complete	......
Pistons and Cylinders		Lubricating oil supply lines	......
Power	......	Lubricating oil strainers or filters	......
Compressor, high-stage	......	Lubricating oil tanks	......
Compressor, intermediate and low-stage	......	Lubricating oil coolers	......
Scavenging pump	......	Engine crank-case	......
Cylinder Head Valves		Pressure feed lubricator and checks	......
Fuel or spray	......	**Piston Cooling**	
Air inlet	......	Cooling passages in piston	......
Exhaust	......	Packing	......
Starting	......	Bearings	......
Starting air check	......	Ball and hinge joints	......
Scavenging valves, mechanical	......	**Scale and Sediment Deposit**	
Safety or relief	......	Cylinder heads and jackets	......
Fuel try or by-pass	......	Air coolers	......
Governor		Oil Coolers	......
Links and bearings	......	**Air Intake System**	
Springs	......	Air filters	......
Compressor Valves		Air suction ducts and mufflers	......
Compressor suction and discharge	......	**Pressure Gages**	
Scavenging pump suction and discharge	......	Air, oil and water gages	......
Scavenging System		**Relief Valves**	
Ports and automatic valves	......	Air and water	......
Exhaust gas flow regulators	......	Lubricating oil	......
Exhaust ducts and mufflers	......	Exhaust gas	......

valve seats. Excessive clearance will change the timing and cause an undesirable increase in impact, leading to breakage or wear.

VALVES AND VALVE TIMING.—It is poor practice to allow valves so to wear that reseating is difficult. Leaking valves wear rapidly, reconditioning soon is impossible and replacement is necessary. No corrections for exhaust and inlet valve timing should be made for variations of less than 5 deg. Fuel or spray valve timing is extremely important. Much care should be used in its adjustment. Recommended settings must be followed exactly.

FUEL PUMP.—Fuel pump settings and adjustments on air-injection engines determine individual cylinder loads, and have a direct effect on engine steadiness. Adjustments should be checked, and pumps made to deliver equal quantities of fuel to each cylinder. Fuel pumps on airless-injection engines measure quantity and also time delivery of fuel to the individual cylinders. It is important that they be accurately adjusted and timed.

PISTON.—The condition of piston and liner is a good indication of the quantity and quality of lubricating oil used. A bright surface on piston rings and liner, with rings free, indicates good oil supplied in proper quantities. Dull, grayish wearing surfaces, with rings stuck, indicates inferior oil. Too much oil, even of good quality, will cause a gummy deposit on rings and piston. If an engine operates with poor combustion, evidenced by smoky exhaust, a gummy residue will be found on the rings, even if a proper amount of good quality lubricating oil has been used.

Table 16.—Diesel Engine Maintenance Log

Engine No........ Year.......... Month....................

Actual Running Time for Month, hours......................

A = Date work is scheduled
B = Date work was performed

No.	Operation	Maximum Elapsed Time Allowed, hours	No. of Operations Done at Same Time	Cyl. 1		Cyl. 2		Cyl. 3		Cyl. 4		Cyl. 5		Cyl. 6	
				A	B	A	B	A	B	A	B	A	B	A	B
1	Engine pistons taken out and cleaned	3600													
2	H.p. compressor pistons disassembled, cleaned and examined	1800													
3	I.p. and l.p. compressor pistons removed, cleaned and examined	1800													
4	Check clearance space between piston and head	300	36												
5	Main cylinder jackets and heads cleaned out for scale	3600	1–2 3–20												
6	Crank-shaft and extended shaft alignment checked	3600													
7	Crank-case flushed out	1800													
8	Exhaust valves removed, cleaned and ground	900													
9	Inlet valves removed, cleaned and ground	1800													
10	Air intake valves removed, cleaned and ground	1800													
11	Air starting check valves cleaned and ground	1800													
12	Safety valves removed, cleaned and ground	1800													
13	H.p. compressor valves removed, cleaned and ground	600													
14	I.p. and l.p. compressor valves removed, cleaned and ground	900													
15	Fuel valves removed, cleaned and needles ground	900													
16	Fuel valves setting checked	300													
17	Fuel pump valves examined, cleaned and ground	1800													
18	Test spray checks for tightness every day	10													
19	Camshaft gears or chains checked and adjusted	600													
20	Cam roller setting should be checked every time a valve cage is changed or cylinder head removed	3600	5												
21	Fuel pump driving mechanism should be examined	600													
22	Valve gears cleaned, examined and corrected	300													
23	Lubricating oil filter cleaned	900													
24	Lubricating oil holes in crank-pins cleaned	1800	36												
25	Lubricating oil piping cleaned	300													
26	Lubricating oil supply piping and tank cleaned	3600													
27	Lubricating oil pumps cleaned and examined	1800													
28	Examine lubricating oil piping for loose joints and leaks	300													
29	Examine lubricator driving gears	300	6-36 37												
30	Air injection piping burned out for carbon	3600													
31	Air cooler coils cleaned out for scale	3600													
32	Air receivers cleaned	3600													
33	Air tanks drained and cleaned	3600													
34	Check fastening of tube in bottle head	900													
35	Wrist-pin bearing checked and adjusted each time pistons are removed	1800	1-2-3												
36	Crank-pin bearing checked and adjusted	1800													
37	Main bearings checked, adjusted for alignment and clearance adjusted	3600													
38	Governor examined and adjusted	1800													

Table 17.—Maintenance Instructions. Engine Clearances and Settings

Make....................Hp....................Type....................Date.........

Bearing Clearances	
Main bearing	0.006
Outboard bearing	.010
Crank bearing	.006
Piston- or crosshead-pin bearing	.005
Crosshead sh	
Air compressor piston-pin bearing	.002
Air compressor crank bearing	.004
Air compressor crosshead shoes	....
Crank-shaft thrust bearing	.006
Cam-shaft thrust bearing	.004
Vertical shaft thrust bearing	.008
Vertical shaft bearing	.004
Piston End Clearance	
High pressure piston	.032
Intermediate pressure piston	.187
Low pressure piston	.062
Scavenging piston (top)	....
Scavenging piston (bottom)	....
Main piston	.375
Cam to Rocker Arm Roller Clearance (Engine Cold)	
Fuel valves	....
Air intake valves	.031
Exhaust valves	.035
Starting valves	.035
Scavenging valves	....
Piston to Liner Clearance	
Main piston skirt	.013
High stage comp. piston	.003
Inter. stage compressor piston	.009
Scavenging piston	....

Main bearing oil pressure	5–10 lb. per sq. in.
Compression pressure	490 " " "
Combustion pressure full rated load	550 " " "
Mean indicated pressure full rated load	96.8 " " "

Valve and Fuel Pump Timing		
Fuel valve	Opens 6 deg. before T.C.	Closes 36 deg. after T.C.
Air inlet valve	Opens 11 deg. before T.C.	Closes 24 deg. after B.C.
Exhaust valve	Opens 24 deg. before B.C.	Closes 11 deg. after T.C.
Starting valve	Opens T.C.	Closes 65 deg. after T.C.

Piston Ring Clearance Ring Cut	Bevel	Butt	Gap Side
High stage comp. ring	0.025	0.045	0.003
Inter. stage comp. ring	.090	.140	.003
Low stage comp. ring	.090	140	.003
Inter. stage wiper ring	.080	130	.003
Main piston ring, top	.110	160	.005
Main piston ring No. 2	.100	150	.005
Main piston ring No. 3	.090	140	.003
Main piston ring No. 4	.080	130	.003
Main piston ring No. 5	.070	120	.003
Main piston ring No. 6	.070	120	.003
Main piston ring No. 7			
Main piston wiper ring	.070	.120	.005

Valve Lift	
High pressure delivery valve	.047
High pressure suction valve	.047
Inter. pressure delivery valve	.100
Inter. pressure suction valve	.100
Low pressure delivery valve	.100
Low pressure suction valve	.100
Scavenging valve plate type	
Scavenging valve strip type	
Fuel pump delivery valve	.025
Fuel pump suction	.130

Drops of Oil per Minute	
Each lead main power cylinder	
Each lead high stage compressor	
Each lead inter. stage compressor	
Each lead low stage compressor	

PISTON END CLEARANCE determines final compression pressure of a power cylinder. Maximum allowable variation for this adjustment is 0.035 in.

PISTON DIAMETRAL CLEARANCE should be checked whenever a piston or liner is replaced, to insure that it compares closely with that recommended by the manufacturer. Aluminum pistons require greater clearance than cast-iron pistons.

PISTON RINGS must be kept in first-class condition. If they remain in the engine after blow-by has started, power will be lost, and piston and cylinder will be scored, as a result of hot gases destroying the lubrication. Ring condition should be investigated at the first sign of blow-by.

RING AND LINER WEAR.—Piston and cylinder parts receive the greatest wear, and require the most frequent renewal. Greatest wear in the liner invariably occurs near upper end of piston travel. Wear in other parts of the liner is inconsequential, even after several years. Wear at upper end of the liner usually necessitates renewal of the part, and also replacement of piston rings. Replacement of liners is necessary when wear at top exceeds 0.005 in. per in. of cylinder diameter.

Ring and liner wear can be reduced by: 1. Using cylinder lubricant of suitable body, applied at the right points, and in correct amounts. 2. Filtering, or otherwise purifying, air and fuel. 3. Maintaining a tight seal between rings and liner. If considerable wear has occurred, this may require the use of two or more two-piece rings. 4. Keeping rings free in their grooves. 5. Keeping water-jackets free of scale, and water supply adequate.

BEARINGS.—Working clearance and lubrication determine the service that can be obtained from a bearing and its pin. A bearing loosely-fitted will flatten the pin and crack the bearing babbitt. If bearing lacks clearance, it will burn out due to excessive friction.

CRANK-SHAFT AND BEARINGS.—Crank-shaft alignment should be checked annually. Difference in quality of individual bearing lubrication, variation in quality of bearing metal, and the different bearing pressures all tend to destroy crank-shaft alignment. Bearing wear should be closely checked. If the change in clearance is not uniform on all bearings, shaft alignment should be checked.

SAFETY.—The oil vapor present in the engine crank-case, particularly at the time of stopping, is explosive. Cigarettes and lights should be kept away.

Poisonous gases collect above fuel oil in storage. These gases at times have such instant effect that they may overcome a man in a few seconds. When inspecting or cleaning fuel oil tanks all possibility of contact with such gases should be eliminated.

MANUFACTURER'S INSTRUCTION BOOKS.—Most engine manufacturers publish comprehensive instruction books giving specific information on their own engines. These books should be available to members of the operating staff who are responsible for maintenance.

13. HAZARDS AND RELIABILITY

HAZARDS.—Diesel engines are comparatively free from hazards to life and property. When accidents, however, do occur, their effects generally are confined to damage to the engine itself. The most common major accidents are seizure of pistons and breaking of connecting-rod bolts. When a piston seizes, generally as a result of faulty lubrication or of overload, piston and cylinder are scored. This may require replacement of parts, depending on depth and location of scores. Fly-wheel explosions on Diesel engines are rare, because governors seldom fail, and fly-wheels are designed for high peripheral speeds. Crank-shaft breakages, which may be caused by misalignment of main bearings, always start with a crack that gradually spreads and gives warning before any other parts are damaged. Oil vapors accumulating in the crank-case and becoming ignited have caused a few accidents resulting in blowing out of crank-case doors. Such accidents can be prevented by a crank-case ventilating system.

DIESEL ENGINE RELIABILITY.—The A.S.M.E. report for 1933 on oil-engine power cost shows 111 plants reporting fully on enforced engine shut-downs. These plants contained 285 engines with a total operating time exceeding 814,000 hours. No enforced shut-downs occurred in 1933 on 221 engines. On 74 engines, 187 enforced shut-downs aggregated 6,825 hours. The enforced shut-down time was about 0.8 of 1% of operating time. Table 18 summarizes a study of Diesel engine outages made by the Prime-movers Committee of the Edison Elect. Inst. (Publication B-87, July, 1934).

Table 18.—Summary of 1933 Operating Record for Diesel Units

	Data for 30 Engine Units with Outage Details		
	Total	Arithmetic Average per Unit	Percent of Period Hours
Number of Diesel units reviewed	30	1	....
Rated capacity of units reviewed, B.Hp.	13,950	465	....
" " " " " kw.	9,716	324	....
Kilowatt-hours generated	17,518,575	583,952	....
Period hours reviewed	262,800	8,760	100
Demand hours	109,567	3,651	41.7
Service hours	103,525	3,451	39.4
Idle hours	159,275	5,309	60.6
Outage hours	7,567	252	2.9
Outage due to engines	4,758	159	1.8
Outage due to generators	72	2.4	0.03
Outage due to auxiliaries	427	14	0.2
Outage due to other causes	2,310	77	0.9
Reserve hours	151,717	5,057	57.7

14. SPECIFICATIONS FOR DIESEL FUEL OILS

While different types of Diesel engines require different fuel characteristics, no final standards as yet (1935) have been adopted. The early Diesel engines were of slow speed and had large cylinders. Ample time was available for fuel injection and combustion, and the engines could burn any ordinary grade of commercial crude or fuel oil. The advent of high-speed engines and cracked fuels led to fuel troubles, and the setting-up by engine builders of fuel specifications. These varied among different builders, and difficulty was encountered in meeting them. A tentative specification drawn up by an A.S.M.E.

committee in 1929 failed of general acceptance, as some oils that met the specifications did not operate satisfactorily. In 1934, the A.S.T.M. subcommittee on fuels formulated the tentative classification given in Table 19.

Heating Value of Fuel Oil.—See formula p. 4–47, and table p. 4–48.

Table 19.—Tentative A.S.T.M. Classification of Diesel Engine Fuel Oils (1934)

Method of Test	Grade No.[a] 1–D	3–D	4–D	5–D	6–D
1. Flash point, min., deg. F.	115[b]	150[b]	150[b]	150[b]	150[b]
2. Water and sediment, max. percent by volume	0.05	0.2	0.6	Note [c]	Note [c]
3. Viscosity					
Saybolt Universal sec. at 100° F. min.	30	30			
Saybolt Universal sec. at 100° F. max.	50	70	500		
Saybolt Furol sec. at 120° F., max.				100	200
4. Carbon residue, max., percent	0.2	0.5	3.0	6.0	10.0
5. Ash, percent by weight, max.	0.01	0.02	0.04	0.08	0.12
6. Pour point, deg. F., max.	35[d]	[d]	35[d]	40[d]	[d]
7. Tentative ignition quality					
A. Diesel-index number, min.	45	30	20	Note [e]	Note [e]
B. Cetene number, min.	50	40	30	Note [e]	Note [e]
C. C.C.R., max.	8.1	8.8	9.8	Note [e]	Note [e]

NOTES.—[a] Grade 1–D is a distillate oil for use in engines requiring a low viscosity fuel; recommended for mechanical (solid) injection, high-speed engines; in general, for engine speeds over 1000 r.p.m.

Grade 3–D is a distillate oil for use in engines requiring medium-low viscosity fuel; recommended for mechanical (solid) injection, medium-speed engines; in general, for engine speeds from 360 to 1000 r.p.m.

Grade 4–D is an oil for use in engines requiring a medium viscosity fuel; recommended for air-injection engines, both 2- and 4-stroke cycle, with speeds not over 400 r.p.m. It can be used in solid injection engines with cylinder diameters over 16 in., and speeds under 240 r.p.m., but engine manufacturers should recommend approved heating equipment.

Grade 5–D is an oil for use in engines permitting a medium-high viscosity fuel; recommended for slow speed (under 240 r.p.m.) air-injection engines. Engine manufacturers should recommend approved heating equipment.

Grade 6–D is an oil for use in engines of special design for high viscosity fuels, and after engine manufacturer's recommendations only. It is not used regularly for Diesel engines and is not recommended unless tested and approved by engine manufacturer. Purchaser should be advised of high cost of engine maintenance and operating problems involved in its use.

[b] Minimum flash point as stated, or as required by local fire regulations, Fire Underwriters' rules, or state laws.

[c] Not determined (1935).

[d] Lower pour points may be specified whenever required by local temperature conditions, to facilitate storage and use, although it should not be necessary to specify a pour point of less than 0° F.

[e] Ignition quality not specified, but burning quality should be determined by actual engine test.

Sulphur apparently need not be considered as regards combustion characteristics. A maximum of 2% is suggested for engines operating intermittently.

COMBUSTION KNOCK AND IGNITABILITY.—Combustion knock, in a rough running engine, is attributed to accumulation of fuel in its cylinders, prior to ignition. Even under conditions that insure ignition, fuel does not ignite instantaneously, but only after a definite delay. The longer the delay, the greater the accumulation of fuel, which then burns more or less simultaneously, accompanied by an audible knock. If the delay is extreme the engine will fail to operate. Practically every factor which tends to aggravate knocking in a spark-ignition engine tends to suppress it in Diesel engines. Fuels of paraffin base, consisting chiefly of saturated, straight-chain hydrocarbons give the smoothest combustion in Diesel engines, while naphthenes and aromatics burn more roughly.

The characteristic of a fuel that determines its speed of ignition under conditions prevailing in an engine cylinder is called ignitability or ignition quality. Crucible and bomb methods of measuring ignition temperatures are not trustworthy methods of determining ignitability. The time lag in bombs is many times the permissible time lag in an engine, and has a pronounced effect on ignition temperature. This effect varies for different fuels. Unfortunately tentative ignition quality in the specifications is diminished in value, for the time being, by three different methods of specifying it being offered without a standardization of test methods.

DIESEL INDEX NUMBER is one of the three methods of specifying ignition quality proposed by the A.S.T.M. It is based on tests that indicate ignitability of an oil to vary in accordance with its aniline point and its gravity. The A.S.T.M. tentative specification uses the formula of Dr. Becker as follows:

Diesel Index Number = (A.P.I. Gravity at 60° F.) × (Aniline Point in deg. F.)/100

Aniline Point is the lowest temperature at which equal parts by volume of freshly distilled aniline and oil are completely miscible. It is determined by heating the mixture in a jacketed test tube to a clear solution and noting temperature at which turbidity appears as mixture is cooled. Although this is a convenient method, the correlation between Diesel index number and ignition quality has been found good only for certain types of oil. It, therefore, is recommended for quick and rough evaluation only.

ENGINE TESTS.—Most investigators have concluded that a compression-ignition engine itself is the best apparatus to test ignition quality of fuels. A variety of methods have been used by Le Mesurier and Stansfield, Boerlage and Broeze, and Pope and Murdock; in principle, however, all agreed. Ignitability, as such, is not measured by any engine test, but is determined indirectly on the basis of the behavior of the fuel in the test engine, as regards such characteristics as combustion knock, ignition lag, rate of pressure rise, or ease of starting.

SCALE FOR EXPRESSING IGNITABILITY.—Whatever testing method is used to measure ignitability, it is essential to determine the manner of rating of the fuel. One way is to use a measured factor or quantity to express rating, as maximum injection advance angle, ignition lag in degrees, or critical compression ratio. The last-named factor is one of the choices set forth in the A.S.T.M. tentative classification.

CRITICAL COMPRESSION RATIO is determined on a C.F.R. Waukesha engine, modified to run as a Diesel by replacing the ignition system with a fuel pump and fuel injection valve, and also replacing the regular piston with one of special design. The operating conditions are: Speed, 600 r.p.m.; fuel feed, 9 c.c. per min.; injection advance angle, 12 deg. before top dead center; injection pressure, 1500 lb. per sq. in.; jacket temperature, 212° F.; air inlet temperature, 100° F.

The critical compression ratio is obtained by motoring the engine at the selected speed. The compression ratio at which firing occurs, as judged by the sound of the exhaust, within three seconds after the fuel is injected, is noted. Although the critical compression ratio so determined for any fuel in the test engine differs from what an actual engine might require to fire that fuel, it nevertheless is of practical value. The engine builder can run tests on his engine with several fuels of known critical compression ratio. He then is able to specify the maximum ratio that will give satisfactory performance.

CETENE NUMBER is not a method of testing ignition quality, but of expressing the rating after the fuel has been tested. The method is derived from that used for testing the knock rating of gasoline. (See p. 14–68.) It is based on the use of two standard reference fuels, cetene ($C_{16}H_{32}$) and alpha methylnaphthalene ($C_{11}H_{10}$), the former having a high, the latter a low, ignitability. The test fuel is matched with a fuel consisting of the two reference fuels mixed in such a proportion that its ignition quality, in the arbitrary scale of the test, is equal to that of the test fuels. The rating of the fuel is then stated as percentage concentration of the high-rate reference fuel, *i.e.*, Cetene Number.

Table 20.—Cetene Number Corresponding to Given Viscosities and Gravities of Diesel Engine Fuel Oil

Viscosities in Saybolt Universal seconds at 100° F.

Viscosity	Gravity, deg. A.P.I., at 60° F.																
	18	20	22	24	26	27	28	29	30	31	32	33	34	35	36	37	38
	Cetene Number																
34	21	25	29	33	37	39	40	42	44	46	47	49	51	52	54	55	57
35	22	26	30	34	38	40	42	44	46	48	49	51	53	54	56	57	59
36	23	27	31	35	39	41	43	45	47	49	51	52	54	56	58	59	61
37	24	28	32	36	40	42	44	46	48	50	52	53	55	57	59	60	62
38	24	29	33	37	41	43	45	47	49	51	53	54	56	58	60	61	63
39	25	29	33	37	41	43	45	47	49	51	53	55	57	59	61	62	64
40	25	30	34	38	42	44	46	48	50	52	54	56	58	60	62	63	65
42	26	30	35	39	43	45	47	49	51	53	55	57	59	61	63	64	66
44	26	31	35	40	44	46	48	50	52	54	56	58	60	62	64	65	67
46	27	32	36	41	45	47	49	51	53	55	57	59	61	63	65	66	68
48	27	32	36	41	45	47	49	51	53	55	57	59	61	63	65	67	68
50	28	33	37	42	46	48	50	52	54	56	58	60	62	64	66	68	69
60	29	34	39	43	47	50	52	54	56	58	60	62	64	66	68	69	70
80	31	35	40	45	50	52	54	56	58	60	63	65	67	69	71	72	...
100	32	37	42	47	52	54	56	58	60	62	65	67	69	71	...	...	...
150	34	39	44	49	54	56	58	60	63	65	67	69	72	...	...	...	...
200	35	40	45	50	55	58	60	62	65	67	69	71	...	...	...	...	...
300	36	41	47	52	57	60	62	65	67	69	72	...	...	...	...	...	...
400	37	42	48	53	59	61	64	66	68	71	...	...	...	...	...	...	...
500	38	43	49	55	60	63	65	67	70					...	...	...	...

The test method used to find the cetene number may be: 1. A starting test with a motored C.F.R. engine, as in the determination of critical compression ratio. 2. Ignition lag method by direct reading the ignition lag with a neon light protractor. 3. Knockmeter method using a bouncing pin. (See p. 14-68.)

Relation of Cetene Number to Viscosity and Gravity.—After using the ignition lag method on a modified C.F.R. engine to determine the cetene numbers of a large number of fuel oils ranging from 34 to 72 cetene number, C. C. Moore, Jr. and G. R. Kaye found a close correlation between the cetene number and the viscosity and gravity of the oil, as shown in Table 20.

SOUR OIL.—Another oil characteristic has (1935) become prominent with the increasing use of high-speed Diesel engines in the sour oil fields of West Kansas and Texas. These crude oils contain hydrogen sulphide in solution which causes considerable corrosion of the finely fitted parts of injection nozzles and fuel pumps. In addition, the oil contains finely-divided abrasives, as sand and ferrous sulphide. To protect engine parts from attack, Behn, at an A.S.M.E. meeting at Tulsa, Okla. (*Diesel Power*, May, 1935), recommended thorough cooling of the injection nozzle, minimum heating of the fuel oil, adequate filtering of the fuel, settling tanks to remove water and vents to discharge gas as emitted.

GAS ENGINES

By H. A. Gehres

1. GAS ENGINE FUELS

A gas engine is a prime mover whose piston is actuated by the combustion of gaseous fuels in air.

GASES AVAILABLE FOR USE IN GAS ENGINE.—**Natural Gas** (dry), is an ideal fuel, of an average heating value of about 1000 B.t.u. per cu. ft., consisting of a mixture of several light hydrocarbons, but mostly methane, CH_4. It is obtained from deposits in the earth, but not connected with the production of oil. It is piped under pressure from the fields to the consuming center. Natural gas pipe lines now serve a considerable portion of the United States. See p. 4–64 for analyses and heating value of natural gas from various fields.

Casinghead Gas is excellent fuel, whose heating value ranges from 1200 to 2000 B.t.u. per cu. ft. It comes from oil wells, and is trapped off at the casinghead as the oil is produced. The wide variation in heating value is caused by the varying quantities of condensible vapors, such as butane contained in it. A considerable quantity of this gas is used in the oil fields for well pumping and other lease uses. Some of it, after being treated for gasoline recovery, is used in pipe lines. Much of it is used to make carbon black, and great quantities are blown to the air.

Refinery Gas is a by-product from the distilling and cracking of oil for making gasoline in oil refineries. The heating value varies from 1200 to 2000 B.t.u. per cu. ft. It is used mostly for heating stills, making steam, and in gas engines for making power in the refineries. In some cases there is enough excess to sell to outside consumers.

Coke-oven Gas is a good gas for engines when properly cleaned. It has a heating value of from 500 to 600 B.t.u. per cu. ft. It is procured from by-product coke ovens. The excess above that required for the firing of the stills is available for distribution. Large quantities are used in steel mills. See p. 4–64.

Producer Gas can be used in gas engines if properly cleaned. It is artificially made in gas producers, and the main combustible constituent is CO. If made from bituminous coal it also contains some of the distilled hydrocarbons. The heating value is from 130 to 150 B.t.u. See p. 13–03.

Illuminating Gas (City manufactured gas) is a good gas for engines but very restricted in its use because of high cost. Its heating value is from 500 to 600 B.t.u. per cu. ft.

Blast Furnace Gas is an excellent fuel for gas engines when properly cleaned, although low in heating value (90 to 100 B.t.u. per cu. ft.). It is a by-product from making iron in blast furnaces. In most cases it is all used in the plant where it is made. See p. 4–64.

Relative Power Value of Gas Engine Fuels.—The value of a gas from an engine rating standpoint is not in proportion to its heating value but is in proportion to the heating value of the combustible mixture. Table 1 gives the approximate heating value of the various gases used in gas engines, and also the heat available from the explosive mixture. From

Table 1, the heating value of coke-oven gas is only about two-thirds that of natural gases, yet their B.t.u. value per cu. ft. of explosive mixture is the same. Therefore, an engine can be rated the same using either one of these two gases.

Table 1.—Approximate Heating Value of Commercial Gases

Taken at 60° F. and at Pressure of 30 in. of Mercury

Gas	Lower Heating Value, B.t.u. per cu. ft.	Average Air required for Combustion, cu. ft. per cu. ft. of Gas	Average B.t.u. per cu. ft. of Explosive Mixture
Natural Gas, Dry	970–1000	9.73	91.0
Refinery and Casinghead Gas	1200–2000	11.5–19	
Oil Gas	820–850	8.07	93.0
Coke-oven Gas	500–603	5.60	91.0
Carbureted Water Gas	550–575	5.25	92.0
Producer Gas from Hard Coal	140–153	1.12	68.0
Producer Gas from Soft Coal .	144–150	1.20	65.5
Producer Gas from Coke. . . .	120–140	0.98	63.0
Blast-furnace Gas.	90–95	0.72	53.0

2. TYPES OF GAS ENGINES

CLASSIFICATION.—Gas engines may be classified in the following various ways:

1. **By Combustion Cycle:** *a.* Four cycle. *b.* Two cycle.
2. **By Power Impulses:** *a.* Single acting. *b.* Double acting.
3. **By Arrangement:** *a.* Vertical. *b.* Horizontal. *c.* V-type. *d.* Opposed. *e.* Radial.
4. **By Speed:** *a.* Low Speed (100 to 250 r.p.m.). *b.* Intermediate Speed (250–600 r.p.m.). *c.* High Speed (600–2000 r.p.m.).

CYCLES OF OPERATION. Four-cycle Principle.—The term 4-cycle is applied to engines in which four complete strokes are required to complete a cycle of events. Fig. 1 shows the various phases of this cycle, in the form of an indicator diagram, in their regular sequence as follows:

1. Suction Stroke, *a–b*. A certain mixture of gas and air is drawn into the cylinder by action of the piston. Pressure in the cylinder drops below atmospheric pressure, due to resistance through passages, valves, etc., but mainly due to the throttling action of the governing or regulating valves.

2. Compression Stroke, *b–c*. After completion of the suction stroke, the inlet valve closes and the returning piston compresses the confined mixture to a predetermined pressure called *compression pressure.*

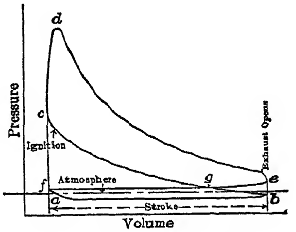


FIG. 1. Four-cycle Indicator Diagram

3. Power Stroke, *c–d–e*. The compressed mixture is ignited shortly before the piston reaches dead center, causing pressure to rise suddenly from *c*, through rapid combustion or explosion, to point *d* or *peak pressure.* Expansion then takes place, pressure gradually falling from *d* to *c*. Just before reaching dead center the exhaust valve opens and the *terminal pressure* suddenly falls to just above atmosphere.

4. Exhaust Stroke, *e–f*. The returning piston expels the remaining burned gases at an *exhaust pressure* slightly above atmosphere. The amount of this pressure depends on the fluid resistance through exhaust valves, piping, mufflers, etc.

Of the four strokes represented only one is a power stroke.

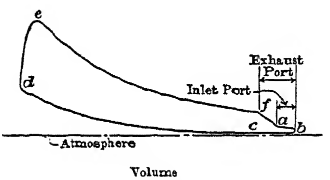
The positive work of this cycle is represented in Fig. 1 by the enclosed area *cdeg*, and the negative work (pumping) by area *abgf*. This negative work varies somewhat with the design of the engine, but mostly with the percentage of load at which the engine is operating. At full load the lost work is very low. As most engines, except some of the smaller sizes, use throttling governing, the lighter the load the lower the suction pressure, and consequently the greater the negative work in proportion to the positive work.

The 2-cycle Principle.—With this principle the complete cycle covers only two strokes of the power piston. To accomplish this the pumping is done by other means than in the power cylinder. The 2-cycle principle lends itself to many variations in design. These variations are mainly concerned with the method of *pumping* and of using the scavenging air. The simplest and most broadly used two cycle is the plain-ported type, which has no valves in the power cylinder. At about the middle of the cylinder are two sets of ports inlet on one side and exhaust on the other. These are so placed that the piston uncovers

both sets of ports at the end of the stroke away from the head. Pumping is done by the under, or crank, side of the piston, using for a pump the crank-case in a trunk piston engine, or a passage sealed from the crank-case, by a partition plate, in a crosshead engine. An automatic suction valve admits the mixture on the crank side on the out stroke of the piston; at the same time a previous charge is being compressed in the opposite end of the cylinder. Near the end of the power stroke first the exhaust port is uncovered, the terminal pressure falling rapidly, and then the inlet port is uncovered. The mixture compressed ahead of the piston flows through, pushing out the remaining burned gases and leaving in the cylinder a fresh charge.

Fig. 2 is a typical diagram of such an engine, admission and scavenging beginning at *a*, continuing to end of stroke and back to *a*, slightly above atmosphere. Compression begins at *c* when exhaust port closes and continues to *d*. Ignition takes place a little before dead center, and pressure rises to peak at *e*. Expansion takes place to *f*, where exhaust port opens and pressure rapidly falls to *a*, where a new charge starts in, all being accomplished in two strokes.

RELATIVE ADVANTAGES AND DISADVANTAGES OF TWO- AND FOUR-CYCLE ENGINES. Advantages, 4-Cycle.—1. Accurate control of fuel-air mixtures. 2. Lower fuel consumption (8000 to 10,000 B.t.u. per Hp.-hr.). 3. Better regulation. 4. Higher speeds obtainable. 5. Lighter weight pistons. 6. Lower lubricating oil consumption.



. 2. Two-cycle Indicator Diagram

Advantages, 2-Cycle.—1. Simple construction. 2. Lower manufacturing cost. 3. Lower maintenance costs.

Disadvantages, 4-Cycle.—1. Less uniform turning effort for equivalent number of cylinders. 2. More costly to build for equivalent horsepower. 3. Necessity of complicated valve gear. 4. More expensive to maintain.

Disadvantages, 2-Cycle.—1. Necessity of using quality-of-mixture regulation, resulting in "missing" at both light and heavy loads. 2. Higher fuel consumption. 3. Inaccurate regulation on variable loads due to missing. 4. Lower rotating speeds due to extremely short time for scavenging. 5. Heavier pistons. 6. Higher oil consumption.

3. COMBUSTION AND REGULATION

CONTROL.—Two methods are used to control the load and speed of a gas engine. The first varies the quantity of a constant mixture of gas and air. The second varies the fuel only.

With gases, a certain ratio of gas to air exists where a perfect mixture will result in perfect combustion when ignited, leaving no unburned fuel and no free oxygen. See Table 1. The range of variation from this perfect mixture, either rich or lean, wherein combustion will take place at all, is limited.

For example, with methane the range is 7% to 13% of gas in the total mixture. As the mixture is varied, either "rich" or "lean" from these limits the engine first will begin to fire irregularly, and finally cease to fire.

Four-cycle engines use the first method of control almost exclusively. The regulating valve controls both gas and air simultaneously, maintaining a constant gas-to-air ratio, but reducing compression at light loads. The 4-cycle engine will, therefore, fire regularly over a wide range of loads.

The 2-cycle engine, of necessity, uses quality regulation, *viz.*, the regulating valve controls gas only. If the air were throttled a corresponding amount of burned gases would be left in the cylinder which would tend to cause prematuring and backfiring. There are two methods of introducing the fuel into 2-cycle engines: 1. Fuel is taken in by suction with the scavenging air; in this way some of the fuel is flushed on through the exhaust ports without burning. 2. Fuel is injected under pressure near, or after, closing of the exhaust ports, but early in the compression stroke.

FUEL CONSUMPTION of gas engines, expressed as heat consumption ranges about as follows: 4-cycle engines, 8,000 to 10,000 B.t.u. per Hp.-hr.; 2-cycle engines (plain), 13,000 to 15,000 B.t.u. per Hp.-hr.; 2-cycle (injection), 10,500 to 12,000 B.t.u. per Hp.-hr.

POWER AND EFFICIENCY FORMULAS.—Power and efficiency formulas are the same for gas engines as for Diesel engines. See page 12–06.

Thermodynamic Analysis.—See page 12–07.

COMPRESSION PRESSURES AND TEMPERATURES.—While the higher the compression pressure the higher will be the thermal efficiency (see Diesel Engines, page 12-07), the gas engine is definitely limited in the compression pressures that can be used, depending on the nature of the gas used as fuel. Since gas engines have the fuel added to the air before or shortly after compression begins, auto-ignition of the charge can occur during compression if compression is too high. This is called preignition. The higher the hydrogen content of the gas used, the lower it is necessary to hold the compression.

The theoretical expressions for temperature-pressure and pressure-volume relations are:

Pressure-temperature relations: $T_2/T_1 = (P_2/P_1)^{(n-1)/n}$;

Pressure-volume relations: $P_1(D + C)^n = P_2 C^n$,

where T_1, T_2 = respectively, absolute temperatures at beginning and end of compression; P_1, P_2 = respectively, initial and final absolute pressures, lb. per sq. in.; D = piston displacement, cu. in.; C = clearance volume, cu. in.; the value of exponent n ranges from 1.32 to 1.35 in water-cooled engines, and is about 1.4 in air-cooled engines.

COMPRESSION PRESSURES USED in gas engines vary with the type of fuel. Usual pressures are as follows:

Refinery and casinghead gas	70–90	lb. per sq. in., gage.
Natural gas (dry)	100–115	" " "
Coke-oven gas	80–100	" " "
Producer gas	120–140	" " "
Blast-furnace gas	150–180	" " "

These values are for dry 4-cycle pistons, not over 19 in. diameter. They should be reduced 5% for dry 2-cycle pistons up to 17 in. diameter, and increased 10% for all water- or oil-cooled pistons of any diameter.

HEAT RECOVERY.— The same principles of heat recovery that apply to Diesel engines (see p. 12–11) can be applied to gas engines, except that due to the lower compression used the cycle is not as efficient and more heat is available from the jackets and exhaust. An approximate heat balance for gas engines is: Brake work, 25%; friction, 7%; jacket water, 30%; exhaust, 35%; radiation, etc., 3%.

4. UTILIZATION OF GAS ENGINES

FOUR-CYCLE ENGINES. Single and Twin Tandem, Double-acting, Crosshead Type.—The largest sizes, 2000 to 7000 Hp., are used mostly in steel plants. They use blast-furnace and coke-oven gas for fuel. They are used to develop electric power, and for direct driving blowers supplying furnace air. Smaller sizes, 800 to 1500 Hp., are used for electric power generation. Many are used for direct driving of compressors for pumping natural gas through pipe lines.

Single-acting, Single-cylinder Trunk-piston Type.—Built in sizes up to 75 Hp. These engines are built for general purposes and for oil well pumping.

Single-acting, Single and Twin Crosshead Type.—Built in sizes from 25 to 250 Hp. These engines are used to generate electric power, and also are used in large numbers for pumping gas for pipe lines, and for gasoline extraction from casinghead gas.

Single-acting, Multi-cylinder Trunk-piston Type.—Built in all sizes up to 1500 Hp. These engines are used for general power generation, direct driving of ammonia compressors, oil line pumps, etc.

TWO CYCLE.—Single and Twin Double-acting Type.—A small number have been built to use blast-furnace gas.

Single and Twin Single-acting Crosshead Type.—Very large numbers of this type of engine are used for oil well pumping and cable tool drilling, casinghead gasoline extraction plants. They are built in sizes of 15 to 250 Hp.

Single and Twin Cylinder Trunk Piston (crank-case scavenging).—Used for oil well pumping, and cable tool drilling.

5. IGNITION SYSTEMS

MAKE-AND-BREAK SYSTEM.—The make-and-break ignition system consists of mechanical igniters, comprising a body, a stationary electrode and a movable electrode. Some types have only one electrode, insulated and connected to the current source, the return current going through the engine structure. Others have complete copper circuits with both electrodes insulated, usually by means of mica, porcelain or lava.

The movable electrode makes contact with the stationary electrode, and at the proper time the contact is broken, either by mechanical or magnetic means, and the resulting spark ignites the charge. A rather heavy induction coil is connected to each igniter

The current usually is supplied by storage batteries, floating on the line, with voltage from 40 to 110, or by a low-tension magneto. When 40-volt current is used on a 110-volt circuit, a resistance with a suitable rheostat for varying the charge is used. This system was used almost universally for many years, but in recent years it rapidly is giving way to the high-tension jump spark system, because of its lower cost and absence of the heavy maintenance expense of the mechanical igniters.

JUMP-SPARK SYSTEM.—The jump-spark system is based on a small plug which has one insulated electrode and one grounded to the engine, with a fixed gap between them. Current is supplied at a sufficiently high voltage to cause a spark to jump the gap. Two systems are in general use: 1. *Battery,* with either high-tension or low-tension distribution. 2. *High-tension Magneto.* With battery ignition, the high-tension distribution is used most generally. It consists of a battery, an interrupter, a breaker, a transformer type of coil, condenser, and the jump-spark plugs in the cylinders. It functions as follows:

The battery is connected in series through the breaker and the primary circuit of the coil. A timing cam allows the breaker points to close at the proper time so that current passes through the primary coil, building up a magnetic flux in the iron core. The cam immediately opens the breaker, causing the collapse of the flux in core, which induces high-voltage current in the secondary winding. A small condenser is bridged across the breaker points to absorb current which would try to flow across the gap as the breaker opens, thus preventing burning of the points. The condenser is discharged the next time the points make contact. Current induced in the secondary is carried to a central terminal of a distributor, which connects to the center of a rotating arm. Arranged around the distributor shell are terminals, each connected through heavily insulated cable to a spark plug. As the outer end of the arm passes each terminal, the breaker closes and opens, inducing a spark at the plug connected to that terminal. Thus one breaker, one condenser and one coil will handle any number of cylinders. The breaker, distributor and condenser usually are built into one unit and the coil into another.

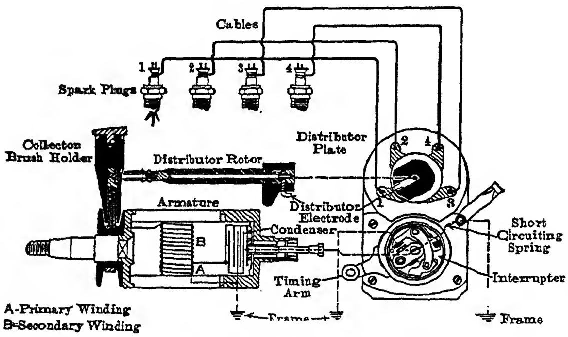


FIG. 3. High-tension Magneto Ignition System

In very large engines, particularly those of the horizontal type, the very long high-tension wires become a disadvantage and low-tension distribution often is used. The principle is the same, except that each spark plug has a separate coil located near it. These separate coils can be operated from a low-tension distributor with a single breaker and condenser. The most reliable, however, have a breaker and condenser for each. In this way, a failure of any of the parts affects only one cylinder.

THE HIGH-TENSION MAGNETO SYSTEM has a basic principle similar to that of the jump-spark system (see above), except that the source of current is one of the various forms of magnetos. The magneto originated as a small electric generator with a rotating armature but with the field flux supplied by permanent magnets. Low-voltage current was supplied to make-and-break and low-tension distributor systems. Later designs produced the high-tension current direct. The armature contains both primary and secondary windings. As the armature rotates, a low-tension current is produced in the primary winding. A pair of breaker points connected to this winding forms a short circuit. At the point of maximum current flow the points are opened, interrupting the current, and thus inducing a high-tension current in the secondary winding, which is connected through a distributor and cable to the plugs. See Fig. 3.

The trend now (1935) is toward the more simple, less costly and more durable inductor type. This type simply reverses the direction of flux through the iron core of a stationary coil, thus inducing current in the winding. The only rotating or oscillating member is the iron pole pieces imbedded in some non-magnetic material. In other types the magnet is rotated and the pole pieces are stationary. In neither type are there windings on the

rotor. As the pole pieces rotate past the magnet or *vice-versa* the magnetic flux through the coil core is reversed, thus inducing current in the primary winding, which when interrupted near the time of greatest current induces the high-tension current in the secondary winding. See Fig. 4.

BATTERY SYSTEMS are used mainly where several engines operate in a group, and the battery up-keep is justified, and also where batteries are necessary for lighting and other purposes. They also are recommended for the very large engines where low-tension distribution is desirable.

ROTARY MAGNETOS require impulse couplings for starting. This is a spring tripping device, which when engaged, either manually or automatically, winds up and trips, giving the rotor a very quick partial rotation, which makes a strong spark at low engine speed. When the speed is increased sufficiently, the impulse device is automatically disengaged and the rotor continues to revolve uniformly.

Rotary magnetos can be converted into an oscillating type by means of a tripping arm fastened to the end of the rotor, and held in position by two strong tension springs. The tripping cam pushes the arm to one side until it slips off. The springs return the rotor to the original position rapidly enough to produce a spark.

Another type of oscillating magneto is made with a bar type magnet extending across the cores of two coils, whose windings are in series. A sliding bar with roller at its outer end, operated alternately with a cam or eccentric, causes the magnet to come in contact with the cores, and close a pair of breaker points, and then breaks the contact between the magnet and cores, at the same time opening the breakers. Current is induced in the primary circuit by the flux collapse when the magnet is pulled away from the coil cores, and in the secondary at the parting of the breaker points, which causes the spark at the plug.

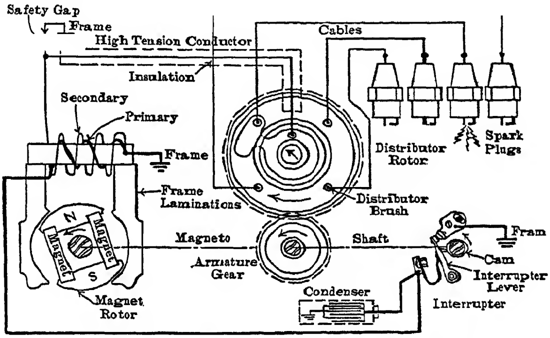


Fig. 4. Rotating-magnet Magneto Ignition System

Oscillating magnetos have the advantage of giving the same degree of spark at all engine speeds, but they are limited to comparatively slow-speed engines.

6. FACTORS AFFECTING ENGINE PERFORMANCE

EFFECT OF AIR INTAKE TEMPERATURE ON ENGINE RATINGS.—As the intake air varies in temperature, the weight of air in each charge varies. The warmer the air the less the weight of the charge, and consequently the lower the maximum rating that can be obtained. Table 2 gives the relative capacity and efficiency, with unity taken at 60° F.

Table 2.—Relative Capacity of Gas Engines at Various Temperatures of Intake Air

Initial Temp., deg.		Relative Capacity Or Intake Efficiency	Initial Temp., deg.		Relative Capacity Or Intake Efficiency
Fahr.	Absolute		Fahr.	Absolute	
−20	441	1.18	70	531	0.980
−10	451	1.155	80	541	.961
0	461	1.13	90	551	.944
10	471	1.104	100	561	.928
20	481	1.083	110	571	.912
30	491	1.061	120	581	.896
32	493	1.058	130	591	.880
40	501	1.040	140	601	.866
50	511	1.020	150	611	.852
60	521	1.000	160	621	.838

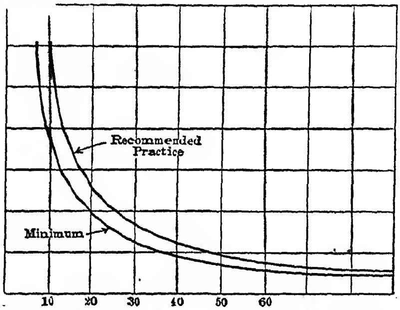
EFFECT OF ALTITUDE ON ENGINE RATING.—See Table 3. The effects of increased air temperature and altitude are similar in their effect on engine rating. It is not

necessary in most cases to reduce the rating under 3000 to 5000 ft. altitude, as an engine may be capable of 100 to 110 lb. mean indicated pressure (M.i.p.) at sea level but for reasons of long life would not be rated over 75 M.i.p. If satisfied with an overload capacity of 12% the top rating could be 84 M.i.p. Therefore, an altitude of 5000 ft., or an inlet temperature of 160° could be used, before a loss in rating would be recognized.

COOLING WATER REQUIRED.—The cooling water required for a gas engine varies with the inlet and outlet temperatures. Table 4 gives the requirements for a medium-speed, single-acting gas engine of 16- to 18-in. bore, with dry pistons, for various inlet temperatures and 140° F. outlet temperature. The recommended amounts are for establishing the capacity of pumps, cooling towers, heat exchangers, etc., in order to provide reserve capacity.

For double-acting engines with water- or oil-cooled pistons and rods multiply the figures of Table 4 by 1.25. For small, high-speed engines multiply them by 0.90. For radiator or tank cooling temperatures up to 180° F. outlet, and 166 to 170° F. inlet are permissible. Where the water supply is 80° F., or below, it is good practice to recirculate part of it to avoid too large a temperature gradient through the engine.

Fig. 5 is a curve giving approximate gallons per B.Hp.-hr., plotted against temperature difference between inlet and discharge.



Temperature Difference between Inlet and Discharge, deg. F.

Fig. 5. Cooling Water Consumption of Gas Engines

Where the water supply tends to deposit scale the outlet temperatures should be kept to 125° F. or below.

7. LUBRICATION

Three methods of general lubrication are in use on gas engines, *viz.*, Splash, Sight Feed and Pressure.

SPLASH SYSTEM.—The crank-case is filled with oil to such a level that some part of the connecting rod, crank, or counterweight dips into it. The resulting splash throws oil into pockets in the various bearings, whence it runs into the bearings. Its main advantage is simplicity, as no pumps and little or no piping are required.

SIGHT FEED SYSTEM.—A low-pressure circulating pump is used with pipes leading to adjustable sight feeds above each bearing. The advantage of this system is that the operator can see and adjust the amount of oil to each bearing and a filter can be used. This system is used mainly on horizontal engines.

THE PRESSURE SYSTEM consists of a pump capable of pumping from 10 to 40 lb. pressure, and a piping system leading directly to the bearings. This system is used mostly on vertical and higher speed engines. Its advantages are: Large quantities of oil flushed through the bearings for both lubrication and cooling; ability to deliver directly to

Table 3.—Influence of Altitude Above Sea Level on Volumetric Efficiency

Altitude Above Sea Level, ft.	Barometric Pressure		Relative Volumetric Efficiency, percent	Decrease in Volumetric Efficiency, percent	Altitude Above Sea Level, ft.	Barometric Pressure		Relative Volumetric Efficiency, percent	Decrease in Volumetric Efficiency, percent
	In. of Mercury	Lb. per sq. in.				In. of Mercury	Lb. per sq. in.		
0	30.00	14.75	100	0	6,000	23.86	11.73	81	19
1000	28.88	14.20	97	3	7,000	22.97	11.30	78	22
2000	27.80	13.67	93	7	8,000	22.11	10.87	76	24
3000	26.76	13.16	90	10	9,000	21.29	10.46	73	27
4000	25.76	12.67	87	13	10,000	20.49	10.07	70	30
5000	24.79	12.20	84	16					

Table 4.—Cooling Water Requirements of Medium-speed, Single-acting Gas Engines

Outlet Temperature, 140° F.

Inlet Temp., deg. F.	50	60	70	80	90	100	110	120
Gal. per B.Hp.-hr. Minimum	4.06	4.57	5.23	6.10	7.32	9.15	13.40	18.30
Gal. per B.Hp.-hr. Recommended	5.33	6.00	6.86	8.00	9.60	12.00	16	24

a piston-pin or crosshead-pin in vertical engines; it allows the use of much higher bearing pressures than other systems, so that reduced weights and more compact dimensions can be attained. Pressure-type filters are used with this system, and on medium- and large-size trunk-piston engines, oil coolers or heat exchangers usually are furnished.

Cylinders on small- and medium-size trunk-piston engines usually are lubricated by splash. On crosshead and large-size trunk-piston engines, the cylinders are lubricated by one or more feeds to the cylinder bore which are piped through separate lines direct from the force feed pump. The type of pump used for this service consists of a small tank with drive, and a separate pump plunger and barrel for each cylinder feed. A different oil from that used in the crank-case then may be used in the cylinders. For engine-driven compressor work, these pumps can be had with two compartments so that a different oil from that used in the engine cylinders may be used in the compressor cylinders.

OIL CONSUMPTION.—Lubricating oil consumption for high-speed, trunk-piston, 4-cycle engines should be from 1500 to 2500 Hp.-hr. per gallon; for medium-speed 4-cycle engines, 2500 to 3500 Hp.-hr. per gallon; for slow-speed 4-cycle engines, up to 5000 Hp.-hr. per gallon.

For slow-speed crosshead engines, lubricating oil consumption should be from 4000 to 7000 Hp.-hr. per gallon; for trunk-piston, 2-cycle engines, 1500 to 2000 Hp.-hr. per gallon; for crosshead, 2-cycle engines, 3000 to 4000 Hp.-hr. per gallon. These figures are based on full ratings.

FILTRATION.—The life of an engine varies directly with the quality of its lubrication. With an ample oil film and a lubricant absolutely free of all abrasives there should be no wear at all in a bearing. Wear is caused by particles of abrasive that are greater in size than the thickness of the oil film.

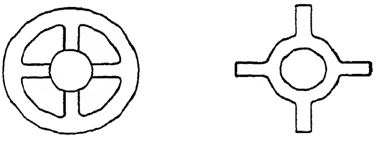


Fig. 6. Elements of Pressure-type Metal-edge Filter

Splash-lubricated engines usually are not provided with filters as a concession to simplicity and economy. A filter can be used with these engines, however, if a pump is provided to pass oil from the crank-case through the filter, whence it returns to the crank-case either by gravity or pressure.

Sight-feed-lubricated engines almost universally are provided with filters capable of continuously handling the entire amount of oil circulated, since this amount is relatively small. Pressure-lubricated engines are furnished with all degrees of filtration from 0 to 100% of the oil circulated.

Because of the large amounts of oil circulated, partial filtration is quite common, particularly on small portable engines. The filter either is installed in a parallel circuit to the main system, filtering what it can and returning it to the crank-case, or is placed in series with the main system where the filtering element takes what it can, the remainder going through a spring-loaded by-pass valve.

Types of Filters.—Filters may be classified as: 1. Gravity type fabric cloth filter. 2. Pressure type fabric cloth or felt filter. 3. Pressure type metal cloth filter. 4. Pressure type metal edge filter.

The first type probably produces the best filtration, as such filters usually are large, with a low rate of flow. They are practical only for large stationary installations.

The second type will produce quality of filtration in inverse proportion to the flow. The slower the flow, the better the quality of filtration. They are used for both full and partial filtration. While the fabric cloth filter produces the best filtration, its capacity is seriously affected by water in the oil, which closes the fabric and reduces the capacity.

The third type is really more of a strainer than a filter. It is more convenient to clean than other types, being built mostly in basket form. When the baskets are lifted out they bring the dirt with them.

The fourth type is used most commonly for portable and marine engines. The filtering elements consist of stacks of thin washers alternating with spacers of from 0.0015 to 0.006 in. thick. The washers have an unbroken outer periphery (see Fig. 6*a*), while the spacers have only radial " spokes " (see Fig. 6*b*). The oil passes from the outside of the stack through the space made by the spacer and passes on through the passage through the center of the stack. Other types have crimped wire wound around a perforated cylinder or a crimped ribbon wound in the form of a disc. The amount of crimping determines the degree of filtration.

These filters can be made for all degrees of filtration. They can be so fine that the collected dirt forms a mat, producing a finer filtration but slower flow until the amount

flowing is so small as to require cleaning of the filter. To be practical, this type requires large surface area. Many of them, as a concession to expense, space and less frequent cleaning, only catch the coarser particles.

Various methods of cleaning are in vogue. The cloth bag filters and some of the edge filters must have the elements removed and washed. Some of the metal edge filters have cleaning discs or scrapers, which can be operated by an outside handle, the sludge dropping to the bottom of the case where it is drained off. Others reverse the circulation of oil for a short time. Still others blow compressed air in a reverse direction through the filter element.

8. OPERATION AND CORRECTION

DEFECTIVE OPERATION AS SHOWN BY INDICATOR CARDS.—Fig. 7 shows a perfect type of indicator card as taken from a horizontal, 4-cycle gas engine. The M.i.p. is 85.5 lb. per sq. in. with only 250 lb. per sq. in. peak and 30 lb. per sq. in. terminal pressure. The line from compression to peak leans away from the vertical, indicating no detonation. The toe of the card is well rounded, indicating proper timing of the exhaust valve. Vertical engines usually do not give such a rounded peak because the combustion space is more compact, and the flame can reach all points in it much more quickly.

Fig. 8 shows two cards superimposed. Both were taken with all adjustments identical, except ignition timing. Both have the same M.i.p., and since the regulating valve adjustment was the same they would show the same power efficiency. Card *B* has the ignition unduly advanced and shows a peak pressure 100 lb. per sq. in. higher than normal card *A*, indicating a corresponding abuse of the engine with no gain in power or efficiency.

Fig. 9 shows two cards, card *A*, normal, and card *B*, with late ignition. The loss of power in this case is shown by the drop in M.i.p. from 83 lb. per sq. in. for *A* to 69.5 lb. per sq. in. for *B*. Bad mixture conditions also are responsible for the decided loss in power.

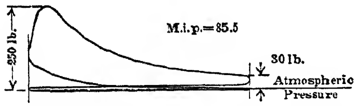


FIG. 7. Perfect Four-cycle Engine Indicator Card

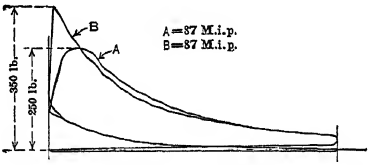


FIG. 8. Effect of Advanced Ignition

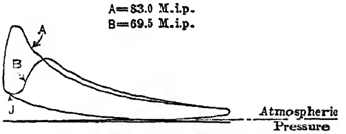


FIG. 9. Effect of Late Ignition

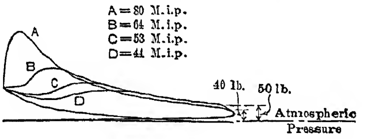


FIG. 10. Effect of Enriching of Mixture

Fig. 10 shows several superimposed cards with all adjustments the same, except mixture. Card *A* is normal, while *B*, *C* and *D*, with their several lowered M.i.p., are the result of too rich a mixture. The engine operation is erratic, and the terminal pressure is 10 lb. per sq. in. higher than normal. High terminal pressures and the corresondingly high temperatures mean abuse of exhaust valves.

Fig. 11 was taken under conditions similar to those of Fig. 10, except that cards *B*, *C* and *D* were taken with mixtures too lean. They also show erratic operation and loss of power, but have lower terminal pressures than normal. Too lean a mixture is a prolific cause of backfires. The mixture burns so slowly that some of it is still burning when the follow-charge starts to enter the cylinder.

Fig. 12 shows the effect of preignition due to an overheated spot in the combustion space. The peak pressure is extremely high, the M.i.p. extremely low. Ignition took place during compression at point *K* instead of near the end of stroke. The upper loop in this card shows negative work. This condition is very detrimental to engines. The effects of early and late ignition are about the same for 2-cycle engines. Mixture conditions are only right in 2-cycle engines for certain ranges in load, since it is necessary to regulate only the gas.

If a 2-cycle engine misses for one stroke, due to the more complete scavenging of the two revolutions, the following power impulse is a very large one, as shown in Fig. 13. *A* is a normal card and *B* was taken following a miss.

Defects in scavenging are very detrimental to 2-cycle operation. Fig. 14 shows the proper shape for a scavenging card for an under-piston scavenging crosshead engine or a trunk-piston engine. The pressure quickly falls while exhaust ports are open from *C* to *D* and the intake starts immediately following the closing of inlet port at *A*. Volumetric efficiency is 89%.

Fig. 15 shows a combination defect of too high an exhaust pressure, as evidenced by the undue distance from end of stroke to point *A*, where suction starts, and too much resistance in the intake pipe or valves as shown by the shaded area below the atmospheric line. The volumetric efficiency is only 67%.

Gas Pressures.—It is extremely important, particularly for 4-cycle engines, where close regulation is required, that the gas pressure be kept at a uniform atmospheric pressure, and checked by a manometer on the gas header. If the gas pressure is different from that of the air a perfect mixture can be had at any one load, but as soon as the load changes the mixture will vary, and the engine becomes erratic. The final regulator should not be required to reduce the pressure from more than 8 oz. per sq. in. If the supply is at a greater pressure an additional high-pressure regulator should be supplied. A tank of

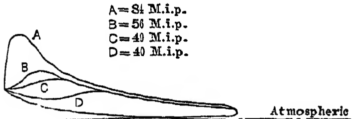


Fig. 11. Effect of Lean Mixture

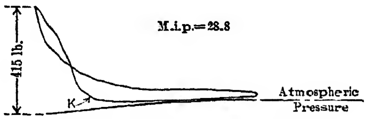


Fig. 12. Effect of Preignition

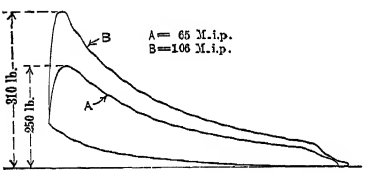


Fig. 13. Effect of Missed Ignition

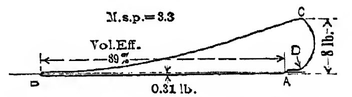


Fig. 14. Proper Shape of Scavenging Card

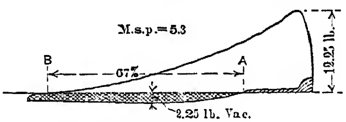


Fig. 15. Effect of Improper Intake

ample volume, say, at least, 5 times the volume of one cylinder should be placed in the fuel line between the final regulator and the engine. For best results there should be a regulator and tank for each engine.

In gasoline plants using 2-cycle engines, where close regulation is not required, it is common practice to run several engines from one fuel header, using one regulator. However, with this arrangement all the engines occasionally "get in step," causing violent pulsations in the header and temporary erratic operation.

EFFECT OF DIFFERENT COMBUSTION CHAMBER SHAPES.—Detonation in a gas engine is caused by too rapid a pressure rise. It can be the result of too early ignition, although the shape of the combustion space and the rate of flame propagation of the mixture have a very decided bearing. The smaller the combustion space, and the more central the spark plug, the quicker the flame started by the plug can reach the most remote part of the combustion space and, therefore, the faster will be the pressure rise. Fig. 16 shows one extreme, the valve-in-head type, with valves in face of the head and the spark plug in the center. This provides the shortest possible path from the plug to the rim of the combustion space. The other extreme is represented by Fig. 17, a horizontal engine with vertical valves. Here the flame has to travel a relatively long distance from the spark plug to the upper edge of the cylinder bore. A design similar to Fig. 16 will produce cards of the form of Fig. 8*B*, while a design like Fig. 17 will produce low peak-pressure cards similar to Fig. 8*A*. In any design the spark plug should be so located as to give the longest possible path for flame propagation without putting it in pockets that would fail to scavenge.

Starting.—Small high-speed gas engines usually are either hand-cranked, or started by means of a battery and starting motor, similar to automotive equipment. The larger engines are started by air, a suitable air compressor and storage capacity being provided. In starting, air at from 150 to 200 lb. per sq. in. pressure is admitted by means of a suitable valve gear, during the power stroke of the engine. When the engine starts to fire, the check valve in the head is held shut by the internal pressure as long as it is higher than the air

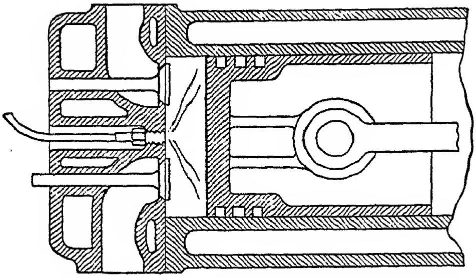


FIG. 16. Compact Combustion Space of Valve-in-head Engine, with Spark Plug in Center

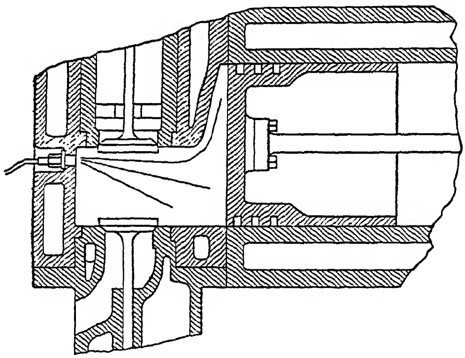


FIG. 17. Attenuated Combustion Space of Horizontal Engine with Vertical Valves

pressure. After the air is shut off this valve remains closed permanently. The air compressors usually are separate from the engine and are driven by either motor, gas or gasoline engine, either belted or direct drive. En-bloc units are available with either one power cylinder and one compressor cylinder or two power cylinders and two compressor cylinders. For larger installations the compressors should be 2-stage.

9. DETAILS

Floor Space.—The figures below give the approximate floor area required for vertical and horizontal gas engines. The horsepower in each case is calculated on a standard basis of 70 lb. per sq. in B.m.e.p., and piston speeds of 900 and 750 ft. per min., respectively, for vertical and horizontal engines.

Vertical engines	4–6 Hp. per sq. ft.
Horizontal engines	0.5–1.0 Hp. per sq. ft.

Air Tank Capacity.—Table 5 gives recommended storage capacities for different classes of engines for single installations, both in proportion to the brake horsepower, and to the engine cylinder displacement on power strokes only.

Where there are many engines in one installation, additional capacity should be added in proportion to the rapidity with which it would be necessary to start engines if they were all down at one time.

Table 5.—Capacity of Air Storage Tanks

Type of Engine	B.Hp. per cu. ft. of Storage Capacity	Cu. ft. of Storage Capacity per cu. ft. of Piston Displacement
Large horizontal, slow-speed double-acting...	13	2.5
Medium size, horizontal single-acting.......	15	4
Medium size, vertical multi-cylinder........	9–12	10–12
Small size, vertical multi-cylinder..........	15	10–12

Exhaust Piping.—The exhaust piping on a 4-cycle engine should be of ample size to prevent undue back pressure, particularly when mufflers are used. This loss of power in a 4-cycle engine is a direct relation to the M.i.p., *viz.*, a 1-lb. per sq. in. back-pressure in an engine operating at 70 lb. per sq. in. M.i.p. shows a loss of $^1/_{70}$ of the indicated horsepower.

Back pressure has much more effect on a 2-cycle engine, as it very seriously affects the volumetric efficiency of the scavenging pump, thereby limiting the rating of the engine and causing it to run hot (see page 12–40, and Fig. 15). The exhaust causes pressure waves which may either help or hinder the scavenging. The frequency of the pressure wave depends on the speed of the engine and the length of the exhaust piping. If possible the length of the exhaust pipe should be varied until the most favorable scavenging conditions are obtained, *viz.*, scavenging to take place while the pressure wave is in the lowest part of its cycle. If mufflers are necessary they should be of large size with very low back-pressure. A large pit makes an excellent exhaust muffler for 2-cycle engines.

Air Filters.—For long life it is just as necessary to filter the air as it is the oil, for most engine locations. Air filters should be of such size that the drop in pressure through them is not over 6 to 8 in. of water. If more than this, it will be difficult to maintain proper mixtures at different loads.

Foundations.—The mass of foundation required varies with the type of engine and the quality of the soil. Small high-speed multi-cylinder engines require little or no foundation, often being mounted on light steel girders or wood sills.

Table 6 gives satisfactory foundations based on both B.Hp. per cubic yard and pounds of engine weight per cubic yard for different classes of larger engines for good soil conditions.

Table 6.—Foundation Volumes for Gas Engines

Type of Engine	B.Hp. per cu. yard	Weight of Engine Per cu. yard of Foundation
Vertical 3-cylinder.....................	8–12	1000–1500 lb.
" 4-cylinder.....................	9–14	1000–1500 "
" 6-and 8-cylinder...............	12–16	1400–2000 "
Horizontal 4-cycle, single or twin, direct-connected compressor engine.........	4–7	800–1300 "
Horizontal 2-cycle, single..............	8–10	1100–1300
Horizontal 2-cycle, twin...............	10–12	1500–1600 "

For bad soil conditions it is good practice to build the foundation with an extended footing at the bottom, extending well outside the main block, thus reducing the bearing pressure on the soil. Where more than one engine is installed this footing should form a solid mat under all of the engines.

GASOLINE ENGINES

The gasoline engine thermodynamically and in construction details is similar to other internal combustion engines. Its most important difference is the addition of a carburetor which volatilizes the liquid gasoline fuel, mixes it with the proper amount of air to form an explosive mixture, which it then delivers to the cylinder. Ignition of the charge is accomplished by an electric spark across the points of a spark plug in the cylinder head. The ignition current is supplied usually from a generator driven by the engine.

Gasoline engines can be operated, with slight alterations, with natural gas, kerosene and other petroleum products as fuel. When operating on natural gas of at least 10,000 B.t.u. heating value, the engine being equipped with high-compression heads, the power developed will be about 15% less than when operating on gasoline.

The effect of altitude on power developed is small up to elevations of 3000 ft. Above 3000 ft. the power developed will decrease about 5% for each 1000 ft. increase in altitude. If the engine is equipped with high-compression heads, the decrease will be about 3% per 1000 ft. increase of altitude. High-compression heads are recommended for high altitudes and also when natural gas is used as fuel.

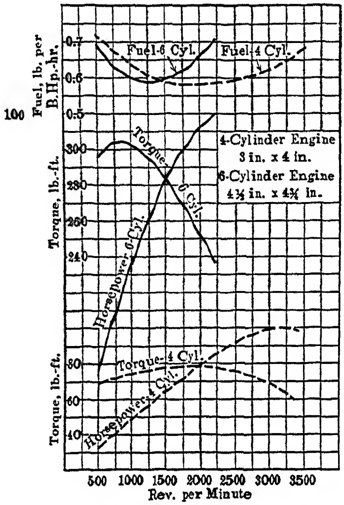


Fig. 1. Typical Performance Curves of Gasoline Engines

Gasoline engines are used mainly for the driving of automobiles, but they also have wide industrial use. Some of the applications of gasoline engines are the driving of agricultural machinery, air compressors, fans and blowers, centrifugal and power pumps, concrete mixers, conveyors, cranes, electric light plants, excavating machinery, tractors, oil field equipment, pile drivers, refrigeration machinery, road building machinery, saw mills, and other wood-working machinery, separators. They also find wide application in the propulsion of small boats.

Fig. 1 shows performance curves of typical 4-cylinder and 6-cylinder gasoline engines. Table 1 gives dimensions and capacities of a range of sizes of gasoline engines made by one manufacturer. The horsepower figures are for stripped engines, corrected to standard temperature and pressure. A deduction of about 15% should be made to determine net horsepower available. This will include an allowance for fan drives and accessories and also variations in temperature and barometric pressure.

For data on proportions and operating characteristics of gasoline engines used in automotive practice see pp. 14-55 to 14-66.

Gasoline engines are built by Allis-Chalmers Mfg. Co., Milwaukee, Wis.; Fairbanks, Morse & Co., Chicago; Hercules Motors Corporations, Canton, O., and others.

Table 1.—Dimensions and Capacities of Gasoline Engines

(Hercules Motors Corp., Canton, O., 1935).

No. of Cyls.	Bore, in.	Stroke, in.	Displacement, cu. in.	Maximum Speed and Power: Intermittent Duty R.p.m.	Intermittent Duty Hp.	Continuous Peak Load R.p.m.	Continuous Peak Load Hp.	Overall Dimensions, in.: Length*	Width	Height
4	2 1/2	3	58.8	2800	19	2500	17.25	25 15/32	17 1/32	21 13/16
4	3	4	113.	2000	29	1800	26.	27 11/32	17 7/8	27
4	3 1/2	4 1/2	173.2	2000	34.5	1800	33.5	34 1/2	17 3/4	30 1/8
4	4	4 1/2	226.2	2000	41	1800	40	34 1/2	17 3/4	30 1/8
4	4	5	251.3	1800	45	1400	40	37 5/16	25 7/8	32 1/2
4	4 1/2	5 3/4	365.8	1600	59	1250	52	42 9/16	26	35 1/4
4	5	5 3/4	451.7	1600	74	1200	65	42 9/16	26	35 1/4
4	5 1/2	7	665.	1150	86	1150	86	54	29 1/8	46 3/8
4	6	7	792.	1100	95	1100	95	54	29 1/8	47 3/8
6	3 3/8	4 1/4	228.	2800	59.5	1800	46.5	40 5/32	18 1/8	31 15/16†
6	4	4 1/2	339.	2400	90	1800	75	42 5/16	25 7/8	34 15/32†
6	4 1/2	4 3/4	453.	2200	98	1800	91	46 5/16	25 7/8	36†
6	4 1/2	5 1/4	501.	2200	106	1800	101	46 5/16	25 7/8	39 1/4†
6	5	6	707.	2000	148	1600	129	55 7/16	29 7/16	44 5/16†
6	5 1/2	6	855.	2000	180	1600	157	55 7/16	29 7/16	44 5/16†

* Exclusive of length necessary for starting crank. † not include carburetor.

Section 13

GAS PRODUCERS

By John Blizard

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GAS PRODUCERS

GAS PRODUCERS*

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1. THEORY OF GAS PRODUCERS

The object of a gas producer is to convert a solid fuel into a combustible gas, by causing a current of air, or air and steam, to traverse a mass of thick incandescent fuel, and the freshly charged fuel above it. Hydrogen, methane, tars and steam are driven off from the freshly-charged fuel, so that the incandescent fuel bed contains the carbonaceous residue and the ash.

THERMO-CHEMICAL REACTIONS BETWEEN OXYGEN, STEAM AND CARBON.—The heat absorbed or given off when any chemical reaction takes place depends solely on the mass, initial and final states of the reacting system, and is independent of the number of stages in which it takes place (Law of constant heat summation).

When 12 lb. of carbon unite with 32 lb. of oxygen at 64° F. to form 44 lb. of CO_2, 14,544 × 12 B.t.u. are liberated. This may be expressed by a thermochemical equation,

$$C + O_2 = CO_2 + 174{,}500 \text{ B.t.u.} \quad [1]$$

where C = atomic weight of carbon, lb.; O_2 and CO_2 = respectively, molecular weight of oxygen and carbon dioxide, lb.; O_2 and CO_2 also represent 380 cu. ft. at 60° F. and 29.92 in. of mercury; or the volume of a 1-lb. molecule of any gas is 380 cu. ft. at 60° F. and 29.92 in.

When 12 lb. of carbon unite with only 16 lb. of oxygen, only 4325 × 12 B.t.u. are liberated, and

$$C + {}^1/_2\, O_2 = CO + 51{,}900 \text{ B.t.u.} \quad [2]$$

Equations [1] and [2] are termed exothermic, because heat is liberated; but when 44 lb. of CO_2 combine with 12 lb. of carbon, 70,700 B.t.u. are used up; this is an endothermic reaction. It is represented by equation [3], obtained by subtracting 2 × equation [2] from equation [1].

$$CO_2 + C = 2CO - 70{,}700 \text{ B.t.u.} \quad [3]$$

Equation [2] represents the best results to be obtained by blowing air up through a thick incandescent bed of carbon; both CO and CO_2 are formed when the air first meets the incandescent carbon. The CO_2 is reduced later in the upper layers as shown in equation [3].

If the CO formed is delivered to the furnace where required, without loss of heat by cooling, no heat will be lost in the process, since the 51,900 B.t.u. liberated in forming the CO will be present in the hot gas; but if the gas be cooled this heat will be wasted and the theoretical producer efficiency will be (174,500 − 51,900) ÷ 174,500 = 70%.

The quantity and composition of the gas formed by oxidizing 12 lb. of carbon with air to CO will be

	Cu. ft. per 12 lb. C	Cu. ft. per lb. C	Percent by Volume
Carbon monoxide, CO	380	32	34.6
Nitrogen, N, (3.78 × 190)	718	60	65.4
Total	1098	92	100.0

where 3.78 is the ratio of N to O by volume in the air, and 380 cu. ft. is the volume of a 1-lb. molecule. The calorific value of the gas formed will be

$$(174{,}500 - 51{,}900) \text{ B.t.u.} \div 1098 \text{ cu. ft.} = 112 \text{ B.t.u. per cu. ft.}$$

To reduce the temperature in the producer, which interferes with its operation, to use some of the heat liberated in burning the carbon for making hydrogen, and to raise the calorific value of the gas, steam is blown through the fuel bed with the air.

* The assistance of Mr. A. K. Bradley, of the Morgan Construction Co., and Mr. J. F. Rogers, of the Wellman Engineering Co., in the revision of this chapter is acknowledged by the editor.

When 4 lb. of hydrogen burn at constant pressure to steam at 64° F., the thermo-chemical reaction is

$$2H_2 + O_2 = 2H_2O + 209{,}500 \text{ B.t.u.} \qquad [4]$$

By subtracting equation [1] from equation [4], equation [5] is obtained. This represents the main reaction between steam and carbon at temperatures between 900° F. to 1100° F.

$$2H_2O + C = CO_2 + 2H_2 - 35{,}000 \text{ B.t.u.} \qquad [5]$$

Above 1100° F., equation [6], obtained from equations [3] and [5], represents the main reaction, known as the water gas reaction;

$$H_2O + C = CO + H_2 - 52{,}850 \text{ B.t.u.} \qquad [6]$$

By adding equation [2] to equation [6], we obtain

$$2C + {}^1/_2\, O_2 + H_2O = 2CO + H_2 - 950 \text{ B.t.u.} \qquad [7]$$

from which it will be seen that comparatively little heat will be required for the reaction to take place; neglecting this small quantity, we have a theoretical efficiency of 100%. The gas formed by the combination of 16 lb. oxygen, 18 lb. steam, and 24 lb. carbon, and mixed with the nitrogen accompanying the oxygen in the air, will consist of

	Cu. ft. per 24 lb. C	Cu. ft. per lb. C	Percent by Volume
Carbon monoxide, CO.	760	32	41.0
Hydrogen, H.........	380	16	20.5
Nitrogen, N..........	718	29	38.5
Total.........	1858	77	100.0

The net calorific value of the gas will be 188 B.t.u. per cu. ft., or of 70% greater calorific value than when no steam is used. But equation [7] is unapproachable in gas producers, as additional heat must be generated to maintain the producer at the required high temperature, less steam per pound of carbon is decomposed and some carbon is burned to CO_2.

Formula [8] represents more closely what actually happens:

$$3C + O_2 + 2H_2O = CO_2 + 2CO + 2H_2 + 68{,}800 \qquad [8]$$

The gas formed will be	Cu. ft. per 36 lb. C	Cu. ft. per lb. C	Percent by Volume
CO_2..................................	380	10.6	11.4
CO..................................	760	21.1	22.8
H..................................	760	21.1	22.8
N..................................	1440	39.9	43.0
Total.	3340	92.7	100.0

The ratio of the net calorific value of the gas formed to the calorific value of the carbon burned is about 87%. The net calorific value of the gas is 137 B.t.u. per cu. ft., and one pound of steam is decomposed per pound of carbon converted into gas.

VOLATILE MATTER FROM COAL.—When coal is heated, gases, tars and steam are driven off. These products react among themselves and with the gases rising from the incandescent fuel bed. Among these gases are hydrogen, methane (CH_4), and ethylene (C_2H_4). The methane and ethylene have a considerably higher calorific value per unit volume than hydrogen or carbon monoxide, and so enrich the gas. The thermo-chemical equations of their combustion at constant pressure to water and CO_2 at 64° F. are

$$CH_4 + 2O_2 = CO_2 + 2H_2O_2 + 382{,}000 \text{ B.t.u.}$$

$$C_2H_4 + 3O_2 = 2CO_2 + 2H_2O + 600{,}000 \text{ B.t.u.}$$

The methane reacts with steam to form either CO or CO_2, and breaks up into C and H_2. Thus

$$CH_4 = C + 2H_2 - 40{,}000 \text{ B.t.u.}$$

$$CH_4 + H_2O = CO + 3H_2 - 92{,}800 \text{ B.t.u.}$$

$$CH_4 + 2H_2O = CO_2 + 4H_2 - 75{,}000 \text{ B.t.u.}$$

CH_4 reacts with CO_2, thus

$$CH_4 + CO_2 = 2CO + 2H_2 - 110{,}700 \text{ B.t.u.}$$

LIMIT OF REACTIONS.—All simple gas reactions are incomplete, though as a rule this can be observed only at high temperatures. Every chemical reaction will continue only until the reacting substances have each attained a definite relative concentration, corresponding to their temperature and total pressure. Thus, CO_2 and CO always will

have the same concentration if left long enough together in the presence of carbon at the same temperature and pressure. When this equilibrium is disturbed by raising the temperature, some CO_2 will react with carbon to form CO with the absorption of heat, which tends to diminish the temperature. In all chemical equilibria, when the temperature increases, the system will change so as to absorb heat and tend to annul the temperature change (principle of Le Chatelier). Thus at high temperatures, since CO_2 combining with C to form CO absorbs heat, less CO_2 and more CO exist at high temperatures when the mixture is in equilibrium.

These equilibria are of great importance in gas producer practice. Messrs. Rhead and Wheeler (*Trans.* Chem. Soc., 1910, 97–2178 and 1911, 99–141) found by experiment that the relative percentages of CO_2 and CO in equilibrium with carbon at one atmosphere pressure were as follows:

Temperature, deg F	1562	1652	1742	1832	2192
CO_2, percent	6.23	2.22	1.32	0.59	0.06
CO, percent	93.77	97.78	98.68	99.41	99.94

Another important reaction in a gas producer is that obtained by subtracting equation [6] from equation [5].

$$CO_2 + H_2 = CO + H_2O - 17{,}850 \text{ B.t.u.} \qquad [9]$$

This reaction also is reversible, and since heat is absorbed, when written as above, the ratio of the product of the concentrations of CO and H_2O to that of the CO_2 and H_2 will be greater as the temperature rises, when the reacting substances are in equilibrium.

The "equilibrium constant" $K = (CO \times H_2O)/(CO_2 \times H_2)$, expressed in terms of the relative concentrations of CO, H_2O, CO_2 and H_2, has been determined by Hahn (*Zeit. für Physikal. Chemie*, 1903, 43, p. 705, and 44, p. 513), as follows:

Temp. deg. F	1447	1627	1807	1987	2201	2561
K	0.81	1.19	1.54	1.95	2.10	2.49

At 1526° F., K is equal to unity; that is, the carbon monoxide and hydrogen are equally strong reducing agents. At higher temperatures, hydrogen, and at lower temperatures, carbon monoxide is the stronger reducing agent (Haber, Technical Thermodynamics of Gas Reactions, p. 145).

EXPERIMENTS ON ACTUAL FUEL BEDS.—Technical Papers 137, 139, U. S. Bureau of Mines, give results of tests by Messrs. Kreisinger, Augustine, Katz and Ovitz on combustion in a fuel bed. The compositions of the gas, and the temperature in different layers are given when burning various coals and coke. They show the oxygen to be practically used up after passing through 4 in. of the fuel bed. Table 1 gives the results of one test when Pittsburgh bituminous coal was burned at the rate of 20 lb. per hour, in a fuel bed 6 in. thick and of 1 sq. ft. area.

It is interesting to note that the sample of gas taken 6 in. from the grate had a gross calorific value of 115 B.t.u. per cu. ft., and that this thin bed of fuel acted as a gas producer with a fairly good efficiency.

Table 1.—Gasification of Pittsburgh Coal

	Distance of sample from grate, in.			
	1.5	3.0	4.5	6.0
CO_2, percent	16.4	13.3	9.8	6.7
Unsaturated hydrocarbons, percent	0.0	0.0	0.1	0.4
O_2, percent	1.3	0.1	0.1	0.2
CO, "	5.2	12.1	17.7	20.5
CH_4, "	0.0	0.0	0.1	2.0
H_2, "	0.2	0.4	1.2	6.9
N_2, "	76.9	74.1	71.0	63.3
Temperature in fuel bed, deg F	2453	2329	1989	1969

2. PERFORMANCE OF GAS PRODUCERS

TESTS OF GAS PRODUCERS.—Tests of Morgan producers at the Youngstown Sheet and Tube Co. in 1930 showed an average rate of gasification of 6108 lb. of coal per hr., equivalent to 77.76 lb. per sq. ft. of producer area per hr. The average thickness of fire was 9 in.; of ash 33 in. The average weight of steam used per lb. of coal was 0.228 lb. at a pressure of 74 lb. per sq. in. Average gas temperature and pressure were 1370° F. and 1.43 in. of water, respectively. Average gas analysis was as follows: CO_2, 5.78%; illuminants, 0.16%; O_2, 0.26%; H_2, 12.85%; CO, 22.19%; CH_4, 3.19%; N, 55.57%. B.t.u. per cu. ft. was, maximum, 152; minimum, 128; average, 139. The proximate

analysis of the coal used was: Volatile matter, 36.30%; fixed carbon, 53.22%; sulphur, 2.10%; ash, 10.48%; moisture, 1.93%.

Table 2 is the log of a test of $61^1/_2$ hr. duration, in regular operation, of a Wellman type L producer, 10 ft. diameter, in a steel plant in the Pittsburgh district. The analysis of the coal was: Moisture, 0.93; C, 76.74; H, 4.66; O, 5.50; N, 1.58; S, 1.97; ash, 9.55; ash fusion temperature, 2190° F.

ESTIMATED HEAT LOSSES.—(Campbell, Manufacture and Properties of Iron and Steel, Ed. 2, 1903, p. 229.)—H. H. Campbell estimated the actual losses from actual practice at Steelton, Pa., as follows:

Lost as carbon in ash	2.1 percent
Sensible heat of dry gas	13.7 "
Sensible heat of steam in gas	0.7 "
Radiation and conduction (by difference)	5.1 "
Total loss	21.6 percent

Dr. C. S. Palmer (*Proc.* Engrs. Soc., West. Penna., vol. xxxiv, p. 338) says that this loss is perhaps about the average.

ECONOMICS OF PRODUCER GAS. (Abstracted from paper by Victor Windett, *Trans.* A.S.M.E., IS-51-8, 1929).—The rate of gasification materially affects the quality of the gas. In the decade 1920–30 advances in producer construction, together with complete mechanical operation, have permitted gasification at rates 8 to 10 times those recorded in experiments in the previous decade, with an improvement in the quality of the gas. See Table 3.

The heating value of producer gas comprises three elements: 1. The heat of combustion or calorific value, as determined by chemical analysis of the gas, amounting to about 80% of the heat. 2. The sensible heat, corresponding to the temperature at which the hot gas enters the furnace, amounting to about 12% of the heat of the coal. 3. The

Table 2.—Test of a Wellman Producer

SCREEN TEST OF COAL

Through Mesh	Over Mesh	Percent, Weight	Through Mesh	Over Mesh	Percent Weight
4 in.	3 in.	0.0	1 in.	3/4 in.	10.5
3 in.	2 in.	17.5	3/4 in.	1/2 in.	7.0
2 in.	1 1/2 in.	11.5	1/2 in.	1/4 in.	9.5
1 1/2 in.	1 1/4 in.	11.5	1/4 in.	1/8 in.	7.0
1 1/4 in.	1 in.	11.5	1/8 in.	....	14.0

Continuous Gas Samples	Gas—Percent by Volume							Net B.t.u.	Gross B.t.u.
	CO_2	Ill.	O_2	H_2	CO	CH_4	N_2		
7.10 P.M. – 11.55 P.M.	5.78	0.69	0.32	14.85	25.59	3.34	52.43	156	168
12.15 A.M. – 6.40 A.M.	5.05	.55	.45	13.80	21.40	3.50	55.25	149	159
7.00 A.M. – 4.50 P.M.	4.90	.40	.25	13.10	23.55	3.80	54.00	154	165
5.10 P.M. – 11.50 P.M.	5.30	.40	.45	14.70	23.15	3.90	52.10	158	169
12.10 A.M. – 7.10 A.M.	4.55	.55	.30	14.00	23.80	4.10	52.70	162	174
7.30 A.M. – 5.00 P.M.	4.55	.45	.45	13.90	23.35	4.40	52.90	162	173
5.20 P.M. – 1.30 A.M.	4.95	.70	.20	14.60	23.00	4.50	52.05	168	180
1.50 A.M. – 8.15 A.M.	4.20	.65	.25	13.70	24.10	4.70	52.40	170	182
Time Average	4.865	.535	.332	14.01	23.165	4.07	53.023	160	171

RATES OF COAL GASIFIED AND FIRE CONDITIONS

No. of Hours	Lb. Coal Charged		Fuel Bed Condition				Gas Temp., deg. F.	Gas Pressure, in. Water	Blast Temp., deg. F.	Steam Pressure, lb. per sq. in.	Lb. Steam per lb. Coal
	Average per hour	Total	Top to coal	Fire zone, center	Ash zone, center	Ash zone, side					
61.5	6,224	382,800	45.1	10.5	11.4	17.6	1,396	0.98	134	74	0.33
24.	6,853	164,483	44.5	11.6	8.8	17.0	1,385	.97	135	77	
10.	7,299	72,990	44.4	10.4	8.2	10.4	1,362	.77	135	80	
1	7,927	7,927	45.0	11.0	10.0	16.3	1,360	.90	134	85	

Table 3.—Improvement in Quality of Producer Gas

	1922	1926	1929 Pittsburgh vein coal		1929 Average coal
CO	22.0%	22.9%	25.00%	27.0%	24.7%
CH_4	2.6 } %		{ 2.27%	3.4%	2.8%
Illuminants	0.4 } %		{ 0.87%	0.8%	0.7%
H_2	10.5%	%	14.49%	12.7%	13.0%
CO_2	5.7%	5.2%	4.69%	4.6%	5.0%
N_2	58.8%	55.2%	52.68%	50.8%	53.8%
Calorific value, gross, at 60° F.	136.0 B.t.u.	135.0 B.t.u.	167.03B.t.u.	176.5 B.t.u.	163.3 B.t.u.
Coal gasified per hr. per sq. ft. of producer area	7.64		Average ... 50.0 lb. Maximum ... 84.6 lb.		

Table 4.—Value of Three Heat Factors in Producer Gas

	Heat of Combustion of Producer Gas		Gross Value	
Carbon monoxide (CO)	25.00%	@ 323 B.t.u.	80.75B.t.u.	
Methane (CH_4)	2.27	1,012	23.07	
Illuminants	0.87	1,853	16.12	
Hydrogen (H_2)	14.49	325	47.09	
Carbon dioxide (CO_2)	4.69			
Nitrogen (N_2)	52.68			
Total	100.00%		167.03B.t.u.	80.2%
Heat of combustion of tar at 0.0006 lb. per cu. ft. gas			15.00	7.2
Total heat of combustion per cu. ft. gas.			182.03B.t.u.	
Sensible heat in gas at 1,400 deg. F.			26.30	12.6
Total available heat in gas per cu. ft.			208.33B.t.u.	100.00%

Table 5.—Heat in Products of Combustion and Cost of Fuel

Based on 20% Excess Air, Preheated

Temperature, deg. F.	1,800	2,000	2,400	2,700	3,000	3,200
Hot Raw Producer Gas from Bituminous Coal						
B.t.u. per cu. ft.	80.5	91.5	113.3	129.94	144.8	155.3
B.t.u. per 1,000 cu. ft.	80,500	91,500	113,300	129,940	144,800	155,300
B.t.u. per lb. coal	5,156.8	5,863.0	7,256.4	8,329.1	9,279.6	10,054.0
Lb. coal per million B.t.u.	193.91	171.56	137.81	120.41	107.76	99.46
Cost coal per million B.t.u.	$0.49	$0.433	$0.348	$0.304	$0.273	$0.252
Cost coal per 1,000 cu. ft. gas	$0.037	$0.045	$0.056	$0.064	$0.071	$0.076
Natural Gas						
B.t.u. per cu. ft.	472.0	523.3	651.8	747.1	836.6	906.2
B.t.u. per 1,000 cu. ft.	471,960	523,300	651,800	747,100	836,600	906,200
Cu. ft. per million B.t.u.	2,120	1,914	1,535	1,340	1,196	1,103
Cost per million B.t.u. at 45c.	$0.955	$0.863	$0.692	$0.604	$0.538	$0.496
Allowable price per M cu. ft.	Average,	$0.228				
Coke-oven Gas						
B.t.u. per cu. ft.	224.0	253.1	315.1	361.5	405.5	440.0
B.t.u. per 1,000 cu. ft.	224,000	253,100	315,100	361,500	405,500	440,000
Cu. ft. per million B.t.u.	4,460	3,956	3,280	2,765	2,470	2,270
Allowable price per M cu. ft.	Average,	$0.109				
Blast-furnace Gas						
B.t.u. per cu. ft.	68.5	77.8	96.7	111.2	122.9	133.0
B.t.u. per 1,000 cu. ft.	68,500	77,800	96,700	111,200	122,900	133,000
Cu. ft. per million B.t.u.	14,600	12,854	10,340	8,993	8,137	7,520
Allowable price per M cu. ft.	Average,	$0.0336				
Fuel Oil						
B.t.u. per lb. of oil	8,259.7	9,381.5	11,669.4	13,251.7	14,941.6	16,168.1
B.t.u. per gal. oil	55,755	62,625	78,768	89,450	100,855	109,135
Gal. per million B.t.u.	17.9	16.0	12.7	11.2	9.9	9.2
Cost per million B.t.u. at 5 [illegible]	$0.94	$0.84	$0.666	$0.588	$0.52	$0.483
Allowable price per gal.	$0.0274	$0.0271	$0.028	$0.0271	$0.0275	$0.0274
Coal						
B.t.u. per lb.	6,541.5	7,438.2	9,248.2	10,569.8	11,782.1	12,765.0
B.t.u. per gal.	62,144	70,663	87,763	100,413	111,930	121,268
Gal. per million B.t.u.	16.1	14.15	11.4	9.97	8.95	8.25
Allowable price per gal.	$0.0304	$0.0304	$0.0304	$0.0304	$0.0304	$0.0304

Table 6.—Cost of Operating Gas Producer Plant

3 Mechanically operated producers, continuous operation; 2 producers shut down 8 hr. each week

Labor	Men per 24-hr. Day	Per Day	Per Month	Total Cost per Month
Foreman at $0.85 per hr.*	0.25	$ 5.10	$158.10	
Gas men at $0.55 per hr.	3	13.20	409.20	
Ash men at $0.50 per hr.	3	12.00	372.00	
Coal handling at $0.65 per hr.	1	5.20	161.20	
Removal of ashes, 330 tons at $0.50			165.00	
Mechanical and electrical up-keep, 200 hr. at $0.75			150.00	
Total labor cost				$1,415.50
Material and Supplies		Per Month		
Producer-hours operated		2,072	....	
Total coal at 2 tons per producer hr.		4,144	....	
Total cost of coal at $4.00 per ton		....	$16,576.00	
Steam used (0.3 lb. per lb. coal) M lb.		2,490		
Cost of steam at $0.40 per M lb.			996.56	
Cooling water, M gal.		670		
Cost of cooling water at 0.10 per M gal.			67.00	
Electric power, kw.-hr.		4,500		
Cost of electric power at 0.015 per kw.-hr.			67.50	
Cost of electric light			30.00	
Lubrication supplies, upkeep			50.00	
Total materials				$17,787.06
Overhead				
Depreciation and Obsolescence			$725.00	
Plant administration			965.00	
Total overhead				1,690.00
Total cost				$20,892.56

Unit Cost	
Cost per ton of coal gasified	$5.041
Cost of gasification only, per ton	1.042
Labor cost per ton of coal gasified	0.342
Material cost per ton of coal gasified	4.292
Cost of 1,000 cu. ft. of gas (130,000 cu. ft. per ton of coal)	0.039
Cost of 1,000,000 B.t.u. calorific value of gas and tar vapor.	0.2881

* Foreman's time prorated from furnace foreman's part time.

calorific value and sensible heat of the tar vapor contained in the gas. Table 4 shows the relative value of each of these factors. The figures are based on routine operation of a large producer plant using Pennsylvania bituminous coal. If the hot raw gas is burned in the furnace practically all of this heat is recovered.

Table 6 gives an analysis of the cost of gas produced in a battery of three mechanically operated gas producers, gasifying 4000 lb. per hr. continuously over a period of one month. The cost per 1,000,000 B.t.u. with a battery of 10 producers, one of which was a stand-by, was practically the same. An analysis of the relative cost of gas, for a steel plant, as produced in 30 small hand-poked producers, and in 6 mechanically operated producers, showed that the cost of gas per ton of ingots was $1.48 with the hand poked producers and $0.98 with the mechanically operated producers.

Tests on some Western coals, of relatively low B.t.u. content, showed a production of 12 gal. of tar per ton of coal, 11.25 gal. being delivered to the furnace, and the balance condensed in the flues. The incandescent tar particles contribute to the luminosity of the gas, and the radiant heat therefrom adds materially to the effectiveness of the gas. Von Helmholtz has established the relative effectiveness of luminous and non-luminous flames as being in the ratio of 1.8 to 1.0.

The relative value of producer gas and other fuels is shown in Table 5. The comparisons are based on mechanically-operated producers, operated at high rates, and include the heat content of the gas, preheating of gas and air, and 20% excess air. Calculations for oil and tar include a 2 1/2% charge for atomization, heating in cold weather, and charge for burners, piping and containers. Producer gas cost is based on a cost of $5.06 per ton of coal gasified, or $4.00 per ton delivered.

If clean gas is required instead of raw hot gas, modifications in producer practice may be necessary. In general, gas temperatures in the producer should not exceed 1200° F.

Table 7 gives the cost of clean gas in one plant. The coal is screened for the producers, the dust being used elsewhere. Tar removed from the gas is burned under boilers. The thermochemical efficiency of the process, based on the original energy of the coal is 72%; by crediting the tar burned under the boilers, it becomes 78%.

Table 7.—Cost of Clean Producer Gas

GAS MAKING, one day of 24 hr.			
Labor .... 3 men @ $5.00		$ 15.00	
Materials			
Coal .... 50.0 tons @ $4.00	$200.00		
Cooling water .... 14.4 M gal. @ 0.03	0.43		
Steam .... 25.0 M lb. @ 0.35	8.75		
Power .... 125.0 kw.-hr. @ 0.02	2.50		
Total		211.68	
Total gas-making manufacturing cost			$226.6
Cost per ton of coal	4.53		
Cost per ton of coal—gasification only	0.53		
GAS CLEANING			
Labor			
Cleaning plant .... 4 men @ $5.40	21.60		
Purifier room .... 1 man	3.00		
Total		$24.60	
Materials			
Cooling water .... 285.6 M gal. @ $0.03	$ 8.57		
Steam—Recirculating pumps .... 12.0 M lb.			
Heating primary water .... 15.5 M lb.			
Heating tar .... 1.5 M lb.			
29.0 M lb. @ $0.35	10.15		
Power exhausters .... 1,625.0 kw.-hr. @ 0.02	32.50		
Purifying gas-coal of 2% sulphur, of which 1.4% is removed by purifier or 1,400 lb. per day. An equal weight of oxide is used = 1,400 lb. oxide	10.00		
Total		61.22	
Total Cleaning Cost			85.82
Cleaning cost per ton of coal	$1.72		
Total			$312.50
TOTAL COSTS			
Total manufacturing cost	$312.50		
Total manufacturing cost per ton of coal gasified	6.25		
Cost of operation exclusive of coal per ton of coal gasified	2.24		

3. FUEL FOR GAS PRODUCERS

Wood, peat, lignite, bituminous coal, anthracite and coke have been used in gas producers. When the gas is to be burned in a nearby furnace, a highly volatile fuel is suitable. If it has to be transported long distances or used in a gas engine, either elaborate scrubbers and tar extractors must be installed or the tars must be destroyed in the producer. For small power plants, anthracite or coke is preferred because the gas, when made in a simple up-draft producer, contains little tar.

A bituminous coal that cakes very little is desirable, since caked coal must be broken up to give the air blast access to the whole fuel bed. On the other hand, it may be advantageous for some of the fine particles of coal to cake together into larger pieces, since they are less likely to restrict the gas passages.

The Wellman Engineering Co., which builds large, mechanically-operated producers for supplying gas for furnaces says: "The most desirable coals for use in the producer are those running from 30% to 40%, volatile, 52% to 65% fixed carbon, with ash content under 15%. When a mechanical feeder is used, the coal should be crushed so as to pass through a 4-in. ring and may contain 35% of fines."

Ash that fuses at a comparatively low temperature will form clinkers, which oppose the passage of air, occupy space in the producer which should be occupied by fuel, and cling to the sides of the producer, reducing its effective area, and sometimes arching over. The fusibility of coal ash has been studied by Messrs. W. A. Selvig and A. C. Fieldner, and the softening temperatures of the ash of the United States coals are found to range from 1900° F. to 3100° F. (See *Coal Age*, Jan. 2, April 17 and June 12, 1919; Jan. 22, Sept. 30 and Oct. 21, 1920. See also p. 4–26.)

Coal having an ash fusion temperature of over 2200° F. is very satisfactory.

4. GENERAL DESIGN AND OPERATION OF GAS PRODUCERS

The essential features of an efficient gas producer are given as: 1. Ability to continuously feed fuel. 2. Maintenance of a uniform fuel and ash level. 3. Maintenance of uniform density of fire. 4. Ability to free the incandescent carbon from ash. 5. Uniform distribution of blast throughout the fuel bed. Other factors to be considered in producer design are: 6. Rate of gasification per square foot of fuel bed. 7. Depth of fuel bed. 8. Method of charging fuel, its rate of descent through the producer, and method of removing ash. 9. Means to control quantity of steam in the blast. 10. Construction and shape of shell.

THE RATE OF GASIFICATION per unit area depends upon the size of fuel, caking properties of fuel, fusibility of ash, size of producer, depth of fuel bed and the steam saturation of the air blast. About 10 lb. of fuel per sq. ft. per hour appears to be the amount allowed in hand poked producers, designed to give off uniform gas over long periods. When the fuel bed of the producer is agitated mechanically, as much as 92 lb. per sq. ft. per hr. may be burned.

THE DEPTH OF FUEL BED depends on the rate of gasification, and will be greater when the rate of gasification is greater. A homogeneous fuel bed 10 in. deep would be sufficient to give a good gas, and small producers have been used successfully with pea size anthracite, with provision for a maximum effective depth of fuel bed of only 18 in. The deep fuel bed requires a stronger blast, and is more difficult to poke, but it enables the ascending gas to give up more of its heat to the descending fuel.

The Wellman Engineering Co. recommends examining the producer fire bed by pushing a rod through it, removing the rod, and observing the temperatures from its color. This company also says that to make a rich gas, the combustion zone should be 6 to 8 in. deep, and below it, the ash zone should never be less than 6 in. nor more than 15 in. above the blast cover; the green coal zone should be from 12 to 18 in. thick. It also points out that if the upper green coal zone is but 6 to 8 in. thick, the gas will leave the producer at 1200° F., and have about 150 B.t.u. per cu. ft., but if maintained 8 to 15 in. deep, the gas will leave at 700° F. and contain 170 to 180 B.t.u. per cubic foot.

C. D. Smith, fuel engineer, in a private communication states: The most suitable thickness depends upon the kind of coal, the rate of burning, and amount of poking. In hand poked producers the fuel bed is invariably thicker than in mechanical producers, due to irregular poking.

CHARGING OF THE FUEL.—The fuel may be charged intermittently into a hopper or hoppers on the producer cover, from which the fuel may be fed to the fire bed, by opening a lower valve, after closing the hopper cover. Or, the fuel may be fed mechanically at rates depending on the required rate of gasification. Preferably, the fuel should be charged as continuously as possible, and distributed uniformly over the fire. In all producers the hoppers are placed with the view of evenly distributing the fuel over the fuel bed. In some mechanically operated producers the fuel bed is fed from a hopper, situated between the central axis and wall, to a rotating fuel bed; in others the cover and hopper are rotated, in order to distribute the falling coal from the hopper evenly over the fuel bed.

DESCENT OF THE FUEL.—Bituminous coal, if properly agitated, will descend through a producer, so as to leave all parts of the fuel bed exposed to the blast. If it is not agitated, the blast will pass through channels, reducing the effective area of the fuel bed, the gas will be poor, the temperature will rise in the gas channels, and troublesome clinkers will form.

Means are provided for poking either by hand or mechanically. With hand poking, the producer, at regular intervals, is carefully searched round the sides for clinkers, which are detached when found; the fuel bed is explored for any channels or hollows which may have formed; the surface of the fuel bed is broken up as it cakes together, and the fuel bed levelled.

The fuel bed of some producers is agitated continuously by mechanical means. This tends to reduce the uneven temperatures, the formation of clinker and the caking of the coal. The shell or base of these mechanical producers may be rotated, the entire depth of the fuel bed may be agitated by pokers, the upper level of the fuel bed may be levelled only, or it may be levelled and agitated for a few inches below the surface to prevent the fuel caking.

The best type of mechanical producer to use depends on the size of coal, its tendency to cake and the fusibility of its ash. The fuel bed of mechanical producers is poked by laborers from time to time, but the actual labor of poking will be less than in producers with no mechanical agitation.

REMOVAL OF ASH.—The ash may be raked through the water lutes or ashpit by hand labor. When a dry ashpit is used, doors are provided for removing the ash. Sometimes the bottom is an inverted cone with a door at the bottom, through which the ash falls on opening it.

When removing ash by hand labor, the intervals between removals are farther apart than when removing ash mechanically, and the fuel bed has to descend further by vigorous poking when the larger quantity of ash is removed. The perfect ash remover would remove the ash as formed. The mechanical ash remover consists usually of a plow which is held against the rotating ash bed, and forces the ashes out through the water lute.

When grates are used, doors are provided through which clinker may be drawn periodically from above the grates.

CONTROL OF STEAM IN BLAST.—The most convenient means of controlling the steam saturation of the air blast lies in observing the saturation temperature; the effect of the steam saturation has been already referred to. The steam for saturation may be supplied from a separate boiler, the exhaust of a steam engine, or it may be generated by waste heat from the gas producer itself, or by the waste heat from a gas engine.

DISTRIBUTION OF AIR BLAST.—The air blast in most producers is admitted through a central tuyere or blast hood. Large producers generally are kept under pressure. Blast is supplied by a constant volume blower, a centrifugal blower or a steam injector blower. Small gas-engine producer plants generally rely on the engine itself to draw air through the fuel bed, and scrubber resistance must be small to permit the gas engine to draw in its full charge.

CONSTRUCTION AND SHAPE.—The producer usually consists of an outside shell of steel plate, open at the bottom if a water lute is used, into which it extends. The top usually is water-cooled, though occasionally it is crowned inside with brick. The shell is lined with good firebrick, which must be well laid. A water jacket sometimes is provided to assist in cooling the brick-work.

Horizontal sections through the producer usually are circular, though sometimes square. The walls are shaped either to keep the sectional area of the fuel bed constant, or to decrease it as fuel descends. Producers have been built with water-cooled walls, and in the Mond producer air and steam blast passes around the shell before entering the producer, being thus preheated.

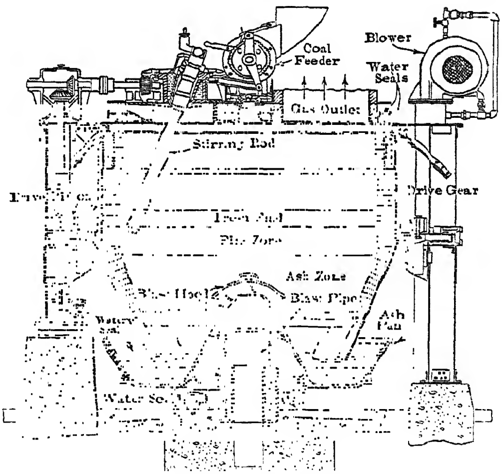

Fig. 1. R. D. Wood Gas Producer

TYPICAL PRODUCERS.—Fig. 1 shows a mechanical producer built by R. D. Wood & Co., Philadelphia. The upper part of the shell is cylindrical. The lower part is conical, terminating in a bosh. Both shell and ash-pan rotate. The feeder is a ratchet-driven drum, with three pockets, rotating in a housing, the escape of gas being prevented by a water-seal. An adjustable deflector plate controls the discharge of coal to the center or wall of the producer. The fuel bed is stirred by a forged steel, water-cooled bar, set at an angle and revolved in a circle of diameter approximately half that of the producer. The shell is rotated by a spur gear meshing into a gear ring on the shell, driven by a motor through a speed reducer. Blast is supplied by a turbo-blower. The blast pipe is in the center of the ash-pan, and terminates in a hood. Ashes are removed by an adjustable steel plow. All joints between stationary and rotating parts, through which gas could escape, are water-sealed. The producer rotates once in every 6 min., and the stirrer rod once every 1.83 min.

The Morgan producer (Morgan Construction Co., Worcester, Mass.) is shown in

Fig. 2. The essential features of this machine are a rotating shell, levelers which continuously level the fuel bed and agitate the fire, continuous coal feeding, and water-cooled walls. A gear ring drives the bottom of the producer, and also a central hub. Three arms on the hub carry the water-cooled shell. The top of the producer is stationary, and is carried by three columns. The top is water-cooled, and supports the continuous feeders and the levelers. Its lower edge extends down into the water-jacket of the shell thus effectually sealing the producer. Ashes are removed by a stationary plow on the rotating bottom. Blast is furnished through a central tuyere by means of a steam ejector, which delivers air through a nest of Venturi tubes underneath a hood. The additional steam necessary for gas making is introduced into the central tuyere through a by-pass. The quantity of steam is readily controlled to coincide with the gas-making requirements. This type of machine has continuously gasified as much as 8000 lb. of coal per hr.

In the Wellman mechanical producer, Fig. 3 (Wellman Engineering Co., Cleveland), the ash pan and shell revolve and the top is stationary. The coal feed comprises two bells so timed that one bell always is closed gas tight. The rate of feed is controlled by a variable speed vane wheel operated by ratchet wheel and adjustable stroke crank. The water-cooled poker is set at an angle, with the point in advance, and swings in an arc from the center to the side wall. This causes a gentle lifting of the semi-plastic coal mass, making it porous for the free exit of the gas and closing holes and pipes. The blast hood is mounted on the ash pan and gives a uniform blast over the entire area of the producer. The ash-pan is stopped three times in each revolution of the shell for approximately 21° total, to loosen up the ash bed. Scrapers and ejectors force the ashes to the outside of the pan where they are removed continuously by the stationary ash plow.

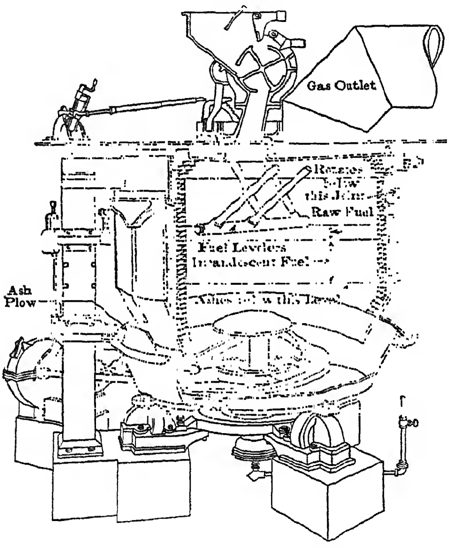

FIG. 2. Morgan Gas Producer

Fig. 4 is a simple suction producer. Coal is charged to the hopper N at the top, thence after closing the cover it is passed through a valve to the fuel bed.

The air enters at the left through A, passes into a saturator B, where it becomes saturated with steam evaporated from water which has been heated by the gas engine. This water enters the scrubber at the top, trickles down over the coke, and passes to the water seal R. The saturated air passes through C to the producer ashpit, up through shaking grates D to the body of the producer E. Gas leaves through F and G to H, where it may be either discharged to the atmosphere when starting, or passed down through K, up through a water scrubber L, and to a moisture trap M. This producer is intended for use with any good, clean Pennsylvania anthracite, with a calorific value of 12,500 B.t.u. per pound or over, less than 15% ash, and less than 1 1/2% sulphur. Either pea or No. 1 Buckwheat sizes may be used.

PRODUCER OPERATION.—The following notes on producer operation are abstracted from the operating instructions issued by R. D. Wood & Co.

Starting.—Ash-pan should be filled level with hard dry ashes free from coal and coke, using only those that will not pass a 1/2-in. screen. If coal-free ash is not available, clinker or firebrick broken to 2-in. cubes, may be used. Freedom from coal and coke is highly important, as combustible in the ash will form clinker, clogging the blast hood. When the ash is in place, porosity of the bed should be ascertained by operating the blower.

Cover ash bed with small dry wood to depth of 18 in., and sprinkle wood with kerosene. Turn water into the water-cooling system, and close fuel entrance from feeder. Light the wood by dropping lighted oil-soaked waste through poke-holes. Gradually increase water supply to cooling

system as heat increases in producer. The wood should be burned as slowly as possible, to heat the brickwork thoroughly and uniformly.

When the wood has burned to a charred mass, purge the producer of smoke, and close poke holes on top plate. Begin feeding coal through the feeder, and start the blower. The pressure at the bottom of the producer should be about 1 1/2 in. of water or less. Exhaust steam from the turbo-blower should be turned into the blast pipe. The fire should not be forced, but burned slowly for several hours to form a heavy coat of tar and soot on the underside of the top plate. When the bed has been built to a gas-making condition, the coal feed should be increased to the amount necessary to produce the desired quantity of gas.

Operation.—Blast pressure should be increased to a point where coal will be burned at the rate it is fed. Moisture in the blast should be kept sufficiently high to prevent formation of clinker. Blast saturation is indicated by the temperature in the blast pipe, which should be from 130° to 150° F. The proper temperature is determined by the amount of coal burned to produce the required quantity of gas. Excess steam will cool the fire, and cause the bed to become tarry. A cool bed causes clinker and hot gas. Excess pressure at the bottom of the producer may cause ignition of gas in the producer. Gas temperature should be between 1,300° and 1,400° F., and bottom pressure about 4 1/2 in. of water when burning 30 tons of coal per 24 hr. Bottom pressure usually must be increased for higher fuel consumption or when the bed becomes packed.

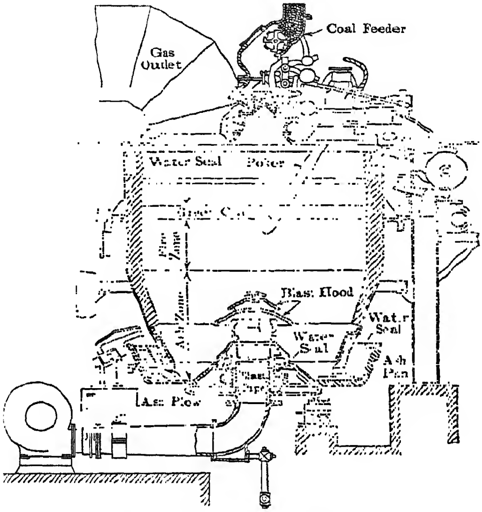

Fig. 3. Wellman Gas Producer

Ash level should be kept at a depth of 21 to 25 in. over the blast hood. Depth of ash should be ascertained every hour by insertion of a measuring rod for a period of 75 sec. in the poke-holes, the holes at the center being used to determine depth over the hood. The heated portion of the rod will show the location of the fire, and the black portion below it the depth of ashes. The producer should be stopped while measurements are being taken. The level of the top of fuel bed is found by a rod inserted in the poke-holes, and resting on top of the coal. The distance below top of producer should be approximately 54 in.

Shutting-down.—To burn out producer, shut off the coal supply and allow feeder to operate until empty. When gas has left producer, insert slide in blast pipe and shut down the blower. Open the poke-holes, open feeder slides and doors on gas exit flue to burn out soot. The producer should continue to rotate until it is time to open the doors. During burning out, the fuel bed should be examined for clinker or fire ring on the walls. Clinker should be broken up with a hand poker, and fire ring cut from the walls.

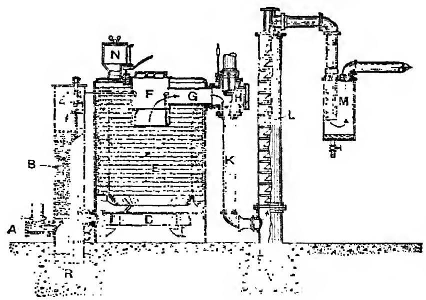

Fig. 4. Suction Gas Producer

CLEANING AND COOLING GAS.—Gas leaving a producer contains tar in the form of vapor, dust and soot produced by the decomposition of hydrocarbons. Weill (Manchester Assoc. Engrs., 1912), referring to up-draft bituminous producers, states that the gas contains 10 to 12 grams of tar per cubic meter, 3 to 5 grams of heavy dust per cubic meter, and 9 to 11 grams of soot per cubic meter. The tar, soot and dust content may remain in the gas if the furnace is near the producer; in fact the tar both adds to the calorific value of the fuel and renders the flame luminous. But if the gas has

to be transported long distances or used in a gas engine, it is cleaned to prevent the deposition of tar, and cooled to raise its calorific value per unit volume.

Gas made from low volatile fuels can be cleaned usually by first passing it up through a simple form of water scrubber, and then through a scrubber filled with sawdust. Special tar extractors are required to remove the tar from gas made from bituminous fuels in up-draft producers.

REMOVAL OF TAR.—There are several methods of removing the tar. A common method is to pass it through a centrifugal cleaner, which really is a badly-designed fan. Weill (Manchester Assoc. Engrs., 1912) gives the following particulars of a cooling and purification plant for a 1000-H.p. producer, gasifying 930 lb. of fuel per hour.

The Mond washer referred to is a water spraying device in which revolves a dasher; the cleaners are the centrifugal cleaners referred to above. Temperatures, deg. F., entering Mond washer, over 752; entering No. 1 cleaner, 120; entering No. 2 cleaner, 90; entering sawdust scrubber, 57; cooling water, 46; atmosphere, 50.

	Grams Tar per cu. meter	Percent Extracted
Leaving producer.........	11.68	0.0
" Mond washer.....	4.30	62.9
" No. 1 cleaner.....	1.55	22.8
" No. 2 cleaner.....	0.30	10.8
" sawdust scrubber..	0.04	2.2

The Smith Tar Extractor. (*Trans.* A.S.M.E., xxxv, p. 837).—The Smith tar extractor consists of a mat of glass wool, through which the gas is forced by a blower under a pressure of several lb. per sq. in. The fall in pressure in passing through the mat is from 2 to 4 lb. per sq. in. The tar particles, in passing through the mat, collect to form larger drops which fall out of the gas into a trap.

Flinn and Dreffein Tar Extractor consists of a tapered plug, heavily threaded, fitted into a female similarly threaded. The plug is mounted upon a vertical shaft, which has a thread the same pitch as the plug. The threaded part of the shaft passes through an outside yoke. Accordingly, the plug may be screwed up or down and remain in mesh with the female thread, but not in contact with it. The width of passage or channel between the threads is varied through changing the position of the plug. The dirty gas enters the tar extractor above the plug and is drawn through by means of the suction produced by the gas exhauster. In the upper chamber of the extractor, above the plug, are water sprays. Water is used to further cool the gas and also to act as a vehicle for maintaining the flow of tar and soot removed from the gas. Only a small volume of water is used for this purpose.

The position of the plug is determined by the drop in pressure necessary to overcome the resistance resulting from the restricted passages, which resistance is indicated by a differential gage. An average drop in pressure of approximately 1 lb. is carried, thereby setting up a gas velocity greater than two miles per minute. At this exceedingly high velocity, the small particles of tar strike against the threads, causing a thin film of liquor, which results in a stream of tar that is drained off from the bottom of the extractors into a settling tank. Wherever there is a thread, the gas must change its direction and the tar particles strike the metal surfaces with great force. At each thread, the gas changes its direction twice, so that in a tar extractor having a plug of ten threads, there are twenty impacts, being more than can be obtained in any mechanical device. This tar extractor can be adjusted for varying rates of gas flow. At light loads, the plug is screwed up to give small width of channel for the gas.

GAS PRODUCERS WITH RECOVERY OF AMMONIA.—About 60 to 75% of the nitrogen of a fuel gasified in a producer will pass off as ammonia, if sufficient steam is used in the blast to prevent the temperature rising. The ammonia formed is brought into contact with sulphuric acid and combines with it to form ammonium sulphate, thus:

$$2NH_3 + H_2SO_4 = (NH_4)_2SO_4,$$

$$34 \text{ lb.} + 98 \text{ lb.} = 132 \text{ lb.}$$

Since 132 lb. of ammonium sulphate contains 28 lb. of nitrogen, the greatest quantity of ammonium sulphate which may be recovered from 2000 lb. of fuel containing 1% nitrogen is 94 lb. The ammonium sulphate formed is in a weak solution, from which the water must be driven off by evaporation. In the Mond ammonia-recovery producer plant about 2 tons of steam are used per ton of coal, one-half of which steam comes from recuperation towers. The air blast is saturated with steam at about 185° F.

Tests by Bone and Wheeler (*Jour.* Iron and Steel Inst., 1907, No. 1, p. 154) on a Mond producer, show the effects of the steam saturation of the blast on the ammonia recovery. The coal contained 1.39% N, which, if all recovered, would correspond to 147 lb. of ammonium sulphate per 2240 lb. of coal; the ton used below is the ton of 2240 lb.

Steam saturation temperature, deg. F.	140	149	158	167	176
1000 cubic feet of gas per ton..................	138	134	141	146	147
Gross B.t.u. per cu. ft. (29.92 in. and 32° F.)......	155	154	146	141	138
Lb. of steam per lb. of coal gasified.............	0.45	0.55	0.80	1.10	1.55
Cu. ft. of air (32° F. and 29.92 in.) per lb. of coal	37	35	37	37	37
Lb. of NH_3 in gas as $(NH_4)_2SO_4$ per ton of coal...	39	45	51	65	72

A large plant, working continuously, is necessary for economical by-product recovery, in order to make the capital investment on the ammonia-recovery plant pay. By-product recovery producer plants are working on coal and peat. With peat, the gases leaving the incandescent zone rise up through the raw peat and dry it, thus permitting peat of a fairly high moisture content to be used.

Tests by Denny and Krubbs (*Engg.*, Jan. 27, 1922) on a Mond producer plant showed the by-product recovery per ton (2240 lb.) of coal with 1.4% N, to be ammonium sulphate 60.4 lb., pitch 35.4 lb., washer dust 100 lb., washer liquor as (NH_3) 0.76 lb. The thermal efficiency, on net coal gas value and gross heat used on plant, was only 42.7%. Mean composition of gas, CO, 10%; CO_2, 14.9%; H_2, 24.2%; N_2, 47.8%; CH_4, 3.1%.

SPACE REQUIRED FOR PRODUCER PLANTS will depend on the rate of gasification per unit area of fuel bed, on the size and number of scrubbers and other factors. See Tables 8 and 9.

Table 8.—Space Required for Anthracite Producer Plant

Rating, 1,000 B.t.u. per hour........	250	500	750	1,000	1,250	1,500	2,000	3,000
Diameter of producer, in............	42	54	60	66	72	78	90	108
Foundation (suction), feet square....	4.5	5.5	6	6.5	7	7.5	8.5	10
Foundation (pressure), width above by	8.5	9.5	10	10.5	11	11.5	12.5	14
Height of producer, ft...............	10	10	10	10	12	12	12	12
Minimum head room, ft.............	19.5	19.5	19.5	20	22	22	22	22
Minimum length of building, inside, ft.	14	14.75	15.25	16.5	16.75	17.75	20.5	22
Minimum width of building, inside, ft.	13.5	14.25	14.75	16	16	17	17.5	19

Producer plants for semi-anthracite would be installed in buildings of about the same size as above for the same output.

Table 9.—Space Required for Mechanically Operated Producer Plants for Bituminous Coal

Rating, 1,000 B.t.u. per hour..................	6,000	20,000 to 25,000	30,000 to 35,000
Size, outside diameter of generator, ft..........	9	14	16
Floor space, approximately, ft................	21 × 33	29 × 46	36 × 55

A modern (1935) mechanical producer, 10 ft. internal diameter requires a building space about 18 ft. square.

5. TESTING GAS PRODUCERS

The usual primary object of a gas-producer test is to determine the ratio of the chemical heat energy of the gas to the energy in the fuel. Other objects may be to determine the calorific value of the gas, its tar and soot contents, the difficulties encountered in operating the producer, the ammonia in the gas, the steam used in the air blast, and the resistance of the fuel bed to the blast. See Test Code for Gas Producers, p. 16–50.

6. PROPERTIES OF PRODUCER GAS

THE CALORIFIC VALUE OF THE GAS may be determined directly by a continuous-flow, constant-pressure gas calorimeter or from the analysis of the gas. Tables 10 to 13, inclusive (prepared by G. W. Jones of the U. S. Bureau of Mines) enable the calorific value to be rapidly determined from the analysis; Table 14 (also prepared by Mr. Jones) enables the cubic feet of gas per pound of carbon in the gas to be determined.

USE OF TABLES. *Example.*—The gross calorific value of a gas containing CO_2, 10.8; C_2H_4, 0.5; CO, 17.7; CH_4, 2.4; H_2, 10.2%, will be $7.9 + 57.2 + 24.2 + 33.1 = 122.4$ B.t.u. per cu. ft. at 60° F. and 29.92 in. of mercury.

The cubic feet of gas at 60° F. and 29.92 in. of mercury per lb. C in the gas is read from the tables by observing the total of the percentage of CO_2, CO, CH_4 and twice the percentage of C_2H_4; this will be $10.8 + (0.5 \times 2) + 17.7 + 2.4 = 31.9$, and the cubic feet of gas per pound of carbon corresponding to 31.9 (Table 14) is 98.7. Supposing the fuel to contain 70% carbon, of which 2% goes to the ash and 1% to tar and soot, the cubic feet of gas per pound of fuel will be $98.7 \times 0.67 = 66.1$ cu. ft. The chemical energy in the gas per pound of fuel will be $66.1 \times 122.4 = 8100$ B.t.u.; so that if the calorific value of the fuel be 10,000 B.t.u. per lb., the efficiency, based on the gross calorific value, will be 81%. To obtain the net calorific value from the gross calorific value, it is necessary to subtract the latent heat of steam, at some arbitrary temperature, formed by burning H_2, CH_4, and C_2H_4, from the gross calorific value. If the latent heat of steam at 60° F.

be chosen (1058.2 B.t.u. per lb.), the net calorific value may be obtained by deducting $0.5H_2\% + CH_4\% + C_2H_4\%$ from the gross calorific value. Thus, the net calorific value of the above gas will be

$$(122.4 - 5.1 - 2.4 - 0.5) = 114.4 \text{ B.t.u. per cu. ft.}$$

THEORETICAL FLAME TEMPERATURE.—Calorific intensity, or theoretical flame temperature, is the highest temperature that can be reached when a gas burns at constant pressure in air so that the sole gases present after combustion are the oxidized gas and nitrogen. It is higher than the maximum temperature in an actual flame, because when gas burns, it does not burn completely, instantaneously, and it loses heat by radiation. The calorific intensity is calculated by dividing the net calorific value of the gas by the mean thermal capacity of the products of combustion over the temperature range.

The calculated values for the same gases vary with the values chosen for the mean

Table 10.—Hydrogen—H_2

Gross calorific value, B.t.u. per cu. ft. at 60° F. and 29.92 in. of mercury

Percent	0.0	0.1	0.2	0.3	0.4	0.5	0.6	0.7	0.8	0.9
5	16.2	16.6	16.9	17.2	17.5	17.9	18.2	18.5	18.8	19.2
6	19.5	19.8	20.1	20.5	20.8	21.1	21.4	21.8	22.1	22.4
7	22.7	23.1	23.4	23.7	24.0	24.4	24.7	25.0	25.3	25.7
8	26.0	26.3	26.6	27.0	27.3	27.6	27.9	28.3	28.6	28.9
9	29.2	29.6	29.9	30.2	30.5	30.8	31.2	31.5	31.8	32.2
10	32.5	32.8	33.1	33.5	33.8	34.1	34.5	34.8	35.1	35.4
11	35.8	36.1	36.4	36.7	37.1	37.4	37.7	38.0	38.4	38.7
12	39.0	39.3	39.7	40.0	40.3	40.6	41.0	41.3	41.6	41.9
13	42.3	42.6	42.9	43.2	43.6	43.9	44.2	44.5	44.9	45.2
14	45.5	45.8	46.2	46.5	46.8	47.2	47.5	47.8	48.1	48.5
15	48.8	49.1	49.4	49.7	50.1	50.4	50.8	51.0	51.4	51.7
16	52.0	52.3	52.7	53.0	53.3	53.6	54.0	54.3	54.6	54.9
17	55.3	55.6	55.9	56.2	56.6	56.9	57.2	57.5	57.9	58.2
18	58.5	58.8	59.2	59.5	59.8	60.1	60.5	60.8	61.1	61.4
19	61.8	62.1	62.4	62.7	63.1	63.4	63.7	64.0	64.4	64.7
20	65.0	65.3	65.6	66.0	66.3	66.6	67.0	67.3	67.6	67.9
21	68.2	68.6	68.9	69.2	69.5	69.9	70.2	70.5	70.8	71.2
22	71.5	71.8	72.1	72.5	72.8	73.1	73.4	73.8	74.1	74.4
23	74.7	75.1	75.4	75.7	76.1	76.4	76.7	77.0	77.4	77.7
24	78.0	78.3	78.7	79.0	79.3	79.6	79.9	80.3	80.6	80.9
25	81.2	81.6	81.9	82.2	82.6	82.9	83.2	83.5	83.8	84.2
26	84.5	84.8	85.1	85.5	85.8	86.1	86.4	86.8	87.1	87.4
27	87.8	88.1	88.4	88.7	89.0	89.4	89.7	90.0	90.3	90.7
28	91.0	91.3	91.6	92.0	92.3	92.6	92.9	93.3	93.6	93.9
29	94.3	94.6	94.9	95.2	95.6	95.9	96.2	96.5	96.8	97.2
30	97.5	97.8	98.1	98.5	98.8	99.1	99.4	99.8	100.1	100.5
31	100.8	101.0	101.4	101.8	102.1	102.4	102.8	103.1	103.4	103.7
32	104.0	104.3	104.7	105.0	105.3	105.7	106.0	106.3	106.6	106.9
33	107.3	107.6	107.9	108.3	108.6	108.9	109.2	109.5	109.9	110.2
34	110.5	110.8	111.2	111.5	111.8	112.2	112.5	112.8	113.1	113.4
35	113.8	114.1	114.4	114.7	115.1	115.4	115.7	116.1	116.4	116.7
36	117.0	117.3	117.7	118.0	118.3	118.7	119.0	119.3	119.6	119.9
37	120.3	120.6	120.9	121.2	121.5	121.8	122.2	122.5	122.8	123.2
38	123.5	123.8	124.1	124.5	124.8	125.1	125.5	125.8	126.1	126.4
39	126.7	127.0	127.3	127.7	128.0	128.3	128.7	129.0	129.4	129.7
40	130.0	130.3	130.7	131.0	131.3	131.7	132.0	132.3	132.7	133.0
41	133.3	133.6	133.9	134.2	134.6	134.9	135.2	135.6	135.9	136.2
42	136.5	136.8	137.2	137.5	137.8	138.1	138.5	138.8	139.1	139.5
43	139.8	140.1	140.4	140.8	141.1	141.4	141.7	142.0	142.3	142.7
44	143.0	143.3	143.6	144.0	144.3	144.7	145.0	145.3	145.6	146.0
45	146.3	146.6	146.9	147.2	147.6	147.9	148.2	148.6	148.9	149.2
46	149.5	149.8	150.2	150.5	150.8	151.2	151.5	151.8	152.2	152.5
47	152.8	153.1	153.4	153.7	154.1	154.4	154.7	155.0	155.4	155.7
48	156.0	156.3	156.7	157.0	157.3	157.7	158.0	158.3	158.6	159.0
49	159.3	159.6	159.9	160.2	160.6	160.9	161.2	161.6	161.9	162.2
50	162.5	162.8	163.2	163.5	163.8	164.2	164.5	164.8	165.2	165.5
51	165.8	166.1	166.4	166.7	167.1	167.4	167.7	168.0	168.4	168.7
52	169.0	169.3	169.7	170.0	170.3	170.6	171.0	171.3	171.6	172.0
53	172.3	172.6	172.9	173.2	173.6	173.9	174.2	174.6	174.9	175.2
54	175.5	175.8	176.2	176.5	176.8	177.2	177.5	177.8	178.1	178.4

Table 11.—Carbon Monoxide—CO

Gross calorific value, B.t.u. per cu. ft. at 60° F. and 29.92 in. of mercury

Percent	0.0	0.1	0.2	0.3	0.4	0.5	0.6	0.7	0.8	0.9
5	16.2	16.5	16.8	17.1	17.5	17.8	18.1	18.4	18.7	19.1
6	19.4	19.7	20.0	20.3	20.7	21.0	21.3	21.6	22.0	22.3
7	22.6	22.9	23.3	23.6	23.9	24.2	24.5	24.9	25.2	25.5
8	25.8	26.2	26.5	26.8	27.1	27.5	27.8	28.1	28.4	28.8
9	29.1	29.4	29.7	30.0	30.3	30.7	31.0	31.3	31.6	32.0
10	32.3	32.6	32.9	33.3	33.6	33.9	34.2	34.6	34.9	35.2
11	35.5	35.8	36.2	36.5	36.8	37.1	37.5	37.8	38.1	38.4
12	38.8	39.1	39.4	39.7	40.0	40.4	40.7	41.0	41.3	41.7
13	42.0	42.3	42.6	43.0	43.3	43.6	43.9	44.2	44.6	44.9
14	45.2	45.5	45.9	46.2	46.5	46.8	47.2	47.5	47.8	48.1
15	48.4	48.8	49.1	49.4	49.7	50.0	50.4	50.7	51.0	51.4
16	51.7	52.0	52.3	52.6	53.0	53.3	53.6	53.9	54.3	54.6
17	54.9	55.2	55.6	55.9	56.2	56.5	56.8	57.2	57.5	57.8
18	58.1	58.4	58.8	59.1	59.4	59.7	60.1	60.4	60.7	61.0
19	61.4	61.7	62.0	62.3	62.6	63.0	63.3	63.6	63.9	64.3
20	64.6	64.9	65.2	65.6	65.9	66.2	66.5	66.8	67.2	67.5
21	67.8	68.1	68.4	68.8	69.1	69.4	69.8	70.1	70.4	70.7
22	71.1	71.4	71.7	72.0	72.3	72.7	73.0	73.3	73.6	73.9
23	74.3	74.6	74.9	75.2	75.6	75.9	76.2	76.5	76.9	77.2
24	77.5	77.9	78.2	78.5	78.8	79.1	79.4	79.8	80.1	80.4
25	80.8	81.1	81.4	81.7	82.0	82.3	82.7	83.0	83.3	83.6
26	84.0	84.3	84.6	84.9	85.2	85.6	85.9	86.2	86.6	86.9
27	87.2	87.5	87.9	88.2	88.5	88.8	89.2	89.5	89.8	90.1
28	90.4	90.8	91.1	91.4	91.7	92.0	92.3	92.7	93.0	93.3
29	93.7	94.0	94.3	94.6	94.9	95.3	95.6	95.9	96.2	96.6
30	96.9	97.2	97.5	97.8	98.2	98.5	98.8	99.1	99.5	99.8
31	100.1	100.5	100.8	101.1	101.4	101.7	102.0	102.4	102.7	103.0
32	103.3	103.7	104.0	104.3	104.6	105.0	105.3	105.6	106.0	106.3
33	106.6	106.9	107.2	107.6	107.9	108.2	108.5	108.9	109.2	109.5
34	109.8	110.2	110.5	110.8	111.1	111.4	111.8	112.1	112.4	112.7
35	113.0	113.3	113.7	114.0	114.3	114.6	115.0	115.3	115.6	115.9
36	116.3	116.6	116.9	117.2	117.6	117.9	118.2	118.5	118.8	119.2
37	119.5	119.8	120.1	120.5	120.8	121.1	121.4	121.7	122.1	122.4
38	122.7	123.1	123.4	123.7	124.0	124.4	124.7	125.0	125.3	125.6
39	126.0	126.3	126.6	126.9	127.2	127.6	127.9	128.2	128.5	128.9
40	129.2	129.5	129.8	130.1	130.5	130.8	131.1	131.5	131.8	132.2
41	132.5	132.8	133.1	133.4	133.7	134.0	134.4	134.7	135.0	135.3
42	135.7	136.0	136.3	136.6	137.0	137.3	137.6	138.0	138.3	138.6
43	138.9	139.2	139.5	139.8	140.2	140.5	140.8	141.1	141.5	141.8
44	142.1	142.5	142.8	143.1	143.4	143.7	144.0	144.4	144.7	145.0
45	145.3	145.7	146.0	146.3	146.7	147.0	147.3	147.6	148.0	148.3
46	148.6	148.9	149.2	149.5	149.8	150.2	150.5	150.8	151.1	151.5
47	151.8	152.1	152.4	152.7	153.1	153.4	153.7	154.1	154.4	154.7
48	155.0	155.3	155.6	156.0	156.3	156.7	157.0	157.3	157.7	158.0
49	158.3	158.6	159.0	159.3	159.6	160.0	160.3	160.6	160.9	161.3
50	161.6	161.9	162.2	162.5	162.8	163.1	163.5	163.8	164.1	164.4
51	164.8	165.1	165.4	165.7	166.1	166.4	166.7	167.0	167.3	167.6
52	168.0	168.3	168.6	168.9	169.3	169.6	170.0	170.3	170.6	170.9
53	171.3	171.6	171.9	172.2	172.5	172.9	173.2	173.5	173.8	174.1
54	174.5	174.8	175.1	175.5	175.8	176.1	176.5	176.8	177.1	177.4

Table 12.—Methane—CH_4

Gross calorific value, B.t.u. per cu. ft. at 60° F. and 29.92 in. of mercury

Percent	0.0	0.1	0.2	0.3	0.4	0.5	0.6	0.7	0.8	0.9
0	0.0	1.0	2.0	3.0	4.0	5.0	6.1	7.1	8.1	9.1
1	10.1	11.1	12.1	13.1	14.1	15.1	16.1	17.1	18.1	19.2
2	20.2	21.2	22.2	23.2	24.2	25.2	26.2	27.2	28.2	29.2
3	30.3	31.3	32.3	33.3	34.3	35.3	36.3	37.3	38.3	39.3
4	40.3	41.3	42.3	43.4	44.4	45.4	46.4	47.4	48.4	49.4
5	50.4	51.4	52.4	53.4	54.4	55.4	56.5	57.5	58.5	59.5
6	60.5	61.5	62.5	63.5	64.5	65.5	66.5	67.5	68.6	69.6
7	70.6	71.6	72.6	73.6	74.6	75.6	76.6	77.6	78.6	79.6
8	80.7	81.7	82.7	83.7	84.7	85.7	86.7	87.7	88.7	89.7
9	90.7	91.7	92.7	93.7	94.8	95.8	96.8	97.8	98.8	99.8

thermal capacity of the gases which over a long temperature range are not known precisely, and various values are used. The calorific intensities calculated below are based on Pier's values for the specific heat of gases, determined by the explosive method. The mean molecular specific heats given in Table 16 are based on Pier's experiments and are in

Table 13.—Ethylene—C_2H_4

Gross calorific value, B.t.u. per cu. ft. at 60° F. and 29.92 in. of mercury

Percent	0.0	0.1	0.2	0.3	0.4	0.5	0.6	0.7	0.8	0.9
0	0.0	1.6	3.2	4.8	6.3	7.9	9.5	11.1	12.8	14.3
1	15.8	17.4	19.0	20.6	22.1	23.7	25.3	26.9	28.5	30.1
2	31.6	33.2	34.8	36.4	38.0	39.6	41.1	42.7	44.3	45.9
3	47.5	49.1	50.7	52.2	53.8	55.4	57.0	58.6	60.2	61.8
4	63.4	64.9	66.5	68.1	69.7	71.3	72.9	74.4	76.0	77.6

Table 14.—Cubic Feet of Gas per Pound of Carbon in Gas

Gas measured at 60° F. and 29.92 in. of mercury

Percent	0.0	0.1	0.2	0.3	0.4	0.5	0.6	0.7	0.8	0.9
10	315	312	309	306	303	300	297	294	292	289
11	286	284	281	279	276	274	271	269	267	265
12	262	260	258	256	254	252	250	248	246	244
13	242	240	238	237	235	233	231	230	228	226
14	225	223	222	220	219	217	216	214	213	211
15	210	209	207	206	204	203	202	200	199	198
16	197	195.5	194.5	193	192	191	189.5	188	187	186
17	185	184	183	182	181	180	179	178	177	176
18	175	174	173	172	171	170	169	168	167	166
19	166	165	164	163	162	162	161	160	159	158
20	157.5	157	156	155	154.5	153.5	153	152	151.5	151
21	150	149.5	148.5	148	147.5	146.5	146	145	144.5	144
22	143	142.5	142	141	140.5	140	139.5	139	138	137.5
23	137	136.5	136	135	134.5	134	133.5	133	132.5	132
24	131	130.5	130	129.5	129	128.5	128	127.5	127	126.5
25	126	125.5	125	124.5	124	123.5	123	122.5	122	121.5
26	121	120.5	120	119.5	119.5	119	118.5	118	117.5	117
27	117	116.5	116	115.5	115	114.5	114	113.5	113.5	113
28	112.5	112	111.5	111.5	111	110.5	110	110	109.5	109
29	108.5	108	108	107.5	107	107	106.5	106	106	105.5
30	105	104.5	104.5	104	103.5	103	103	102.5	102	102
31	101.5	101	101	100.5	100.5	100	99.7	99.3	99.0	98.7
32	98.5	98.2	97.9	97.6	97.3	97.0	96.7	96.4	96.1	95.8
33	95.5	95.2	94.9	94.6	94.3	94.0	93.8	93.5	93.2	92.9
34	92.7	92.4	92.1	91.9	91.6	91.3	91.1	90.8	90.5	90.3
35	90.0	89.8	89.5	89.3	89.0	88.8	88.5	88.3	88.0	87.8
36	87.5	87.3	87.0	86.8	86.5	86.3	86.1	85.8	85.6	85.4
37	85.1	84.9	84.7	84.5	84.2	84.0	83.8	83.6	83.4	83.2
38	82.9	82.7	82.5	82.3	82.1	81.9	81.6	81.4	81.2	81.0
39	80.8	80.6	80.3	80.1	79.9	79.7	79.5	79.3	79.1	78.9
40	78.7	78.5	78.3	78.1	77.9	77.8	77.6	77.4	77.2	77.0
41	76.8	76.6	76.5	76.3	76.1	75.9	75.7	75.5	75.3	75.2
42	75.0	74.8	74.6	74.5	74.3	74.1	73.9	73.7	73.6	73.4
43	73.2	73.1	72.9	72.7	72.5	72.4	72.2	72.1	71.9	71.7
44	71.6	71.4	71.3	71.1	70.9	70.8	70.6	70.5	70.3	70.2
45	70.0	69.8	69.7	69.5	69.4	69.2	69.1	68.9	68.7	68.6
46	68.5	68.3	68.2	68.0	67.9	67.7	67.6	67.5	67.3	67.2
47	67.0	66.9	66.7	66.6	66.4	66.3	66.1	66.0	65.9	65.7
48	65.6	65.5	65.3	65.2	65.1	64.9	64.8	64.7	64.5	64.4
49	64.3	64.1	64.0	63.9	63.7	63.6	63.5	63.4	63.3	63.1
50	63.0	62.9	62.7	62.6	62.5	62.3	62.2	62.1	62.0	61.9
51	61.7	61.6	61.5	61.4	61.3	61.1	61.0	60.9	60.8	60.7
52	60.5	60.4	60.3	60.2	60.1	60.0	59.9	59.8	59.8	59.7
53	59.5	59.3	59.2	59.1	58.9	58.8	58.7	58.6	58.5	58.4
54	58.3	58.2	58.1	58.0	57.9	57.8	57.7	57.6	57.5	57.4
55	57.3	57.2	57.1	57.0	56.9	56.8	56.7	56.6	56.5	56.4
56	56.3	56.2	56.1	56.0	55.9	55.8	55.7	55.6	55.5	55.4
57	55.3	55.2	55.1	55.0	54.9	54.8	54.7	54.6	54.5	54.4
58	54.3	54.2	54.1	54.0	53.9	53.8	53.7	53.6	53.6	53.5
59	53.4	53.3	53.2	53.1	53.0	52.9	52.8	52.7	52.6	52.5

B.t.u. per pound-molecule per deg. F. difference, or in calories per gram-molecule per deg. C. difference. To obtain the specific heat (B.t.u. per lb. per deg. F.), the values given must be divided by the molecular weight of the gas.

Thus, $8.02 \times (4500 - 32)$ = B.t.u. required at constant pressure to raise 28 lb. of CO or N_2 or 32 lb. of O_2 from 32° F. to 4500° F., and $8.02 \div 28 = 0.286$, is the mean specific heat of N_2 or CO over this range.

CALCULATION OF CALORIFIC INTENSITIES.—The net calorific value of a pound-molecule of hydrogen is 104,750 B.t.u. and the mean thermal capacity per degree of the products of combustion in air (1.89 vol. N_2 and 1 vol. H_2O per vol. H_2) over the approximate temperature range = $(1.89 \times 7.9) + 11.34 = 11.34 + 14.93 = 26.27$, and the rise in temperature above 32° F. = $104{,}750 \div 26.27$ = 3985° F.

Similarly for CO, the thermal capacity per degree of 1.89 molecule of N_2 and one molecule CO_2 = $(1.89 \times 8) + 12.65 = 15.12 + 12.65 = 27.77$, and rise in temperature = $122{,}600 \div 27.77$ = 4420° F.

For methane, the theoretical flame temperature = 3720° F., which is lower than that of either H_2 or CO, although the calorific value of methane is about three times that of hydrogen and carbon monoxide.

The actual maximum flame temperature measured by Fery (*Comptes Rendu*, 137, 1903, p. 909) for a free hydrogen flame in air was 3452° F., or about 600° F. less than the calculated flame temperature for hydrogen.

The maximum temperature measured in a Bunsen flame, burning CO, when 29.5 volumes of CO were present in 100 volumes of the air-CO mixture, is given in the *Gas World*, April 10, 1915, p. 413, as 3220° F., or about 1270° F. below the calculated temperature; a maximum temperature of 3330° F. was obtained with 34% CO in the mixture. The maximum speed mixture of CO and air contains 45 to 50% CO, and evidently the higher temperature above the complete combustion mixture (29.5% CO) is due to a smaller, more intense flame with 34% CO.

THEORETICAL FLAME TEMPERATURE OF PRODUCER GAS.—The temperature of a producer gas flame is less than that of either of the component combustible gases, because it contains inert gases.

Tables 17 and 18 and the subjoined calculations show the flame temperatures and other calculations of the combustion of producer gas. From these tables, the maximum rise in temperature is (2940 − 60) = 2880° F., since the sensible heat content of the gas

Table 15.—Properties of Combustible Gases Found in Producer Gas

	Carbon Monoxide	Hydrogen	Methane	Ethylene
Formula....	CO	H_2	CH_4	C_2H_4
Molecular weight.........	28.00	2.016	16.03	28.03
Specific gravity (air = 1).	0.9672	0.0695	0.5545	0.9748
Gross B.t.u. per lb..	4370	61,100	23,800	21,400
Gross B.t.u. per cu. ft. at 60° F. and 29.92 in.	323	325	1,008	1,583
Net B.t.u. per cu. ft........................	323	275	908	1,483
Cu. ft. of air to burn 1 cu. ft. gas..................	2.39	2.39	9.56	14.34
Cu. ft. gases per cu. ft. of gas (burned in air) composed of:	2.89	2.89	10.56	15.34
CO_2..	1.00		1.0	2.00
H_2O..	0.00		2.0	2.00
	1.89		7.56	11.34
Gross B.t.u. per cu. ft. of gas-air mixture............	95.3	95.9	95.5	103.2

Table 16.—Molecular Specific Heats of Gases per Degree at Constant Pressure

(Pier's Values)

t, Deg. F.	B.t.u. per lb.-molecule per deg. F. or Calories per gram-molecule per deg. C. Mean Values 32° F. or 0° C. to t			t, Deg. C.
	N_2, CO, O_2	H_2O	CO_2	
32	6.90	8.07	8.80	0
500	7.02	8.20	9.60	260
1,000	7.14	8.40	10.31	538
1,500	7.27	8.65	10.91	816
2,000	7.39	8.87	11.40	1,094
2,500	7.52	9.27	11.80	1,371
3,000	7.64	9.79	12.10	1,649
3,500	7.77	10.45	12.34	1,927
4,000	7.89	11.31	12.52	2,204
4,500	8.02	12.37	12.66	2,482

formed by burning 100 lb.-molecules of gas for this range is 4,530,000 B.t.u., which is shown in Table 17 to be the net calorific value of 100 lb.-molecules. If the air be raised 1000° F. before mixing with the gas, then, since 100.4 lb.-molecules of air are required to burn 100 lb.-molecules of gas, and the mean specific heat per pound-molecule per deg. F. of the air equals 7.15 B.t.u., the sensible heat of the gas after burning will be greater by 7.15 × 100.4 × 1000, or 718,000 B.t.u., and the total sensible heat after combustion will be 4,530,000 + 718,000, or 5,248,000 B.t.u. This (line 2, Table 18) corresponds to a maximum temperature a little over 3300° F. If the gas be preheated as well as the air, the sensible heat will be increased by 7.48 × 100 × 1000 B.t.u. per 100 lb.-molecules, and the total sensible heat of the gases after combustion will be 5,248,000 + 748,000 = 5,996,000 B.t.u. Since Table 18 shows that 5,860,000 B.t.u., corresponds to a flame temperature of 3660° F., the flame temperature when preheating air and gas is a little above this.

Table 17.—Calorific Values, Volumes of Products of Combustion and Air Required to Burn Constituents of Producer Gas

Composition by Volume		Net Calorific Value (29.92 in. and 60° F.)		Volume of Products of Combustion per 100 Volumes of Gas Burned			Volume Total Products of Combustion	Volumes of Air to Burn 100 Volumes
		B.t.u. per cu. ft. gas	1,000 B.t.u. per 100 lb.-mol. Gas	CO_2	H_2O	N_2		
CO	20	65	2,450	20	..	37.8	57.8	47.8
H_2	10	27	1,050	..	10	18.9	28.9	23.9
CH_4	3	27	1,030	3	6	22.7	31.7	28.7
CO_2	5	..	....	5	..	....	5.0	....
H_2O	2	..	....	..	2	....	2.0	....
N_2	60	..	....	..	..	60.0	60.0	....
Totals	100	119	4,530	28	18	139.4	185.4	100.4

Table 18.—Thermal Capacity of Total Products of Combustion of Producer Gas

Temperature Range	60° F. to 2,940° F.	60° F. to 3,300° F.	60° F. to 3,660° F.
Per 100 lb.-mol. gas burned per deg. F., B.t.u.	1,574	1,602	1,628
Total per 100 lb.-mol., 1,000 B.t.u.	4,530	5,200	5,860
Per cu. ft. gas burned, per deg. F., B.t.u.	0.0414	0.0423	0.0429
Total per cu. ft. gas burned, B.t.u.	119	137	154
Actual mean specific heat (B.t.u. per lb. per deg. F.)	0.289	0.294	0.299

The theoretical flame temperatures of the above producer gas are tabulated as follows: cold gas and air, 2940° F.; cold gas, air heated 1000° F., 3340° F.; gas and air heated 1000° F., 3710° F.

The theoretical flame temperature varies considerably with the composition of the producer gas. Thus a gas richer than the above, containing 2% CO_2, 30% CO, 12% H_2, 3% CH_4, and 53% N_2, requires 1.28 volumes of air to burn one volume of gas, and has a theoretical flame temperature of about 3400° F., or about equal to that of the leaner gas with preheated air.

RADIATION FROM FLAMES.—Helmholtz (Die Licht und Warmestrahlung verbrennender Gase, Berlin 1890) found that in non-luminous flames, hydrogen radiated 3.61% and carbon monoxide 8.74% of the total heat of combustion. This accounts partially for the actual CO flame temperature being so much lower than the calculated flame temperature, and shows that for most heating purposes carbon monoxide is to be preferred to hydrogen, since it radiates twice as much heat per unit volume burned. The presence of tar in a producer gas flame enables it to radiate more heat.

IGNITION TEMPERATURES.—The ignition temperature must be reached at some point in an explosive mixture before the whole mixture will burn. Dixon and Coward (*Jour.* Chemical Society, 1909, 95, p. 514) give the following:

	Ignition range °F. in air
Hydrogen	1076–1094
Carbon monoxide (moist)	1191–1216
Methane	1202–1382

LIMITS OF INFLAMMABILITY.—Gaseous mixtures are inflammable only between two extreme limits, which are given by Thomas (*Engg.*, July 8, 1921, p. 78), as follows:

Hydrogen, 4.2 to 75.0%; carbon monoxide, 12.3 to 75.0%; methane, 5.5 to 13.3%. The figures refer to the proportion of combustible gas in the mixture.

The limits of inflammability of mixtures of combustible gas may be calculated from the proportions of the gases present and the limits of inflammability of the individual gases, but when considerable quantities of inert gases, as CO_2 and N_2 are present in producer gas, it is difficult to estimate the limit of inflammability. Payman (*Chem. Soc. Jour.*, 1919, part II, p. 1461) shows that from 24.7% to 61.6% of producer gas is inflammable in air-gas mixture.

SPEED OF MIXTURES OF COMBUSTIBLE GASES.—The rates of uniform flame speed in a 25 cm. tube over the range of inflammability have been found by Payman (*Chem. Soc. Jour.*, 1919, part II, p. 1461) for various gases. Table 19 shows some of Payman's values:

Table 19.—Rates of Uniform Flame Speed in Various Gases

Coal Gas, percent	Speed, cm. per sec.	Producer Gas, percent	Speed, cm. per sec.	CH_4, percent	Speed, cm. per sec.	$CO + H_2$, percent	Speed, cm. per sec.
7.2	21.5	24.7	20.0	5.80	23.3	9.25	18.2
11.9	87.1	46.0	62.7	7.47	42.0	41.50	309.7
16.8	153.9	49.0	72.2	9.52	66.6	45.92	315.2
20.4	115.6	54.3	69.7	10.64	63.5	51.23	280.0
21.8	74.3	58.8	43.5	12.25	35.0	58.55	178.5
24.3	22.0	61.6	24.0	13.35	19.1	71.34	44.4

The producer gas shown in Table 19, contained by volume, 5% CO_2, 21.3% CO, 12.6% H_2, 3.1% methane and higher paraffins and 58% N_2 (by difference). CH_4 is the principal constituent of natural gas; and CO and H_2 of water gas. The coal gas contained 1.1% benzine and higher olefines, 0.3% CO_2, 2.6% C_2H_4, 9.6% CO, 33.9% methane and higher paraffins, 49.2% H_2 and 3.3% N_2 (by difference). Payman also states that the mixture of air and producer gas with the fastest speed of uniform movement of flame contains slightly more inflammable gases than are required for complete combustion. Usually the mixture with the fastest movement of flame contains more inflammable gas than that corresponding to the mixture in combining proportions. Thus a hydrogen-air mixture in combining proportions contains 29.5% hydrogen, but a mixture containing 40% hydrogen has the fastest speed of uniform movement.

Mason and Wheeler (*Trans.* Chem. Soc., 1920, p. 1238) show the important part that convection currents play in the transmission of flame. They found that a mixture of 6.35% methane in air, when passed along a tube at the rate of 23 cm. per second, had double the flame speed the same mixture had when stationary.

7. DESIGN OF FURNACES FOR PRODUCER GAS

General rules for furnace design cannot be given. The character of the work, the temperatures to be maintained in the furnace, the character of atmosphere (oxidizing or reducing) in the furnace, the method of burning the gas, the method of handling the work in the furnace, and many other variables influence the design. A slight change in one variable may cause a wide variation in operating conditions, and render a given design unsuitable. For a complete discussion of the subject of industrial furnace design, see Trinks, Industrial Furnaces (John Wiley & Sons). See also Industrial Furnaces in Vol. 3 of this series.

In general, producer gas is a suitable fuel only in large installations. The initial cost of gas making equipment makes the cost of producer gas prohibitive when only a few furnaces are to be fired. If, however, a sufficient number of large furnaces or a large number of small furnaces are in use, producer gas will be an economical fuel. The gas may be used raw and hot, that is, as it comes from the producer, or it may be clean and cold. In the first form it must be burned in furnaces as close to the producer as practicable. In the second form it may be distributed to furnaces located at considerable distances from the producer.

Trinks, in Industrial Furnaces, Vol. 2, Chap. vi, compares the relative advantages of the several kind of fuel available for furnaces. The following is a summary of his statements relative to producer gas.

RAW PRODUCER GAS generally is used in large furnaces in which close regulation of furnace temperature and furnace atmosphere is not necessary. Its advantages include a low cost per B.t.u., simplicity of operation, low installation cost, and availability, since gas can be made wherever bituminous coal is available. Also, the gas burns with a lumin-

ous flame. Among the disadvantages are the necessity of locating the furnace close to the producer to avoid condensation of tarry vapors and the loss of sensible heat in long pipes. Further, the mains become choked with soot and tar and require cleaning at regular intervals. Uniformity of composition and quality of gas is difficult of maintenance, and the flow cannot be measured because of the dirt in the gas. Among the operating disadvantages are the necessity of delivering coal to, and removing ashes and clinker from the producer, the maintenance of a steam plant to supply steam to the producer, and the difficulty of maintaining an intelligent labor force in the gas house, because of the unfavorable working conditions.

COLD CLEAN PRODUCER GAS is limited generally to large installations, and to maximum furnace temperatures of about 2000° F., unless regenerative or recuperative furnaces are used. Among the advantages of this fuel are the low temperature of combustion, making it desirable for furnaces where temperatures should not exceed 1600° F., and the efficiency with which it can be burned. Depending on correct furnace operation, accurate temperature and furnace atmosphere control is possible, quality of heated product can be kept uniform, damage to it minimized, and scaling kept low. The gas burns without smoke, even if the atmosphere is reducing, and it can be distributed to a large number of small furnaces. The fuel cost is relatively low. Among the disadvantages are the high initial cost of gas making plant, the necessity of handling coal and ashes, the low heat content of the gas, and the necessity of disposing of the tar made in the producers. Also, the cost of operating labor at the producer is high, and good furnace operation depends largely on the quality of labor in the producer plant.

Section 14

TRANSPORTATION

RAILROAD ENGINEERING

By G. E. Rhoads

AUTOMOTIVE VEHICLES

By Ralph A. Richardson

AERONAUTICS

By Edward P. Warner and S. Paul Johnston

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RAILROAD ENGINEERING

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TRANSPORTATION

RAILROAD ENGINEERING

By G. E. Rhoads

1. TRAIN RESISTANCE

RESISTANCE DUE TO SPEED.—(*Bull.* No. 1001, Am. Locomotive Co., Feb., 1910.) The best data available show resistance to vary from 2.5 to 3 lb. per ton (2000 lb.) for 72-ton freight cars, including weight of car, to 6 to 8 lb. for 20-ton cars. At speeds between 5 to 10 and 30 to 35 mi. per hr. resistance of freight cars is practically constant. Resistance R_E of engine and tender is figured separately. $R_E = F + H + W + G + R$, where F = engine friction = 1.11% of weight on drivers, H = head-on air resistance = $0.002\,AV^2$, W = resistance due to weight on engine trucks and trailing wheels, and to tender, being the same per ton as for cars, G = grade resistance = 20 lb. per ton for each one percent of grade, R = curve resistance = $0.4 + cD$, A = frontal area, taken as 120 sq. ft., V = speed, m.p.h., D = degree of curve. Values of c are:

Wheel base of engine, ft.	5	6	7	8	9	12	13	15	16	20
c =	0.380	.415	.460	.485	.520	.625	.660	.730	.765	.905

If T_E = tractive force of locomotive and T_C = tractive force to overcome resistance of cars, T_B = tractive force of locomotive due to boiler pressure, $T_C = T_E - R_E$. At low speeds a maximum value of $T_E = 0.85\,T_B$. For piston speed over 250 ft. per min., $T_E = 0.85\,T_B \times F$, where F = speed factor given below.

Piston speed, ft. per min...	250	300	350	400	450	500	550	600	650	700	750
Speed factor, F..........	1.00	0.954	.908	.863	.817	.772	.727	.680	.636	.592	.550
Percent of max. Hp.......	60.4	69.1	77.2	83.7	89.0	93.5	96.8	98.7	99.7	100	100
Piston speed, ft. per min...	800	850	900	950	1000	1100	1200	1300	1400	1500	1600
Speed factor, F..........	0.517	.487	.460	.435	.412	.372	.337	.307	.283	.261	.241
Percent of max. Hp.......	100	100	100	100	100	99.0	97.8	96.8	95.7	94.7	93.5

Fig. 1 (Univ. of Illinois *Bull.* 43, 1910) shows the results of tests of resistance of freight

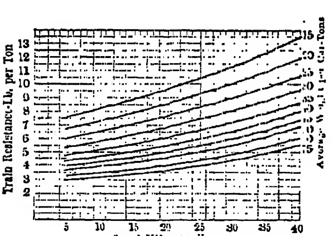


FIG. 1. Freight Train Resistance

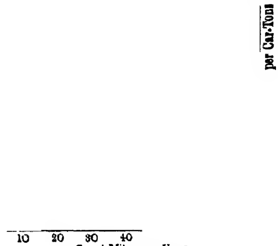


FIG. 2. Passenger Train Resistance

trains of average American box, gondola and tank cars, on straight level track of good construction, with temperatures above 30° F. and wind velocity not more than 20 mi. per hr. Fig. 2 (Univ. of Illinois *Bull.* 110, Dec., 1918) shows the resistance of passenger trains under the same conditions as in the freight train tests. Weight of cars ranged from 30 to 75 tons. The American Locomotive Co. gives (1917) the resistance of freight cars on straight level track, at speeds from 5 to 25 mi. per hr., as

Total weight of car, tons..	20	25	30	40	50	60	70	80
Resistance, lb. per ton....	7.00	5.89	5.13	4.20	3.64	3.27	3.00	2.80

The resistance of an empty freight car weighing 30 tons, carried on 6-wheel trucks, has been found to be 7.27 lb. per ton, and 3.17 lb. per ton for the loaded car weighing 121 tons. Tables 1 and 2 give data on the resistance of both passenger and freight cars.

Table 1.—Resistance of Loaded and Empty Freight Cars

Weight, tons		Resistance, lb. per ton		Weight, tons		Resistance, lb. per ton	
Loaded	Empty	Loaded	Empty	Loaded	Empty	Loaded	Empty
75	21	2.90	5.63	40	14.0	4.40	7.65
70	20.3	3.07	5.82	35	12.6	4.74	8.05
65	19.5	3.24	6.00	30	11.1	5.07	8.45
60	18.6	3.43	6.26	25	9.5	5.44	9.05
55	17.6	3.65	6.50	20	7.8	5.91	9.60
50	16.5	3.90	6.85	15	6.0	6.40	10.3
45	15.3	4.18	7.26				

Table 2.—Resistance of Passenger Cars

Computed from formula $R = 5.4 + 0.002\ (V - 15)^2 + 100/(V + 2)^3$

Speed, mi. per hr. V	Resistance, lb. per ton R	Hp. per ton	Speed, mi. per hr. V	Resistance, lb. per ton R	Hp. per ton	Speed, mi. per hr. V	Resistance, lb. per ton R	Hp. per ton
5	5.89	0.079	30	5.85	0.468	60	9.45	1.52
10	5.51	.147	35	6.20	.579	70	11.45	2.37
15	5.42	.217	40	6.65	.709	80	13.85	2.95
20	5.46	.291	45	7.20	.864	90	16.65	4.00
25	5.60	.374	50	7.85	1.047	100	19.85	5.29

RESISTANCE DUE TO GRADES may be taken as 20 lb. per ton per 1% of grade. The length of grade compared to the length of train should be taken into consideration. (Am. R. R. Assoc., 1916.)

RESISTANCE DUE TO CURVES, R_c ranges for cars, from $0.7c$ to $1.0\ c$, per ton of train on the curve; c = degree of curvature. The lower figure is for large capacity cars, and the higher for low capacity cars. For locomotives R_c may be taken as 1.4. For mine cars, with short wheel-bases, and wheels loose on axles, Baldwin Locomotive Works gives the formula: Resistance, lb. $= 0.20\ lW/r$, where l = wheel-base, ft.; W = weight, lb.; r = radius of curve, ft.

Tests of five freight trains, on track of fair construction laid with 70-lb. rail, to determine the excess of their resistance on curved track over that on straight track are reported in Univ. of Illinois *Bull.* 167, July, 1927. Average gross weights of cars ranged from 15.1 to 48.6 tons. Tests were run in warm weather, and tests at speeds of 10, 20 and 30 mi. per hr. were made with each train on each curve. Results are summarized in Table 3.

Table 3.—Resistance of Curves

Train Numbers	Curve Resistance—Pounds per Ton per Degree					
	On One-degree Curve			On Three-degree Curve		
	10 m.p.h.	20 m.p.h.	30 m.p.h.	10 m.p.h.	20 m.p.h.	30 m.p.h.
1	0.35	0.47	1.00	0.70	0.67	0.61
2	0.54	0.46	0.25	0.57	0.47	0.27
3	0.57	0.50	0.60	0.55	0.45	0.48
4	0.46	0.12	0.52	0.69	0.54	0.29
5	0.66	0.75	0.39	0.70	0.75	0.41
Average (all trains)	0.52	0.46	0.55	0.64	0.58	0.41
Average, disregarding speed		0.51			0.54	

RESISTANCE DUE TO ACCELERATION, R_A, lb. per ton, may be calculated from

$$R_A = 70\ V^2/S = 96.5\ V/t = 70\ (V_2^2 - V_1^2)/S,$$

where V = velocity, mi. per hr.; S = distance, ft.; and t = time, sec., during which acceleration or retardation occurs; V_1 and V_2 = respectively, initial and final velocities, mi. per hr. Am. Ry. M. M. Assoc. (*Proc.*, 1914) formula for correction in drawbar pull for acceleration is $F = 31.1\{1 + (0.6w/W)\} \times \{(V_2 + V_1)(V_2 - V_1)/S\}$, where F = correction factor, lb. per ton; W = weight of car, tons; w = weight of wheels, tons; other notation as above. Constants are based on 33-in. wheels. Positive values of F are subtracted from drawbar pull; negative values are added.

TOTAL RESISTANCE, $R_T =$ $+ R_G + R_C + R_A$ as calculated above. Wm. Elmer (*Trans.* A.S.M.E., 1921) gives the Kiesel train resistance formula as

$$R_T = 100\,N + 1.5W + 0.01\ (V + 16)\ V\sqrt{WN} + (C + 20\ G)W,$$

where R_T = total train resistance lb.; N = number of cars + 3; W = weight of entire

train, tons; V = speed, mi. per hr.; C = curvature of track, deg.; G = grade, percent. Locomotive friction and head-on wind resistance are not considered, and the resistance of the locomotive is taken as equal to that of 3 cars.

RESISTANCE DUE TO FRICTION of locomotive machinery, R_F, lb. per ton, is given by W. F. M. Goss as results of tests at Purdue locomotive testing laboratory as $R_F = 3.8\,d^2\,s/D$, where d and D = diameter of cylinder and drivers, respectively, in.; s = stroke of piston, in. The Am. Ry. Eng. Assoc., 1915, for freight equipment under normal conditions on straight level tangent in warm weather, at 7 to 35 mi. per hr., gives $R_F = 2.2\,T + 121.6\,N$, where T = total weight of cars and contents, tons; N = number of cars. R_F ranges from 4 to 8 lb., depending on whether cars are loaded or empty. In extreme cold, R_F may be as high as 30 lb. with empty cars. American Locomotive Co. (1917) states that R_F is constant at all speeds, and is equal to 25 lb. per ton of weight on drivers.

WIND RESISTANCE OF A TRAIN.—The resistance of air to a plane moving normally to itself is represented by the expression KAV^2, where A = frontal exposed area, sq. ft.; V = speed, mi. per hr.; K = a constant = 0.0033. K includes the suction on the back of the moving body, which is $^1/_3$ the total resistance. Frontal pressure then may be taken as $0.0022\,AV^2$, and Hp. = $(0.0022\,V^3/375)$, where Hp. = horsepower required per sq. ft. of exposed surface. Wind resistance is greatly diminished by streamlining, removing all exposed projecting surfaces, and enclosing all openings between cars and engine. The Union Pacific Ry. (1934) has placed in service a streamlined train, capable of speeds up to 110 mi. per hr. with a total expenditure of 600 Hp. The weight of the train was 85 tons.

WATER SCOOPING RESISTANCE (H. C. Webster, *Ry. Engr.*, Jan., 1918).—If R_s = resistance due to water scooping, lb., a = area of delivery pipe, sq. in., g = 32.2, $R_s = (88.246\,aV)/(144g\sqrt{V^2/2})$. For speeds of 25 to 60 mi. per hr., with a = 50 sq. in., values of R_s are

Speed, mi. per hr.	25	30	35	40	45	50	55	60
Resistance, R_s, lb.	901	1310	1779	2330	2940	3610	4320	5220

At speeds of 40 mi. per hr., the relation of a and R_s is

Area, a, sq. in.	35	40	45	50	55	60	65	70
Resistance, R_s, lb.	1631	1860	2100	2320	2560	2800	3040	3260

ADJUSTED TONNAGE RATING (*Proc.* Am. Ry. M. M. Assoc., 1914).—As the resistance per ton of light cars is greater than that of heavy cars a locomotive can pull more tons of heavy cars at the same speed over the same track. To compile tonnage ratings the cars that a locomotive can haul over a division are determined by test or calculation for both heavy and light cars. Then $C = (W-w)/(n-N)$, where C = car allowance or adjustment factor, W and N = respectively total weight and number of loaded or heavy cars of first train, and w and n = weight and number of empty or light cars of second train. Then $T_L = W + N \times C$, and $T_E = w + n \times C$, where T_L and T_E = respectively adjusted tonnage for loaded and empty cars. In general, C will be lower the steeper the grades; it may be as low as 2 or as high as 80.

2. TRACTIVE FORCE OF LOCOMOTIVES

Notation. T = tractive force, lb.; C, c = respectively, diam. of high-pressure and low-pressure cylinders, in., of compound engines; D = diam. of drivers, in.; D_b = diam. of drivers of booster engines, in.; d, d_1 = respectively, diam. of outside and inside cylinders of simple engines; H = total area of heating surface, sq. ft.; K = evaporation per sq. ft. of heating surface, sq. ft. (assumed = 10 lb.); k, M, m = constants; n = No. of pairs of drivers; P = boiler pressure, lb. per sq. in.; Q = weight on drivers, tons (2000 lb.); R = ratio of m.e.p. in booster cylinder to boiler pressure = 0.80 for 75% cut-off, 0.774 for 70% cut-off, 0.73 for 50% cut-off; r = gear ratio of boosters; S = stroke, in.; s = stroke of inside cylinders of simple engines, in.; V = speed, mi. per hr.; w = wt. of 1 cu. ft. of steam, lb. (taken as 0.39 lb. corresponding to 100° F. superheat).

TRACTIVE FORCE OF SIMPLE LOCOMOTIVE.—The fundamental formula (Kiesel), which takes into account engine friction and head-on wind resistance is

$$\frac{2\,(P -}{[\{110\,wV}$$

The Am. Ry. Eng. and Maint. of Way Assoc. (*Bull.* 112) recommends

$$T = d^2P(S/D)\,\{0.95 - (392\,S/11{,}000\,D)\,V\}.$$

For 3-cylinder simple locomotives ... d practice of the Am. Ry. Assoc. (1926) is $T = mPd^2\,S/D + mPd_1^2\,s/2\,D$. Values of m are given as follows:

Main valve cut-off, maximum, %	90	80	70	60	50
m, without auxiliary ports	0.85	0.80	0.74	0.68	0.60
m, with aux. ports, 80% min. cut-off	....	0.80	0.78	0.77	0.75

locomotives, ... Assoc. (1924) is
max. cut-off;

OF COMPOUND
ocomotive is given b
ds the standard of

$T = (1.7\,Pc^2S)/\{(c^2/C^2) + 1\}\,D$, for 90%

$T = (1.5\,Pc^2S)/\{(c^2/C^2) + 1\}\,D$, for 50% max. cut-off.

V OF MALLET ARTICULATED LOCOMOTIVES.—The Baldwin Works s the formula: $T = 1.7\,Pc^2S/\{(c^2/C^2) + 1\}D$ for varying cylinder a cylinder ratio of 2.4 this becomes $T = 1.2\,Pc^2S/D$. The American Loco- rmula is $T = C^2SMP/D$. Values of M are given in Table 4. This formula i.e.p. equal to 91% of boiler pressure.

Table 4.—Values of Constant M

Percent Cut- High-pres- e Cylinder	Ratio of Low-pressure to High-pressure Cylinder Volume						
	2.2	2.3	2.4	.5	2.6		
90			0.571	0.557	0.542	0.528	0.513
89			.565	.550	.536	.521	.507
88		0.573	.559	.543	.529	.515	.500
87		.567	.552	.537	.523	.509	.494
86	0.575	.560	.546	.531	.517	.502	.489
85	.570	.555	.540	.526	.511	.497	.483
84	.564	.550	.534	.520	.506	.491	
83	.559	.544	.529	.515	.500	.486	
82	.553	.541	.524	.510	.496		
81	.548	.534	.520	.505	.490		
	.543	.531	.515	.500	.486		

Table 5.—Ratio of Tractive Force at Various Speeds to Tractive Force at 10 Miles per Hour

(Merriman)

Stroke, S, in.						Stroke, S, in.					
Drivers, D, in.	[illegible]	[illegible]	[illegible]			Drivers, D, in.	56	62	50	56	56
	0.429	0.453	0.480	0.500	0.536	S/D	0.429	0.453	0.480	0.500	0.536
Mi. per hr.						Mi. per hr.					
10	1.000	1.000	1.000	1.000	1.000	21	0.789	0.775	0.758	0.746	0.723
11	0.981	0.980	0.978	0.977	0.975	22	.770	.755	.736	.723	.698
12	.962	.959	.956	.954	.950	23	.751	.734	.714	.700	.673
13	.942	.939	.934	.931	.925	24	.731	.714	.692	.677	.648
14	.923	.918	.912	.908	.899	25	.712	.694	.671	.654	.623
15	.904	.898	.890	.885	.87	26	.693	.673	.649	.631	.597
16	.885	.877	.868	.862	.849	27	.674	.653	.627	.608	.572
17	.866	.857	.846	.838	.82	28	.655	.632	.605	.584	.547
18	.847	.837	.824	.815	.799	29	.636	.612	.583	.561	.522
19	.82	.816	.802	.792	.77	30	.616	.592	.561	.538	.497
	.808	796	.780	.769	.748						

Table 6.—Revolutions per Minute for Various Diameters of Wheels and Speeds

Diameter of Wheel, in.	Miles per Hour							
	10	20	30	40	50	60	70	80
50	67	134	201	268	336	403	470	538
56	60	120	180	240	300	360	420	480
60	56	112	168	224	280	336	392	448
62	54	108	162	217	271	325	379	433
66	51	102	153	204	255	306	357	408
68	49	99	148	198	247	296	346	395
72	47	93	140	187	233	279	326	373
78	43	86	129	172	215	258	301	344
80	42	84	126	168	210	252	294	336
84	40	80	120	160	200	240	280	320
90	37	75	112	150	186	224	261	299

TRACTIVE FORCE OF LOCOMOTIVE BOOSTER.—Recommended practice of the Am. Ry. Assoc. (1930) gives the formula $T = RPd^2Sr/D_b$.

HORSEPOWER OF A LOCOMOTIVE is determined fundamentally from the formula for horsepower of a steam engine Hp. $= P\,L\,A\,N/33{,}000$ (see p. 7-05). If M = speed of train, mi. per hr., d = diam. of cylinders, in., p = mean effective pressure, lb. per sq. in., S = length of stroke, in., D = diam. of driving wheel, in., then for two cylinders, Hp. $= p\,d^2\,S\,M/375\,D$.

3. TYPES OF STEAM LOCOMOTIVES

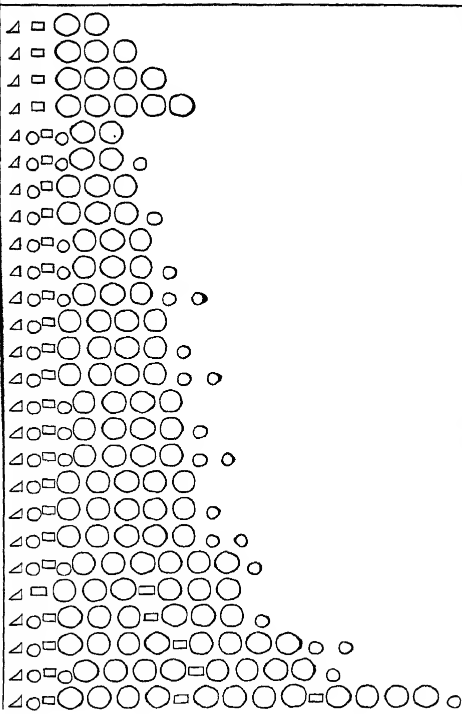
CLASSIFICATION.—Table 7 gives a generally-used classification of steam locomotives, with the names in common use. The type symbol was suggested by F. M. Whyte.

DIMENSIONS OF TYPICAL AMERICAN LOCOMOTIVES in service (1935) are listed in Table 8, together with details of the service in which they are used.

MALLET ARTICULATED LOCOMOTIVES have exceptionally high tractive force, combined with ability to traverse sharp curves. This type was developed by Anatole Mallet and first introduced in Europe in 1889. It is extensively used on American railways, especially in mountain service. The Mallet locomotive has a single boiler, fitted with a mechanical stoker, set over two groups of driving wheels, each with its own frame, cylinders and other equipment. The rear frames are aligned rigidly with the boiler.

(Continued on p. 14-10)

Table 7.—Classification of Steam Locomotives



TYPE SYMBOL	WHEEL ARRANGEMENT	NAME
0-4-0		4-wheel switcher
0-6-0		6- " "
0-8-0		8- " "
0-10-0		10- " "
4-4-0		American
4-4-2		Atlantic
2-6-0		Mogul
2-6-2		Prairie
4-6-0		10-wheel
4-6-2		Pacific
4-6-4		Hudson
2-8-0		Consolidation
2-8-2		Mikado
2-8-4		————
4-8-0		————
4-8-2		Mountain
4-8-4		————
2-10-0		Decapod
2-10-2		Santa Fe
2-10-4		Texas
4-12-2		————
0-6-6-0		Articulated or Mallet
2-6-6-2		" " "
2-8-8-4		Yellowstone
4-8-8-2		Articulated
2-8-8-8-2		Triplex

Table 8.—Dimensions of Typical American Standard Gage Locomotives

Railroad (See Notes, p. 14–09) / Builder	B. & A. Amer.	P. R.R. P. R.R.	N. Y. C. Amer.	C.&N.W. Baldwin	G. N. Baldwin	I. H. B. Amer.	P. R.R. P. R.R.	Wab. Baldwin	C. & O. Lima	U. P. Amer.	N. P. Baldwin
Type	4–6–6	4–6–2	4–6–4	4–8–4	4–8–4	0–8–0	4–8–2	4–8–4	2–10–4	4–12–2	2–8–8–4
Service	Sub.	Pass.	Pass.	P. & F.	Pass.	Switch	P. & F.	Frt.	Frt.	Frt.	Frt.
Year placed in service	1928	1914–27	1927–31	1929	1930	....	1930	1931	1930	1930	1930
Fuel	Bit.	Bit.	Bit.	Bit.	Oil	Bit.	Bit.	Bit.	Bit.	Bit.	Semi-bit.
Tractive force, lb.	41,600	44,460	53,200	76,500	58,300	89,500	64,550	70,750	106,584	96,650	153,400
Weight in working order, lb.	352,000	308,890	352,000	498,000	420,900	294,000	390,000	454,090	566,000	515,000	723,400
Weight on drivers, working order, lb.	180,000	201,830	187,000	288,000	247,300	294,000	271,000	274,100	373,000	372,000	558,900
Weight on truck, leading, lb.	62,000	53,600	65,000	87,000	78,000	....	59,000	78,590	61,000	81,000	48,400
Weight on truck, trailing, lb.	110,000	53,420	100,000	123,000	95,600	....	60,000	101,400	132,000	62,000	116,100
Weight of engine and tender, w.o., lb.	....	480,290	654,700	818,000	747,800	515,300	774,020	755,800	981,000	823,800	1,125,400
Wheel-base, driving, ft. and in.	15–0	13–10	14–0	20–6	20–9	16–4	18–10	18–3	24–4	30–8	44–6
Wheel-base, total, ft. and in.	42–8	36–2	40–4	48–7	47–9	16–4	41–9 1/2	45–0	49–3	52–4	66–8
Wheel-base, engine & tender, ft. & in.	....	72–9	83–7 1/2	91–1	91–2	56–9 3/4	96–6 3/8	86–10	99–5 3/4	91–6 1/2	111–11
Cylinders, diam. and stroke, in.	23 1/2×26	27×28	25×28	27×32	29×29	23 1/2×28 23 1/2×32	27×30	27×32	29×34	Outside 27×32 Inside 27×31	26×32
Valves, kind	Piston	Piston	Piston	Piston	Piston	Piston	Piston	Piston	Piston	Piston Piston	Piston
Valve gear, kind	Baker	Walsch'rt	Baker	Walsch'rt	Walsch'rt	Baker	Walsch'rt	Walsch'rt	Baker	Walsch'rt	Walsch'rt
Valve diam. (piston), in.	14	12	14	14	14	....	12	12	14	14 14	14
Steam lap, in.	1 5/16	1 5/16	1 7/16	1 1/2	1 3/16	....	1 7/16	1 3/8	1 15/16	1 1/4 1 1/16	1 7/8
Exhaust clearance, in.	3/16	3/8	1/4	1/8	1/8	....	7/16	1/8	0	1/8 C.L. 1/8 Lap.	0
Lead in full gear, in.	5/16	9/32	5/16	1/4	3/32	....	9/32	1/4	3/16	1/4 3/16	3/16
Valve travel, in.	9	7	9	7 3/4	7 1/2	....	7	8	9	7 5 7/8	7 1/2
Driving wheel diam., in.	63	80	79	76	80	57	72	70	69	67	63
Truck wheel diam., leading, in.	33	36	36	36	36	....	33	33	33	30	33
Truck wheel diam., trailing, in.	33	50	F.36, R.51	F.44, R.50	45	....	50	42 1/2	F.36, R.43	45	36 and 42
Driving axle journals, main, in.	11×12	11×15	11 1/2×14	13 1/2×14	13×14	....	12×16	13×14	13 1/2×14	12×13	12 1/2×14
Driving axle journals, others, in.	10×12	11×15	11×13	12×14 1/2	11 1/2×14	....	11×16	11 1/2×14	12×14	10×3	11 1/2×14
Truck axle journals, leading, in.	7×12	6 1/2×12	7×12	7 1/2×14	7 1/2×14	....	6 1/2×12	6×11	7×14	7 1/2×13	7×14
Truck axle journals, trailing, in.	6×11	6 1/2×12	6 1/2×12 F 9×14 R	9×14	9×14	....	6 1/2×12	9×14	7 & 9×14	9×14	7 & 9×14
Boiler, kind	Str. Top.	Belpaire	Con. Conn.	Conical	Conical	Conical	Belpaire	Conical	Conical	Conical	Str. Top.
Working pressure, lb. per sq. in.	215	205	225	250	225	200	250	250	260	220	250
Boiler diam., outside first ring, in.	82 7/16	82 1/2	82 7/16	90 3/16	84	86	84 1/2	86 1/2	99 3/4	90	103 1/4 *
Firebox, width and length, in.	86×102	80×126	90×130	96×150	102×138	102×102	80×126	96×144	108×162	108×185	115×266
Combustion chamber, length, in.	None	....	None	60	60	....	98	60	66	80 1/2	72 1/2
Tubes, number and outside diam., in.	234–2	236–2 1/4	37–2 1/4 19–3 1/2	51–2	38–2 1/4	315–2	120–2 1/4	49–2 1/4	59–2 1/4	40–3 1/2	92–2 1/4
Flues, number and outside diam., in.	48–5 3/8	40–5 1/2	182–3 1/2	214–3 1/2	195–3 1/2	50–5 1/2	170–3 1/2	214–3 1/2	275–3 1/2	222–3 1/2	280–3 1/2
Tubes and flues, length, ft. and in.	13–6	19–0	20–6	21–0	22–0	16–0	19–0	21–0	21–0	22–0	22–0

Heating surface, firebox, sq. ft.		213	315	281	558	379	248	395	495	645	591	866
Heating surface, tubes & flues, sq. ft.		2548	3731	4203	4656	4402	3769	4303	4694	5990	5262	6,800
Heating surf. total evaporative		2761	4046	4484	5214	4781	4017	4698	5189	6635	5853	7,666
Heating surf. superheating		788	1147	1951	2357	2265	953	2052	2360	3030	2560	3,224
Heating surf. combined super. & evap.		3549	5193	6435	7571	7046	4970	6750	7549	9665	8413	10,890
Grate surface, sq. ft.		61	70	82	100	98	73	70	96	122	108	182
Booster		None	None	Trailer	Trailer	None	Tender	None	None	Trailer	None	Trailer
Tender, frame		....	....	Cast steel	....	....	....	Cast steel	....	....	....	....
Tender, weight loaded, lb.		....	171,400	302,700	320,000	326,900	221,300	384,020	301,710	415,000	308,800	302,000
Tender, wheel diam., in.		....	36	36	36	33	....	33	36	36	33	37
Tender, journals, in.		....	5 1/2×10	6×11	Roll. bear.	6 1/2×12	....	6 1/2×12	6×11	7×14	6 1/2×12	7×14
Tender, water capacity, gal.		5000	7850	14,000	18,000	17,000	12,000	22,090	15,000	23,500	18,000	21,200
Tender, fuel capacity, tons or gal.		6 T.	13 1/2 T.	28 T.	20 T.	5800 G.	15 T.	36 1/2 T.	18 T.	30 T.	20 T.	27 T.
See notation below	$W_d/T.F.$ =	4.3	4.5	4.4	4.4	4.2	3.9	4.2	3.9	4.1	3.8	4.0
	$T.F./H.S._E$ =	15.1	11.0	9.4	12.5	12.2	18.9	13.7	13.7	13.8	16.5	18.3
	$H.S._E/G$ =	45.4	57.7	54.7	52.1	48.8	55.0	67.0	54.0	54.4	54.3	42.1
	$T.F. \times D/H.S._E$ =	949	879	745	950	976	1074	989	954	952	1106	1150
	$W_t/H.S._E$ =	127.5	76.3	78.5	95.5	88.0	73.2	83.0	87.5	85.3	88.0	94.4
	$H.S._S/H.S._E$ =	0.285	0.283	0.434	0.452	0.473	0.237	0.437	0.455	0.456	0.437	0.421

* Inside.

NOTATION.—W_d = weight on drivers, lb.; W_t = total weight of locomotive, lb.; $T.F.$ = tractive force, lb.; $H.S._E$ = evaporative heating surface, sq. ft.; $H.S._S$ = superheater heating surface, sq. ft.; G = grate area, sq. ft.; D = diam. of drivers, in.

1. *Boston and Albany* (B. & A.).—A suburban type operating out of Boston, Mass.

2. *Pennsylvania Railroad* (P. R. R.).—Over 400 of these locomotives are used to haul the "Broadway" 16 1/2-hr. trains between New York and Chicago, the "American" 23-hr. train between New York and St. Louis, the "Congressional" between New York and Washington, and main-line expresses.

3. *New York Central* (N. Y. C.).—About 200 of these engines are used on the "20th Century Limited" 16 1/2-hr. train between New York and Chicago, the "Southwestern Limited" 23-hr. train between New York and St. Louis, the "Empire State Express" 8 1/2-hr. train between New York and Buffalo, and other main-line expresses.

4. *Chicago and Northwestern* (C. & N. W.).—35 of these locomotives haul the "San Francisco Limited" out of Chicago and also fast freights between Chicago and Omaha.

5. *Great Northern Railway* (G. N. R.).—Used to haul the "Empire Builder" and the "Oriental Limited."

6. *Indiana Harbor Belt* (I. H. B.).—A typical switching engine in hump yard and transfer service between trunk lines.

7. *Pennsylvania R. R.* (P. R. R.).—A mountain-type engine for heavy passenger trains and scheduled fast freights with running speeds approaching those of passenger service. About 100 are in use, with 200 more quite similar.

8. *Wabash R. R.* (Wab.).—Used to haul fast freights between St. Louis and Buffalo.

9. *Chesapeake and Ohio* (C. & O.).—Said to be most powerful 2-cylinder locomotives ever built. Forty are used to haul coal trains of 12,250 tons over 237 miles of line between Russel, Ky., and Toledo, O., with maximum grades of 0.7%, maximum curves of 7°.

10. *Union Pacific* (U. P.).—A 3-cylinder freight locomotive. Eighty have been built.

11. *Northern Pacific* (N. P.).—Designed to use a semi-bituminous (Rosebud) coal of low heating value; are said to have the largest boiler ever used on a locomotive. They weigh, with tender, over 1,000,000 lb., and can develop over 6000 Hp. Twelve are in use.

The front frames are hinged to the rear frames by a pin on the center line of the locomotive, and support the front end of the boiler on sliding bearings, or waist bearers. The lower section of the waist bearer is a lubricated brass plate, on which the upper section slides, and which takes the wear. Clamps on the front waist bearer prevent the frames from dropping in case of derailment. On entering a curve the front frames swing on the hinge pin as center. Controlling springs on the front boiler bearing tend to hold the front and rear frames in line, and assist the rear unit to follow into a curve.

The cylinders are compounded. High-pressure cylinders drive the rear set of wheels, and take steam direct from the boiler. They exhaust to the low-pressure front cylinders, whose exhaust is up the stack. Cylinder ratios range from 2.35 to 2.50. A ball joint, located on the hinge pin center, is fitted to the back end of the receiver pipe from high- to low-pressure cylinders, to permit it to swing with the frames. The exhaust pipe from low-pressure cylinders to stack also has flexible joints. Both pipes have packed glands and slip joints for expansion and contraction.

Superheated steam is used almost universally in Mallet locomotives, the superheater being arranged as in a single expansion locomotive. Reheaters between high- and low-pressure cylinders are unnecessary. The high-pressure steam pipes extend back from the superheater in the smoke box to the high-pressure cylinders, where steam distribution is by piston valves. Either piston or balanced slide-valves, the latter preferably double ported, may be used on low-pressure cylinders. Lubrication is satisfactory if one lubricator connection is run to each high-pressure cylinder and one to the receiver pipe. The two sets of valve motion are controlled simultaneously by a power reverse mechanism, usually operated by compressed air.

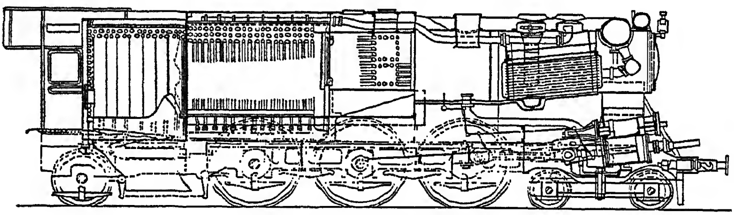


Fig. 3. Loeffler Locomotive

BALANCED COMPOUND LOCOMOTIVES have two high-pressure cylinders between the frames and two low-pressure cylinders outside. The inside cranks are 90° apart, and the outside crank pins are 180° from the inside crank on the same side. High-pressure steam can be admitted to the low-pressure cylinders for starting.

LIMITED CUT-OFF LOCOMOTIVES (*Engg.* Oct. 22, 29, 1920). The Penna. R. R. in 1920 replaced with 2–10–0 locomotives, a large number of engines that, in the service to which they were assigned, were required to work with little or no expansion of steam. In the new engines, the cylinders were enlarged and the cut-off arranged to be maximum at 50% of stroke. At this cut-off the engine gives a turning moment diagram similar to that of the earlier engines in full gear. The engines have piston valves 12 in. diam., operated by Walschaerts gear, and have inside admission. Steam lap is 2 in.; lead is $^3/_{16}$ in.; exhaust lap is $^1/_8$ in. To obtain sufficient starting effort, auxiliary ports $^1/_8 \times 1\,^1/_2$ in. admit steam to the cylinders before the main ports are uncovered. Cylinders are $30\,^1/_2 \times 32$ in., and the boiler pressure is 250 lb. per sq. in. The locomotives weigh 371,800 lb. in working order, have a tractive force of 90,000 lb., and a drawbar pull of 76,200 lb. at 7.4 mi. per hr. At 14.5 mi. per hr., in full gear (50% cut-off) drawbar pull is 65,000 lb. They develop 1 Hp. on 3 lb. of coal, and use 17.3 lb. of steam per I.Hp.-hr.

HIGH-PRESSURE LOCOMOTIVES.—A committee of the Am. Ry. Assoc., 1932, gives the advantages of high steam pressure as: The energy that can be stored in a given quantity of steam is increased by increasing the pressure and temperature, hence requiring less water to develop a given amount of power; the increased power is obtained by greater expansion of the steam in the cylinders; engines of greater capacity can be designed for the same weight, and more efficient use of the steam is had by use of smaller cylinders, valve chambers, and steam pipes, with reduced area for loss of heat by radiation. Disadvantages include: difficulty with cylinder and valve lubrication; increasing maintenance costs; piston valves probably must be replaced by a poppet valve which does not require lubrication; stronger pipes, globe valves and cab fittings are necessary.

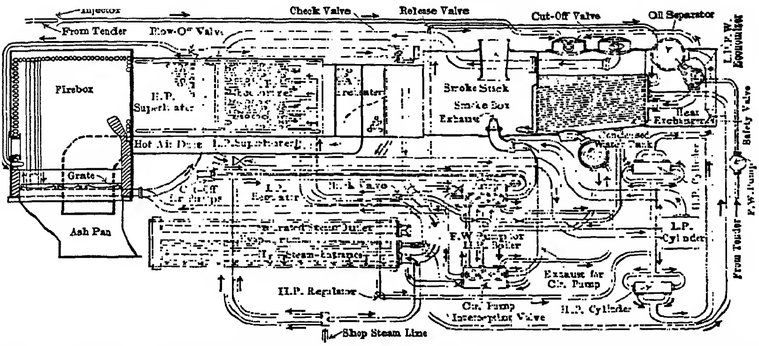


Fig. 4. Diagram of Loeffler Principle of Steam Generation Applied to German State Ry. 4-6-2 Locomotive

The German State Railways placed in service, in 1927, high-pressure locomotives of the Schmidt-Henschel and the Schwartzkopf-Loeffler types. The Schmidt 4–6–0 locomotive has a water-tube firebox with a fire-tube boiler barrel. A pressure between 1300 and 1700 lb. per sq. in. is developed in a closed tube system, filled with distilled water, which transfers heat from the firebox to water in a drum carrying 850 lb. per sq. in. steam pressure. The fire-tube part of the boiler carries a pressure of 200 lb. per sq. in. Feedwater enters the steam space of the low-pressure boiler, and is pumped into the high-pressure boiler. Steam at 850 lb. is used in an inside high-pressure cylinder. The exhaust, mixed with 200 lb. steam, is used in two outside low-pressure cylinders. Cylinder diameters are: high-pressure 11 7/16 in.; low-pressure 19 5/8. Stroke of all cylinders is 23 3/4 in. Driving wheel diameter is 79 in.

The Loeffler 4–6–2 locomotive, Figs. 3 and 4, is a 3-cylinder compound, with a rated tractive force of 37,000 lb. Its steam generator is a cylindrical drum, partly filled with water from which steam is forced at high velocity by two pumps through a grating of steel tubes forming the firebox. The steam is superheated in the firebox tubes. From the firebox tubes, a part of the superheated steam, at 1470 to 1760 lb. per sq. in. pressure and 840° F. goes to the two high-pressure outside cylinders, but the greater part returns to the drum. It is discharged into the water in the bottom of the drum, and gives up heat absorbed from the firebox. The exhaust from the high-pressure cylinders, at a pressure of about 265 lb. per sq. in., flows through an oil separator, and then into the tubes of a heat exchanger where it generates steam at 214 lb. per sq. in. pressure. This steam,

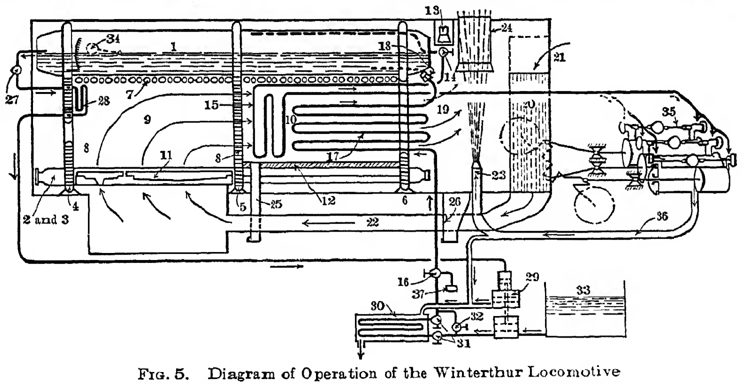


Fig. 5. Diagram of Operation of the Winterthur Locomotive

1, top boiler drum; 2 and 3, bottom boiler drums; 4, back wall of firebox; 5, front wall of firebox; 6, front wall of boiler; 7, tube elements; 8, stay-tubes; 9, firebox; 10, superheater and preheater chamber; 11, firegrate; 12, refractory floor; 13, safety valve; 14, throttle; 15, superheater; 16, check valve; 17, feedwater preheater; 18, check valve; 19, smokebox; 20, air preheater; 21, intake openings of air preheater; 22, air duct; 23, blast pipe; 24, stack; 25 and 26, cleaning shafts; 27, stop valve for auxiliary steam supply; 28, superheater for feed pump; 29, feed pump; 30, exhaust steam preheater; 31, stop valve (normally open); 32, stop valve (normally closed); 33, water tank; 34, water-level float; 35, steam engine; 36, exhaust pipe; 37, filling connection

Table 9.—Data of Condensing Turbo-Locomotives

Year	Builder and Railroad	Rated Tractive Force, lb.	Wheel Arrangement		Weight in Working Order, lb.			Turbine			Steam Pressure, lb. per sq. in.	Condenser		Rated Hp.	Max. Speed, m.p.h.	Weight Efficiency Engine and Tender, lb. per Hp.
			Engine	Tender	Total Engine	Total Tender	On Drivers	Design	Type	Stages		Type	Cooling Surface, sq. ft.			
1921–22	Aktiebolaget Ljungstrom Angturbin, Stockholm. Swedish State Rys.	30,240	4–6	6–2 (Driver)	138,880	143,360	107,520	Ljungstrom	Axial-flow Impulse-Reaction	2–15	285	Air-cooled	10,764	1800	68.3	156.7
1921–22	Swiss Loco. Wks. & Escher, Wyss & Co. Swiss State Rys.	25,000	4–6 (Driver)	4–4	130,000	90,000 Approx.	91,200	Zoelly	Impulse	6	200	Water-cooled	......	1000	47	220
1922	Sir W. G. Armstrong, Whitworth & Co., Scottswood, Eng. L. M. & S. Ry.	22,000	2–6	6–2 (Driver)	293,000		90,000 Approx.	Ramsay	Impulse	9	200	Water-film evap.	......	1100	60	267
1923–27	Fried. Krupp, Essen, Germany. German State Rys.	41,000	4–6–2 (Driver)	4–4	224,800	135,000	120,000	Zoelly	Impulse	6	220	Water-cooled	......	2000	62.3	179.6
1925	Beyer, Peacock & Co., Ltd., Manchester, Eng. L. M. & S. Ry.	38,000	4–6	6–4 (Driver)	......	......	......	....	....	......	285	Air-cooled	......	2000	......	....
1925	North British Loco. Co., Glasgow, Scotland. L. M. & S. Ry.	15,000	4–4	4–4 (Driver)	......	......	......	....	Impulse	4	200	Water-cooled	......	1000	60	....
1926	J. A. Maffei, Munich, Germany. German State Rys.	35,000 Approx.	4–6–2 (Driver)	4–4	230,000	150,000	132,000	Maffei	Impulse-Reaction	3–5	324	Water-cooled	2,370	2500	78	152
1926	Nydgvist & Holm, Trollhattan, Sweden. Swedish State Rys.	32,800	4–6	6–4 (Driver)	148,000	168,000	110,000	....	....	......	280	Air-cooled	12,900	1800	56	176
1926	Nydgvist & Holm Trollhattan, Sweden. Argentine State Rys.	33,000	4–6	8–2 (Driver)	127,000	138,000	115,000	....	....	......	280	Air-cooled	12,900	1750	40.5	151.5

after superheating, enters the single low-pressure cylinder. Cylinder diameters are: high-pressure 8 5/8 in.; low-pressure 23 5/8 in. Stroke is 26 in. Driving wheel diameter is 79 in. Total wheel-base is 63 ft. 2 in., and the overall length is 75 ft. 7 1/2 in.

A 2–6–2 tank locomotive carrying a pressure of 850 lb. per sq. in. was built in 1927 by the Swiss Locomotive and Machine Works, Winterthur. The water-tube boiler has one large drum at the top and two smaller ones at the bottom, all connected by water chambers and tubes. A superheater, feed-water heater, economizer and air preheater are provided. The 3-cylinder uniflow engine is mounted in front of the boiler. It is geared to a jack shaft, the ratio being 2 : 5. Engine and valve gear are enclosed in an oil tight casing. Cylinders: diameter, 8.5 in.; stroke, 13.5 in. Driving wheel diameter is 60 in. Fig. 5 is a diagram of the operation of this engine. Tests made on two divisions of the Swiss Federal Railway provide a comparison with a conventional steam locomotive of approximately equal capacity. These show a coal economy of 35 to 40% and a water economy of 47 to 55%.

Since 1924 the Delaware and Hudson R. R. has placed in service three high-pressure locomotives with water-tube firebox and fire-tube boiler barrel, and a cross compound two-cylinder engine. The boiler pressure of the first locomotive was 350 lb. per sq. in.; of the second 400 lb.; and of the one built in 1930, 500 lb. The 1930 locomotive is of the 2–8–0 type. It exerts a maximum tractive effort of 70,300 lb., working compound. Cylinder diameters are: high-pressure, 20 1/2 in.; low-pressure, 35 1/2 in. Driving wheel diameter is 63 in. Grate area is 82 sq. ft.

Two high-pressure locomotives were built in 1931, one by the American Locomotive Co. for the New York Central and one by the Canadian Pacific Ry. at its Angus shops. Both are of the Schmidt type. Their general dimensions follow:

	N. Y. C.	C. P. R.
Type	4–8–4	2–10–4
Steam pressure (low), lb. per sq. in.	250	250
" " (high), lb. per sq. in.	850	850
" " (closed circuit)	1350	1350
Cylinder diam. (high-pressure), in.	13 1/4	15 1/2
" " (low-pressure), in.	23	24
Stroke (high-pressure), in.	30	28
" (low-pressure), in.	30	30
Driving wheel diam., in.	69	63
Tractive force, lb	66,000	90,000

English Locomotive for Coal Traffic.—(*Ry. Mech. Engr.*, Aug., 1920).—A typical 0–8–0 engine tested with a train of total weight of 1402 tons (2240 lb.) developed over 1100 drawbar horsepower on a 60-mile section. The ruling grade was 0.57% for a short distance. On the test in the opposite direction, a ruling grade of 0.93% for 4 miles was encountered. The train load was 787 tons and a drawbar pull of 28,000 lb. was developed in starting on the ruling grade. Speed reached 16 mi. per hr. 7 min. after starting. The engine was of the 3-cylinder, piston-valve type, with cylinders and valve chests in one casting. The principal dimensions were: Cylinders, 18 1/2 in. diam. × 26 in. stroke; piston valves, 8 3/4 in. diam.; wheel base, 18 ft. 6 in.; driving wheels 55 1/4 in. diam.; weight on drivers, 160,380 lb.; weight of engine and tender, working order, 259,200 lb.; tractive effort, 36,960 lb. The boiler was of the straight type, outside diam. 66 in., with a firebox 9 ft. × 3 ft. 11 in. It contained 102 2-in. tubes, 24 5 1/4-in. flues and 7 2-in. stay tubes, all 16 ft. 2 5/8 in. long. Heating surface, sq. ft., was: tubes, 1407; firebox, 166; total, 1573; superheater surface, 530 sq. ft. Grate area was 27 sq. ft., and working pressure 180 lb. per sq. in. The 6-wheel tender weighed 98,800 lb. and carried 4125 gal. of water and 5 1/2 tons of coal. A Schmidt superheater was used.

TURBO-LOCOMOTIVES—CONDENSING.—The steam turbine-driven condensing locomotive has had limited application in Europe and Argentina. V. P. Dolen-go Kozerovsky (*Mech. Engg.*, Feb., 1929) states its principal advantages to be: Overall efficiency, 14 to 22%, or 1.5 to 2.5 times that of the piston locomotive; continuous driving torque; smooth initial starting torque; elimination of boiler washing; reduction in weight of feedwater and fuel to be carried. It costs more than the piston locomotive and its condenser is somewhat dependent on atmospheric conditions.

The brothers Ljungstrom (Aktiebolaget Ljungstrom Angturbin), Stockholm, designed and built the first turbo-locomotive, which was placed in service on the Swedish State Railways during the winter of 1921–22. The usual reciprocating engines were replaced by a turbine on the tender, exhausting into an air-cooled condenser. Air for combustion was preheated by the waste gases, feedwater was heated by exhaust steam from the auxiliaries. Draft was created by a turbine-driven fan. A maximum of 1500 Hp. and a maximum drawbar pull of 30,000 lb. was developed. Weight efficiency was 157 lb. per Hp.

Almost at the same time Henry Zoelly, of Escher, Wyss & Co., Zurich, converted an ordinary piston locomotive into a turbo-locomotive, using, however, ideas and principles entirely different from those of Ljungstrom. Both types were successful, and most of defects observed in actual service were eliminated in later designs. Several large con-

cerns in Europe then designed and constructed nine turbo-locomotives, four under the Ljungstrom patents, three embodying Zoelly principles, and two of independent designs; one of the latter had an electrical transmission. Table 9 gives details of these locomotives.

Ljungstrom Locomotive.—The distinctive features of the Ljungstrom locomotives are a single turbine, good distribution of weight, and a good weight efficiency. The impulse-reaction turbine has an unusually large number of rows of blades, giving it almost constant efficiency over a wide speed range. This feature enables the engine to work economically under variable service conditions, whether hauling freight or passenger trains. See curves of the coal consumption of turbine and reciprocating locomotives at different speeds in *Ry. Mech. Eng.*, Nov., 1922. Usually the turbine is uneconomical at slow speeds, and the economy of the Ljungstrom locomotive is below that of a reciprocating locomotive of the same power at speeds below 4 mi. per hr. Another important feature of the Ljungstrom turbine is a rapid rise of the horsepower curve or decline of the tractive-effort curve as speed increases.

The condenser (old type) is unusual in that the plant requires no surplus water, apart from the make-up required for boiler leakage, heating, whistle steam, etc. Test data show the average water requirements to be 170 gal. per 100 mi., a saving of 95%.

The last three locomotives in Table 9 embody improvements and refinements over the first one, among them two whirling blasts of highly preheated air over the fire and provision for heating feedwater. The first locomotive smoked badly, but the latest ones are practically smokeless, cinderless and noiseless.

The last engine has the best combined weight efficiency known for a steam locomotive. Average fuel consumed is 22 lb. of oil per 1000 ton-miles; average vacuum is 16 in. Hg at 100° F. air temperature, and 22 in. at 88° F. The large quantity of distilled water in the condenser acts as a cold reservoir for relatively short overloads.

Disadvantages of the Ljungstrom system are: Location of fuel (not so pronounced with liquid fuels); high power required for driving condenser fans to supply 4500 cu. ft. of cooling air per sec.; flexible live-steam connection between boiler and tender units.

Krupp-Zoelly Locomotives.—All the German locomotives use a water-cooled surface condenser, with the disadvantage of having both a condenser and a recooler for the condenser cooling water. The condenser is small as compared with the air-cooled surface type. The heat transmission from steam to water is about 500 B.t.u. per deg. F. per sq. ft. per hr., whereas for an air-cooled condenser of the Ljungstrom type it is about 25 B.t.u. at an air velocity of 4000 ft. per min.

The principal items of performance of the Krupp-Zoelly type are: 1. Part of the water passing through the sprinklers evaporates, with resulting water consumption approximately one-half that of a piston locomotive of equal capacity; 2. As heat transfer is effected almost exclusively through this evaporation, the recooling plant is practically independent of outside temperature; 3. For the same reason, the air consumption in, and hence power required to move it through, the sprinkler-cooler is small.

Descriptions of the turbine locomotives in Table 9 will be found in the following: Ljungstrom—*Engg.* July 21, Aug. 4, 11 & 18, 1922; *Ry. Mech. Engr.*, Oct. and Nov., 1922. Zoelly—*Trans.* A.S.M.E., 1924. Maffei—*Ry. Mech. Engr.*, Feb., 1927. Ramsay—*Ry. Mech. Engr.*, Jan., 1925.

Turbo-Locomotive—Non-Condensing.—A non-condensing 2-8-0 turbo-locomotive was built by Ljungstrom for a Swedish railway. (See *Ry. Age*, Oct. 29, 1932.) The tractive force is 47,400 lb.; boiler pressure is 185 lb. Total weight is 259,044 lb., with 158,760 lb. on drivers. Drivers are 53 1/8 in. diam.

In spite of the many advantages of the condensing turbine-driven locomotive, the manufacturing cost and complicated arrangement has prevented its general adoption. Experiences with turbo-locomotives have proved the turbine to be a satisfactory prime mover for locomotives. The complications of the condensing type have been avoided in the non-condensing turbo-locomotive. The combined reaction-impulse turbine, in front of the smoke-box, is connected to the side rods through a gear and jack-shaft. The turbine exhausts to the exhaust nozzle as in reciprocating locomotives. Road tests in 1932, comparing the turbine locomotive with a 3-cylinder piston locomotive show an average fuel saving per drawbar Hp.-hr. of approximately 10%.

UNIFLOW LOCOMOTIVES.—The Prussian State Rys. have experimented with locomotives having cylinders working on the Stumpf or uniflow principle, where the flow of steam is always in one direction through the cylinder. (See p. 7–39.) The North Eastern Ry. in England also has a locomotive of this type. The long cylinder (twice the usual length) and the heavy piston which must be used, are disadvantages, difficult to overcome in applying the uniflow principle to locomotives.

GEARED LOCOMOTIVES frequently are used on private railroads in rough country. The entire weight of engine and tender is utilized for adhesion and the locomotives can climb steep grades at low speeds. A 3-cylinder vertical engine, attached to the side of

the boiler, drives a shaft extending the whole length of engine and tender. The shaft has flexible joints and is coupled to the axles by bevel gearing. The wheels are arranged in three 4-wheel swiveling trucks, giving a highly flexible wheel-base. See *Ry. Age*, Dec. 17, 1921. A Shay locomotive of this type, used on the Greenbrier, Cheat and Elk R. R., weighs 150 tons. The gear ratio is 1 to 2.45. Total weight, including tender, is 308,000 lb., all on driving wheels. The U. S. Ry. Administration heavy Mikado locomotive has practically the same hauling capacity on level track. Comparative dimensions of the two locomotives are:

	U. S. R. A., Mikado	Shay, Geared	Shay in % of Mikado
Tractive effort (85%), lb.	60,000	59,740	
Cylinders, in.	Two 27×32	Three 17×18	
Weight on drivers, lb	240,000	308,000	
Total weight of engine, lb.	325,000		
Total weight of engine and tender, lb.	497,000	308,000	
Driving wheels, diam., in.	63	48	
Heating-surface, evaporative, sq. ft.	4,297	1,882	
Heating-surface, superheater, sq. ft.	993	411	
Grate-surface, sq. ft.	70.8	48.	
Drawbar pull, on level	56,000	58,500	104 1/2
" " 1% grade	51,000	55,400	108 1/2
" " 2% "	46,000	52,350	111 1/2
" " 4% "	36,100	46,200	128
" " 6% "	26,150	-40,000	153
" " 8% "	16,200	33,850	209
Sharpest curve	300 ft. rad.	179 ft.	60

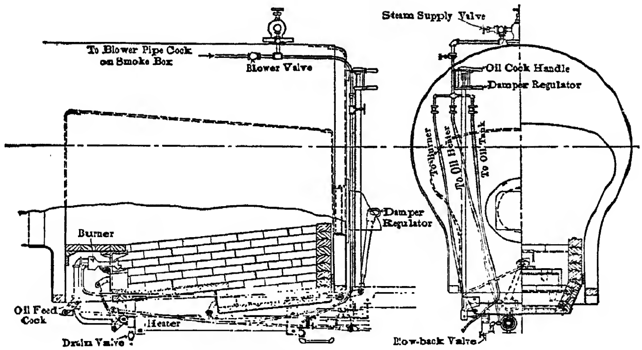


Fig. 6. Firebox Construction for Oil Burning

OIL BURNING LOCOMOTIVES.—The oil fuel tank holds approximately 3000 gal., equivalent to about 17 tons of coal. Oil expands 1% per 25° F. increase in temperature, and 2 in. space over the oil in a square tank, and 6 in. in a semi-cylindrical tank is necessary.

J. N. Clark (*Ry. Mech. Engr.*, Dec., 1921) states that 50 to 175 gal. of oil are required to fire up engines at terminals, requiring from 30 to 110 min. A 2–8–2 locomotive, 207,000 lb. on drivers, was fired up with 1/3 glass of water at 120° to 200 lb. steam pressure in 108 min., using 119 gal. of oil. On the So. Pacific R. R. a train weighing 1000 tons, hauled by a 2–10–2 superheater locomotive, showed fuel consumption per 1000 ton-miles of

Speed, miles per hr.	5	10	15	20	25	30	35	40	45	50
Oil consumed, gal.	0.3	1.1	2.5	4.5	7.0	10.0	13.7	18.0	22.7	28.0

The stand-by fuel consumption was 45 gal. of oil per hr. to hold the train and operate the air pump. A light engine standing under steam burned 37 gal. per hr.

Firebox Construction for Oil-Burning.—The required equipment consists essentially of an injector or atomizer through which the oil is fed into the furnace, a suitable fire-pan replacing the regular ash-pan, and an arrangement of firebrick to protect the lower parts of the furnace sheets from direct action of the flame. Flow of oil is controlled by a plug cock in the feed line. A heater may be placed in this line as it is essential to have the oil warm enough to insure a steady flow to the burner. Figs. 6 and 7 show the general

arrangement. To secure complete combustion and fill the firebox with flame, it is necessary to spray the oil into the furnace. This is done with a steam jet. The burner used by the Baldwin Locomotive Works, Fig. 8, is rectangular in cross-section with two separate ports or chambers (one above the other) running its entire length, with a free outlet for oil at the nose of the burner. The steam outlet, is contracted at this point by an adjustable plate to give a thin wide aperture. The burner is of brass, as an iron casting is sufficiently porous to permit steam to penetrate the oil passage.

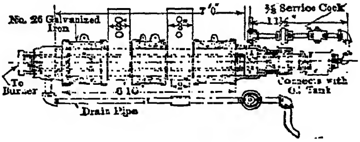


FIG. 7. Oil Heater

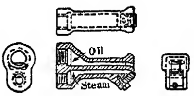


FIG. 8. Baldwin Locomotive Works Oil Burner

Miscellaneous Locomotives

INTERNAL COMBUSTION LOCOMOTIVES are fitted especially for work in contracting operations, plantations, quarries, brickyards, etc., including light switching in railroad yards. Their radius of operation is limited only by the capacity of the fuel tank. The locomotives weigh 5, 7 1/2, 10, 15 and 25 tons. The vertical engine drives through a friction clutch, change gear, jack shaft and the usual locomotive connecting-rods. The gear ratios are: A low gear for starting and accelerating; second gear for exceptionally heavy hauling; third gear, giving a speed of approximately 6 mi. per hr. for light hauling; fourth, high-speed gear for running light, giving a speed of approximately 12 mi. per hr. At rated loads, gasoline consumption will be about 0.1 gal. per Hp.-hr.

COMPRESSED-AIR LOCOMOTIVES.—A compressed-air locomotive comprises essentially a storage tank mounted on driving wheels and two engines similar to those of a steam locomotive. The locomotive tank is charged from storage tanks along the line. These usually are riveted steel cylinders designed for about 1000 lb. per sq. in. working pressure; sometimes small-diameter seamless steel cylinders designed for a working pressure of 2000 lb. or more are used.

COMPOUND AIR LOCOMOTIVES WITH ATMOSPHERIC INTERHEATERS.—Air enters the high-pressure cylinders at 250 lb. gage pressure and is expanded down to 50 lb., while the temperature drops about 140° F. Practically all the lost heat is restored in the atmospheric interheater, which is a cylindrical reservoir filled with brass tubes, placed between the high- and low-pressure cylinders. The air enters the low-pressure cylinder at 50 lb. gage pressure and a temperature within 10 or 20° F. of that of the surrounding atmosphere. The exhaust induces a draft of atmospheric air through the tubes of the interheater. This combination permits air to expand from 250 lb. down to atmosphere without unmanageable refrigeration.

Relative Economy of Simple and Compound Locomotives.—Assume 11.2 cu. ft. of free air to be compressed to 150 lb. gage pressure. Its volume will be 1 cu. ft. Compressed

Table 10.—Data of Internal-combustion Locomotives

(Baldwin Locomotive Works, Philadelphia, 1934)

Weight of locomotive, lb.	10,000	15,000	20,000	30,000	50,000
Rated horsepower of motor	35	50	65	100	135
Normal speed of motor, r.p.m.	650	650	550	550	520
Number of cylinders	4	4	4	6	6
Diam. and stroke of cylinders, in.	5×7 1/2	6 1/4×8	7 1/4×9	7 1/4 ×9	7 3/4×12
Drawbar pull, straight level track					
At 4 mi. per hr., lb.	2400	3550	4700	7100	9400
At 6 mi. per hr., lb.	1600	2300	3000	4600	6100
At 12 mi. per hr., lb.	700	1000	1350	2100	2700
No. and diam. of wheels	4–24'	4–26"	4–28"	6–30"	6–36"
Wheel-base, in.	45	60	72	76	96
Height over cab or canopy	7'–6"	8'–6"	9'–0"	10'–4"	11'–0"
Height without cab or canopy	5'–8"	6'–8"	7'–6"	8'–0"	
Length over frames	12'–1"	14'–3"	16'–0"	18'–4"	19'–10"
Minimum gage, in., inside frames	36	42	48	48	56 1/2
Minimum gage, in., outside frames	24	24	24	36	
Fuel tank capacity, gal.	25	35	35	50	50

to 250 lb. gage pressure, its volume will be 0.623 cu. ft. Allow these volumes to expand by moving pistons of 1 sq. ft. and 0.623 sq. ft. area, respectively, a distance of 1 ft. The energy developed will be: At 150 lb., 21,600 ft.-lb.; at 250 lb., 22,425 ft.-lb. Now assume the 11.2 cu. ft. of free air compressed to 250 lb. gage to be expanded in two cylinders of 4 : 1 ratio. The energy developed will be 35,880 ft.-lb. if the heat be restored between the two cylinders. The gain due to compounding from 250 lb. pressure, with interheating, as compared to simple expansion from 150 lb. pressure is 35,880/21,600 = 166%.

These results are about as good as can be obtained with either simple or compound locomotives. Any improvement due to expansive working just about balances losses due to clearance and initial refrigeration. The work done per cu. ft. of free air in the two systems is: Simple cylinders, 21,600/11.2 = 1840 ft.-lb.; compound cylinders and atmospheric interheater, 35,880/11.2 = 3205 ft.-lb. Actual tests show 1900 ft.-lb. of work per cu. ft. of free air with the simple locomotive, and 3000 ft.-lb. with the compound, the gain due to expansive working and the losses due to internal friction being somewhat greater in the compound than in the simple engine.

In the operation of compressed air locomotives the air compressor generally delivers compressed air at a pressure ranging from 800 to 1000 lb. per sq. in. to the storage reservoir. An average of about 12,000 ft.-lb. per cu. ft. of free air is required to compress and deliver it at these pressures. The efficiency of the two systems then is: 1900/12,000 = 16% for the simple locomotive; 3000/12,000 = 25% for the compound locomotive with atmospheric interheater.

OIL-ELECTRIC LOCOMOTIVES.—A report of a committee of the Amer. Ry. Assoc. shows that there were about 125 oil-electric locomotives in service on U. S. railroads at the beginning of 1932. The rated horsepower ranges from 300 to 800. Compared with steam locomotives on a horsepower basis, their first cost is considerably higher. The total operating costs per hour, including fixed charges and depreciation, show wide fluctuation, but are somewhat less than for steam operation. From the standpoint of smoke and noise prevention, the oil-electric locomotives have an advantage for terminal and city yard operations, tunnel runs and operations in yards adjacent to residential sections.

In 1928 the Canadian National Rys. placed in service a 2660-Hp. oil-electric locomotive of the 4–8–2–2–8–4 type. Each of the two units of the locomotive is driven by a Beardmore 12-cylinder, solid injection type engine, 12-in. bore and 12-in. stroke, rated at 1330 Hp. at 800 r.p.m. The engines are coupled to generators; four Westinghouse traction motors geared to the driving axles, are used in each unit. The locomotive develops a tractive force of 100,000 lb. during accelerating periods, and 42,000 lb. continuously. The engines develop their rated horsepower on a fuel rate of 0.43 lb. of oil per B. Hp.-hr.

GASOLINE-ELECTRIC RAIL CAR.—(*Ry. Mech. Engr.*, Aug., 1932). An unusual arrangement of rail car, in service on the A., T. & S.F.R.R., consists of a power unit 30 ft. long, and a baggage and express unit 60 ft. long, the two being joined by a center

Table 11. Data of Two-stage Compressed-air Locomotives

(H. K. Porter Co., Pittsburgh)

Type	0–4–0	0–4–0	0–6–0	0–4–0	0–6–0	0–4–0
Cylinder diameter, h. p., in.	4 1/2	7	7	9 1/2	10	15
Cylinder diameter, l. p., in.	9	14	14	14 and 14	14 and 14	30
Stroke, in.	10	14	14	14	14	24
Driving wheels, diameter, in.	22	26	26	26	26	46
Wheel base, rigid, ft. and in.	2–9	4–0	5–6	4–6	6–6	6–6
Length over bumpers, ft. and in.	9′ to 12′–6″	12′ to 18′–6″	14′ to 20′	18′ to 21′	20′ to 23′	22′ to 26′
Height above rail, ft. and in.	....	....	5′ to 5′–10″	5′ to 5′–10″	5′ to 5′–10″	12′
Reservoir capacity, cu. ft.	20 to 60	60 to 120	97 to 160	160 to 275	160 to 290	260 to 375
Weight in working order, lb.	7,000 to 10,000	18,000 to 22,000	20,000 to 27,000	32,000 to 42,000	43,000 to 46,000	95,000 to 100,000
Tractive force, lb.	1450	4400	4400	8000	9200	19,500
Hauling capacity, maximum, exclusive of locomotive, tons:						
On level	68	210	206	380	437	925
" 1% grade	32	100	96	180	207	437
" 2% "	19	63	59	113	130	275
" 3% "	13	45	31	80	92	193
" 5% "	7	26	22	46	53	112
Weight of lightest rail advised, lb. per yd.	16	30	25	45	40	80
Radius of sharpest curve advised ft	15	30	50	40	70	70

articulated truck. Under the car are three 4-wheel trucks of the drop-equalizer type; one is at the junction between the two parts. The V-type Winton engine has 12 cylinders, 9 in. bore and 12 in. stroke. It is rated at 900 Hp. at 900 r.p.m. The electrical equipment comprises a General Electric direct-current generator on the engine shaft, and four traction motors geared to the axles of the two forward trucks. A twin-cylinder 2-stage air compressor displacing 70 cu. ft. per min. is built into the engine. The engine has a carburetor for each cylinder, to use either gasoline or distillate. Throttling is by control of the intake valves. The individual carburetors shorten the travel of fuel and air mixture, permit the use of cheap non-volatile fuels and insure even distribution to all cylinders. Air for combustion enters through felt air cleaners under the roof. Picking up of fuel at the intake valve eliminates all handling of fuel and air mixtures outside of the cylinders and overcomes back-fire hazards. Each cylinder has two exhaust valves. The single-bearing generator has a built-in exciter and differential field control of the voltage. Total weight of the car is 245,000 lb. It can haul four heavy passenger cars as trailers at speeds up to 80 m.p.h.

LIGHT-WEIGHT STREAMLINED TRAINS (*Ry. Age*, Feb. 2, 1934).—Gasoline-electric trains, designed for normal speeds of 90 m.p.h. on level tangent, and maximum speed of 110 m.p.h. have been built for the Union Pacific Ry. To reduce weight, the trains were built throughout of aluminum alloy, except that truck wheels and axles were alloy-steel castings of 50,000 lb. per sq. in. tensile strength. The cars are of tubular shape, streamlined to reduce air resistance, and form a deep stiff beam. Four 4-wheel trucks carry the three articulated car bodies comprising the first train. The first car, 72 ft. 8 in. long, contains the power plant, a mail compartment and baggage room; the second car, 59 ft. 10 in. long, seats 60 passengers; the third car, 71 ft. 11 in. long, seats 56 passengers and contains a buffet. Total overall length is 204 ft. 5 in., and total weight is 170,000 lb., of which 85,000 lb. is carried on the first truck. The center of gravity is kept low to insure safety and easy riding, being 38 in. above rails, or 20 in. lower than in the usual car. The car bottoms are 9 1/2 in., and the car tops 11 ft., above the rails. Trucks are shrouded, reducing air resistance 20%. Windows and doors are set flush for the same purpose. The streamlining was based on wind-tunnel tests with a model built on a scale 3/8 in. per ft.

The power plant comprises a 12-cylinder V-type distillate-burning engine, developing 600 Hp. at 1200 r.p.m., driving a 425-kw. generator with built-in exciter. The engine cylinders are 7 1/2 in. diam., 8 1/2 in. stroke. Pistons are aluminum alloy, and connecting-rods drop-forged alloy steel. The 7 main high-lead-bronze bearings are 5 1/2 in. diam. The 12 babbitted connecting-rod bearings are 4 3/4 × 3 in. Forced feed lubrication is used, with crank-case supply. The engine block and crank-case are fabricated steel. Generator voltage is regulated by traction demand to give a constant load on the engine at any speed. Two 300-Hp. motors, with forced ventilation, on the front truck are geared to the wheels. The braking system is designed to give uniform retardation by controlling brake shoe pressure in proportion to speed by means of a decelerometer control.

4. ELECTRIC LOCOMOTIVES

CLASSIFICATION (Recommended by Committee of Am. Ry. Assoc.).—Idle or guiding axles are designated by numbers, driving axles by letters, an articulated joint by a plus sign, and swivel type trucks, not articulated, by a minus sign. Table 12 gives examples. This classification is recommended also for internal combustion locomotives with electric transmission.

STANDARD METHOD OF RATING ELECTRIC LOCOMOTIVES (From *Recommended Practice*, Am. Ry. Assoc., 1930).—Ratings are taken as at rims of drivers, with locomotive at constant speed on tangent, level track, and are: *a*, Maximum start; *b*, One-hour; *c*, Continuous. At each rating: *a*. Speed (V), mi. per hr.; *b*. Tractive effort (T), lb.; and horsepower (Hp.) are given. Hp. $= VT/375$.

Locomotive ratings are based on motor shaft ratings for maximum start (except as noted below) for 1 hr. and continuously, with motors operating at rated voltage. The 1-hr. motor ratings are with motors starting cold. Motor torque is reduced 3% when determining tractive effort, to compensate for mechanical losses, unless test data are available. Motor shaft ratings are determined by stand tests, and under agreed conditions of temperature rise, ventilation, etc., except with respect to 1-hr. rating.

If test data are not available, tractive effort and speed are determined by $T = 0.97\ (24\,tG/dP)$; $V = SPd/336.1\,G$; where t = motor shaft torque, lb.-ft.; G = number of gear teeth; P = number of pinion teeth; d = driver diameter, in.; S = r.p.m. of motor.

The maximum start rating of locomotive is at the maximum torque exerted by motors

Table 12.—Classification of Electric Locomotives

TYPE SYMBOL	WHEEL ARRANGEMENT
B – B	
C – C	
B – B + B – B	
2 – B + B + B + B – 2	
1 – B + D + D + B – 1	
2 – C + C – 2	
2 – C – 1 + 1 – C – 2	
2 – B – 2	
2 – C – 2	
1 – D – 1	

with any combination of connections, and at maximum speed attainable with this combination at maximum torque. If tractive effort so derived exceeds 25% of total weight on drivers, tractive effort is taken as 25% of total weight on drivers, and the speed is the maximum attainable at such tractive effort, with any combination of motor connections. The 1-hr. and continuous ratings of locomotive are determined from the 1-hr. and continuous ratings of motors.

TRACTIVE RESISTANCE OF ELECTRIC LOCOMOTIVES AND CARS (W. J. Davis, Jr., *Gen. Elec. Rev.*, Oct., 1926).—Table 13 summarizes formulas derived from an analytical study of various tests and investigations of train resistance. The first two terms of the equations, derived from dynamometer and coasting tests on standard freight and passenger cars and electric locomotives, represent journal friction almost entirely, and are based on oil lubrication at average temperatures. Journal friction may increase 20 to 40% at temperatures below 32° F. The third term comprises resistances due to flange friction, concussion, swaying and other frictions proportional to speed. The factor of this term decreases with increase of truck wheel-base, and increases with poor roadbed conditions and inferior riding qualities of motor cars. The last term gives air resistance, lb. per ton, for average weight of car or locomotive for standard types of equipment. No allowance is made for head or strong side winds. Locomotive resistance represents tractive effort delivered to driving axles, excluding friction losses in gears,

Table 13.—Train Resistance Formulas for Electric Locomotive and Motor Car Service

Notation.—R = tractive resistance, lb. per ton (2000 lb.) on tangent, level track; A = area, sq. ft., of cross-section of locomotive or car body and trucks; V = speed, mi. per hr.; n = No. of axles per car; w = average weight per axle, tons; wn = average weight of locomotive or car, tons.

Values of A: *Locomotives.*—50-ton, 105; 70-ton, 110; 100-ton and over, 120. *Freight cars,* 85–90; *Passenger cars,* 120. *Multiple-unit cars,* 100–110. *Motor cars:* 2-truck, 80–100; 1-truck, 70–75.

Where Used	Usual Formula Recommended for convenience in calculation. Approved for axle weights over 5 tons.	General Formula Applicable to all axle weights. To be used when axle weights are less than 5 tons.
Locomotives	$R = 1.3+\frac{29}{w}+0.03V+\frac{0.0024AV^2}{wn}$	$R=\frac{9.4}{\sqrt{w}}+\frac{12.5}{w}+0.03V+\frac{0.0024AV^2}{wn}$
Freight Cars	$R = 1.3+\frac{29}{w}+0.045V+\frac{0.0005AV^2}{wn}$	$R=\frac{9.4}{\sqrt{w}}+\frac{12.5}{w}+0.045V+\frac{0.0005AV^2}{wn}$
Passenger Cars (Vestibuled)	$R = 1.3+\frac{29}{w}+0.03V+\frac{0.00034AV^2}{wn}$	$R=\frac{9.4}{\sqrt{w}}+\frac{12.5}{w}+0.03V+\frac{0.00034AV^2}{wn}$
Multiple Unit Trains: Leading Car (Vestibuled)	$R = 1.3+\frac{29}{w}+0.045V+\frac{0.0024AV^2}{wn}$	$R=\frac{9.4}{\sqrt{w}}+\frac{12.5}{w}+0.045V+\frac{0.0024AV^2}{wn}$
Trailing Cars	$R = 1.3+\frac{29}{w}+0.045V+\frac{0.00034AV^2}{wn}$	$R=\frac{9.4}{\sqrt{w}}+\frac{12.5}{w}+0.045V+\frac{0.00034AV^2}{wn}$
Motor Cars	$R = 1.3+\frac{29}{w}+0.09V+\frac{0.0024AV^2}{wn}$	$R=\frac{9.4}{\sqrt{w}}+\frac{12.5}{w}+0.09V+\frac{0.0024AV^2}{wn}$

motor bearings and other parts of driving equipment. These usually are covered in motive power efficiency. The formulas are based on tests in mild weather conditions. Values obtained from them may be used in calculations of electric distributing systems, substations, energy consumption, and power demand. In determining electric motor charactristics and gear reductions to meet particular speed requirements, it may be desirable to add a small percentage to the required speed, as insurance against unusual conditions.

Table 14.—Data of Electric Locomotives (See Notes p. 14–21)

Railroad	C.U.T.[1]	N.Y.C.[2]		Va.[4]		P.R.R.[6]	P.R.R.	P.R.R.
Builder: Electrical	G.E.	G.E.	G.E.	Westinghouse	Westinghouse	West. & G.E.	West. & G.E.	Westinghouse
Mechanical		G.E.	G.E.	Am. Loco.	Baldwin	P.R.R.	P.R.R.	P.R.R.
Wheel arrangement	2-C+C-2	C+C	2-C+C-2	1-D-1	1-D-1	2-C-2	1-D-1	0-C-0
Service	Pass.	Freight	Pass.	Freight	F. & P.	Pass	Freight	Shift.
Series No. or class	P1A	R2	0351–0360	100–115	5000–3	P-5	L-6	B1
Placed in service in	1930	1930–1	1931	1925–6	1926–8	1932	....	1926
A.C. or D.C.	D.C.	D.C.	A.C. & D.C.	A.C.	A.C.	A.C.	A.C.	A.C.
Frequency			25	25	25	25	25	25
Contact conductor:								
Voltage		0.6 kv.	11 kv. / 0.6 kv.	11 kv.	11 kv.	11 kv.	11 kv.	11 kv.
Type	Catenary	3d rail	Cat. / 3d rail	Cat.	Cat.	Cat.	Cat.	Cat.
Current collector	Pantograph	Shoe & overhead collector	Pant. / 3d rail shoe	Pant.	Pant.	Pant.	Pant.	Pant.
Driving wheels:								
Number	12	12	12	8		6		6
Diameter, in.	48	44	56	61	56	72	62	62
Truck wheels:								
Number	8		8	4	4	8	4	
Diameter, in.	36		36	33	42	36	36	
Weight, lb.:								
Total	419,000	266,500	403,000	1,282,400*	368,600	375,000	330,000	158,000
On drivers	312,000	266,500	272,000	922,600*	282,700	225,000	240,000	158,000
Per driving axle	52,000	44,417	45,333	76,900*	70,700	75,000	60,000	52,667
Mechanical parts	265,780	158,800	230,000	812,300*	191,800			
Electrical parts	144,220	106,200	173,000	472,300*	176,800			
Length overall	80′–0″	54′–0″	77′–0″	50′–9″	47′–2″	62′–8″	51′–1[illegible]	30′–2 3/4″
Width overall			10′–0″	11′–1″	11′–0″			10′–1″
Height, pantograph down	14′–9″		14′–3 3/16″			15′–0″		15′–0″
Driver wheel-base	15′–0″		13′–8″	16′–6″		20′–0″	20′–0″	
Total wheel-base	69′–0″	40′–0″	66′–0″		31′–5″	49′–10″	36′–0″	
Traction motor:								
Armatures, No.	6	6	12			6	4	3
Armatures moun'd	Single		Twin			Twin	Single	Single
Type			S.P.C.‖	Induct.	D.C. series	S.P.C.‖	S.P.C.‖	S.P.C.‖
Method of drive	Gear	Gear	G. & Q.†	G. & S.R.‡	Gear	G. & Q.†	Gear	Gear
Gear ratio	2.741	3.45	4.05	4.762	5.06	2.935	4.30	5.438
Tractive force, lb.:								
Estimated, 25% of wt. on drivers	78,000	66,625	68,000	230,650	70,675	56,250	60,000	39,500
			D.C.					
1-hr. rating, forced ventilation	30,600	41,800	25,200 / 31,200	54,000 / 31,500	56,250			22,100
Cont. rating, forced ventilation	19,200	21,500	18,000 / 22,500	45,000 / 26,200	44,250	22,300 @ 63 m.p.h.	24,800 @ 7.8 m.p.h.	13,500 @ 5.9 m.p.h.
Horsepower:								
1-hr. rating, forced ventilation	3030	2490	3440 / 3140	2030 / 2375	2165			730
Cont. rating, forced ventilation	2635	2010	2740 / 2610	1700 / 2000	1770	3750	2500	570
Speed, m.p.h.: 1-hr. rating	37	22.3	51.2 / 37.4	14.1 / 28.3	14			12.4
Cont. rating	51.5	35.1	57.0 / 43.5					
Max. (safe or sustained)	70 (Safe)	60 (Safe)	70 (Safe)	38 (Safe)	45 (Safe)	90 (Sus.)	54 (Sus.)	25 (Sus.)
Equipped for:								
Regeneration	No	No	No	Yes	Yes	No		No
Multiple-unit	Yes	Yes	Yes	Yes	Yes	Yes	Yes	Yes

* Weights apply to complete 3-unit locomotive.
‡ Gear and side-rod.
† Gear and quill.
‖ Single-phase commutator.

ELECTRIC LOCOMOTIVE DATA.—Table 14 gives data of typical electric locomotives in service in 1934. In general, single units which may be coupled in service to form operating locomotives of one, two or three units, are listed.

TYPICAL ELECTRIC LOCOMOTIVES.—N. Y. N. H. and H. R. R.—General data of these locomotives are given in Table 14. J. C. Hassett and W. M. Guynes describe them in detail in *Ry. Age*, Sept. 19, 1931–June 11, 1932. They operate over maximum grades of 1.5% between New Haven and Grand Central Terminal, New York, and between Penna. Terminal, New York, and the Sunnyside yards, with 11,000-volt, 25-cycle, single-phase alternating current, and also with 600-volt direct current. Two twin-arrangement motors, mounted on each axle, are geared to a quill surrounding the axle. Spiders on the quill engage the driving wheel spokes through soring cups. A twin motor complete with quill gear, pinions, gear case and quill linings weighs 14,000 lb., or 25.5 lb. per continuous Hp., and requires 4300 cu. ft. of cooling air per min. To supply a total of 30,000 cu. ft. of air per min. for motors, grids, etc., requires 34 Hp. Tests show 550 Hp. per axle to be a safe continuous rating of the motors; the estimated rating was 457 Hp. Troubles due to poor commutation at starting and at high speeds, evident in former A. C. traction motors, have been overcome by a low flux per pole and the use of 12 poles, and by varying commutating field strength, and by providing brush holders of low inertia in their moving parts. The contract ratings of the motors are

A. C. continuous,	18,000 lb.	tractive effort at	57	mi. per hr., or	2740 Hp.
A. C. hourly,	25,200 "	" " "	51.2	" " " "	3440 "
D. C. continuous,	22,500 "	" " "	43.5	" " " "	2610 "
D. C. hourly,	31,200 "	" " "	37.4	" " " "	3140 "

The train capacity on tangent level track at full voltage in express service is 15 80-ton cars at 65 m.p.h. in the A. C. zone, and at 58 m.p.h. in the D. C. zone.

Penna. R. R. (J. V. B. Duer, *Ry. Age*, May 21, 1932).—The 11,000-volt, 25-cycle, electrified section between New York and Washington, 227 miles, uses locomotives of four types, viz.:

Type	Class	Service	Hp. at driver rims	Rating speed, m.p.h.	Rated tractive force, lb.	Max. operating speed, m.p.h.
2-C-2	P5	Heavy-duty Passenger	3750	63	22,300	90
2-B-2	O1	Light-duty Passenger	2500	63	14,900	90
1-D-1	L6	Freight	2500	37.8	24,800	54
0-C-0	B1	Shifting	570	15.9	13,500	25

The weight on a pair of drivers does not exceed 75,000 lb.; largest single locomotive unit weighs 375,000 lb. The main motors are single-phase commutator type, with a continuous rating of 625 Hp. per armature at driver rims. The motors of passenger locomotives are twin-armature, frame-supported, geared to the driving wheels through a quill drive. The freight locomotive has single, axle-mounted motors geared to each driving axle. Driving and truck axles and motor armatures are mounted in anti-friction bearings. These locomotives can be used singly or in combination to meet traffic de-

Notes to Table 14

[1] **Cleveland Union Terminal** (C. U. T.).—A 3000-volt direct-current installation with overhead catenary, of 22 locomotives to operate a terminal electrification, 17 miles long at Cleveland, Ohio.

[2] **New York Central** (N.Y.C.).—A 600-volt direct-current installation of 42 locomotives to handle freight trains between Harmon, N. Y., and New York City, 31 miles. While rated as freight locomotives, they sometimes are used on passenger trains into Grand Central Terminal.

[3] **New Haven** (N.Y.N.H. & H.).—Designed to operate on either alternating or direct current and used in passenger service between New Haven, Conn., and New York City. See above.

[4] **Virginian** (Va.).—A split-phase, constant-speed locomotive. Single-phase 11,000-volt, 25-cycle current is taken from an overhead catenary. A phase-converter on the locomotive supplies 3-phase induction motors. Regenerative braking is used. The running speeds are 14 and 28 m.p.h. These locomotives operate coal trains between Mullens, W. Va., and Roanoke, Va., 133 route miles. The data cover a 3-cab locomotive operated as a complete articulated locomotive. The momentary maximum tractive force is 277,000 lb.

[5] **Great Northern** (G.N.).—A motor-generator locomotive receiving 11,000-volt alternating current from an overhead catenary, converted to direct current on the locomotive. Regenerative braking is used on descending grades. The electrified line extends from Wenatchie, Wash., to Skykomish, Wash., 80 miles.

[6] **Pennsylvania** (P.R.R.).—Locomotives used on the New York-Washington electrification. See above.

mands. Thus, 2500 nominal B.Hp. is available in the single O1 locomotive; two O1 units coupled have 5000 Hp.; and an O1 and a P5, 6250 Hp.; two P5 units, 7500 Hp.

Each locomotive assembly consists of: 1. A chassis, made up of main frame, drivers, trucks, spring and brake rigging, motors, flexible drives, main transformer, and train heating boiler, together with the necessary storage reservoirs for air, oil and water. The foundation is a single steel casting in which are cast the air reservoirs and the oil and water reservoirs for the train heating boiler. 2. The deck unit, carrying the main control groups, air compressor, main wiring and most of the control wiring. The foundation is a structural aluminum framework, mounted on the chassis unit. 3. The cab unit.

Instead of a current-rupturing device between the pantograph and the primary of the main transformer, a pantograph lowering relay is used. This is so connected as to use the substation circuit breakers to open high-tension short-circuits, grounds, or overloads, on the locomotive. The P5, L6 and O1 locomotives also have a grounding device which functions automatically under transformer short-circuit or overload conditions to ground the pantograph. This opens the substation circuit breakers, after which the pantograph automatically lowers and locks.

Relay devices protect against motor overload, thus permitting maximum use to be made of tractive force available from driving wheel adhesion. At the same time they protect the motors should adhesion exceed normal values due to the application of sand or mis-handling of equipment. This relay protection is interlinked with anti-wheel-slip protection to direct the engineman's attention to driving wheel slippage.

Swiss (*Ry. Age*, Feb. 20, 1932).—A 1—B—1—B—1+1—B—1—B—1 electric locomotive for the St. Gothard line of the Swiss Federal Railroads was designed for heavy grades between Ertzfeld and Bellinzona. The locomotive is 111 1/2 ft. long and comprises two symmetrical sections with a joint between to permit their separation. Each section is complete in itself and has four motors and four driving axles. The locomotive, designed for both freight and passenger service, weighs 245 tons and develops 8500 maximum Hp. and maximum speed of 62 m.p.h. It will haul 1500-ton trains of 37 40-ton cars, on level track, and 750-ton trains over the 2.7% grades. The mechanical parts were built by the Locomotive and Machinery Wks., Winterthur, and the electrical parts by Brown-Boveri and Co., Baden, Switzerland.

5. LOCOMOTIVE DESIGN AND CONSTRUCTION

FACTORS IN LOCOMOTIVE DESIGN.—The general dimensions of a locomotive for any given conditions of service can be determined most conveniently, with sufficient accuracy for a preliminary estimate, by means of design factors. Let T = rated tractive force, lb.; W_a = adhesive weight, or weight on drivers, lb.; W_t = total weight of locomotive, lb.; S_h = evaporative heating-surface, sq. ft.; S_s = superheater surface, sq. ft.; G = grate area, sq. ft.; D = diameter of drivers, in.; d = diameter of cylinders, in.; s = stroke, in.; P = boiler pressure, lb. per sq. in.; A = adhesion factor; B = boiler factor; C = combustion factor; B_d = boiler demand factor; E = efficiency of design factor; F = superheater factor. Then according to Lawford H. Fry (*Ry. Mech. Engr.*, April, 1921), the most important factors in common use are:

$A = W_a/T$	$C = S_h/G$	$E = W_t/S_h$
$B = T/S_h$	$B_d = TD/S_h$	$F = S_s/S_h$

The rated tractive force is computed from $T = (0.85\,d^2\,sP)/D$, and is the tractive force deliverable by the cylinders at the driver rims, if cut-off is long enough to give 92% of boiler pressure as mean effective pressure in the cylinders, and if machine friction absorbs about 8% of the indicated cylinder power. Table 15 gives values of factors for different types of locomotives.

Adhesion factor A shows the weight holding the driving wheels to the rail per pound of average force tending to rotate them. $A \geqq 4.0$ probably will prevent slipping of wheels when starting.

Boiler factor B measures the pounds of rated tractive effort to be developed per sq. ft. of boiler heating-surface with the engine working in full gear. Two locomotives at the same speed V will develop horsepowers directly proportional to the tractive effort, according to formula Hp. = $TV/375$.

Boiler demand factor B_d is proportional in full gear to the foot-pounds of work per revolution done by each square foot of heating surface. This factor is used to determine rated tractive effort T.

Combustion factor C depends on quality of fuel. With anthracite C = 35 to 40;

with bituminous coal, $C = 55$ to 60, for superheater locomotives. With saturated steam these figures should be increased by from 10% to 20%.

Efficiency of design factor E shows relation between weight and total area of heating-surface.

E. C. Poultney (*Ry. Mech. Engr.*, Sept. and Oct., 1921) gives tables of dimensions and proportions of British locomotives, using the above factors. A summary of these factors for locomotives built between 1910 and 1920 is given in Table 16.

LOCOMOTIVE BOILERS.—The weight and dimensions of locomotive boilers are in nearly all cases determined by the limits of weight and space to which they are necessarily confined. In general, within these limits a locomotive boiler cannot be made too large. The Am. Ry. Master Mechanics Assoc. Committee of 1902 advised the ratios of heating to grate surface given in Table 17. These figures should be modified by the formulas on locomotive proportions adopted by the same body in 1916. See p. 14–36.

LOCOMOTIVE PROPORTIONS (Committee of Am. Ry. Master Mechanics Assoc., 1916).—Obtain tractive force from weight limitations on drivers, service, type, etc. See p. 14–05.

Let A = area of cylinder, sq. in.; A_g = area of grate-surface, sq. ft.; D = diam. of drivers, in.; d = diam. of cylinders, in.; F = tractive force, lb.; H_f = heating surface of firebox, sq. ft.; P = boiler pressure, lb. per sq. in.; s = stroke, in.; W_c = weight of coal, lb. per hr.; W_s = total weight of steam required, lb. per hr.; W_f, W_t = respectively, evaporation by firebox (including combustion chamber and arch tubes), and by tubes and flues, lb. per hr. Then $d = \sqrt{FD/0.85\,Ps}$; Hp. $= 0.212\,PA$ for saturated steam; $= 0.0229\,PA$ for superheated steam. Maximum Hp. is assumed at piston speeds of 700 ft. per min. with saturated steam, and 1000 ft. per min. with superheated steam. $W_s = 27.0 \times$ Hp. for saturated steam; $= 20.8 \times$ Hp. for superheated steam. $W_c = 4 \times$ Hp. for saturated steam; $= 3.25 \times$ Hp. for superheated steam. $A_g = W_c/120 =$ Hp./30 for saturated steam; $=$ Hp./36.9 for superheated steam. $W_f = H_f \times 55$; $W_t = W_c - W_f$.

Table 15.—Average Values of Design Factors for American Locomotive Types

Type	Steam	Date	No. of Locomotives	A	B	B_d	E	S_s/S_h
4–6–0	Sat.	1910	11	4.6	10.7	720	66	......
4–6–0	Sup'htd.	1910	3	4.3	13.9	990	80	0.169
4–6–0	"	1920	3	5.0	13.5	950	87	.206
4–6–2	Sat.	1910	31	4.3	9.0	620	61	......
4–6–2	Sup'htd.	1910	8	4.5	11.2	800	73	.241
4–6–2	"	1920	28	4.3	12.0	840	76	.229
4–8–2	"	1920	10	4.3	12.6	860	80	.235
2–8–0	Sat.	1910	21	4.3	13.7	800	66	......
2–8–0	Sup'htd.	1920	16	4.1	17.8	1015	82	.214
2–8–2	"	1920	28	4.1	14.1	870	75	.228
2–10–2	"	1920	12	4.2	15.1	920	78	.240

Table 16.—Design Factors of Typical British Tender Engines

Type	Steam	A	B_d	C	$F.B.S./S_t$*	$F.B.S./G$*	T/G	E
4–4–0	Superheated	4.25	1126	70.0	8.8	6.2	828	82.7
"	Saturated	4.3	918	72.5	8.5	6.0	830	75.3
4–4–2	Superheated	4.2	967	74.4	7.8	5.7	764	77.3
"	Saturated	4.3	717	77.7	6.9	5.5	700	67.2
4–6–0	Superheated	5.1	1000	83.0	7.5	5.8	890	75.7
"	Saturated	5.2	833	79.4	7.1	5.4	884	76.3
0–8–0	Superheated	4.8	1097	81.5	7.9	6.3	1250	68.6
"	Saturated	4.5	874	80.7	7.9	6.0	1290	72.4
0–6–0	Superheated	4.6	1103	70.5	9.0	6.1	1160	80.0
"	Saturated	4.7	980	64.8	8.9	5.9	1020	76.2

* $F.B.S.$ = firebox surface; S_t = total heating surface, including superheater when used.

Table 17.—Recommended Ratios of Heating-surface to Grate-surface

Fuel	Passenger		Freight	
	Simple	Compound	Simple	Compound
Free-burning bituminous	65 to 90	75 to 95	70 to 85	65 to 85
Average bituminous	50 to 65	60 to 75	45 to 70	50 to 65
Slow-burning bituminous	40 to 50	35 to 60	35 to 45	45 to 50
Bituminous slack and free-burning anthracite	35 to 40	30 to 35	30 to 35	40 to 45
Low grade bituminous, lignite and slow-burning anthracite	28 to 35	24 to 30	25 to 30	30 to 40

Basis of evaporation is 10 lb. of water per hr. per sq. ft. of heating surface. Ratio of cu. ft. of firebox volume to sq. ft. of grate surface should range from 5.5 to 6 for bituminous coal, and from 4.5 to 4.85 for anthracite. Ratio of tube length to tube diameter should be about 100. Ratio of superheating to total saturated steam heating-surface should be about 0.22 without combustion chamber, and 0.29 with combustion chamber.

CYLINDER SIZES usually are such that the engine just will overcome the adhesion of the wheels to the rails under favorable conditions. Let W = weight on drivers, lb.; P = tractive force, lb., $= 0.25\,W$ for passenger engines, $= 0.24\,W$ for freight, $= 0.22\,W$ for switching; p, p_b = respectively, mean effective and boiler pressure, lb. per sq. in.; D, d, = respectively, diam. of drivers and cylinder, in.; s = length of stroke, in.; $p = 0.85\,p_b$. Then $W = 4\,P = 4\,d^2\,ps/D = 4\,d^2 \times 0.85\,p_b\,s/D$.

$$d =$$

Von Borrie gives for the diameter d_1 of the low-pressure cylinder, $d_1^2 = 2\,ZD/ps$, where Z = tractive force required, usually 0.14 to 0.16 × adhesion. Internal machine friction is deducted in ascertaining p, whose value depends on the cylinder volume ratio. Indicator experiments give the following values:

	Cylinder Volume Ratio	p/p_b	p when $p_b = 176$
Large-tender engines................	1 : 2 or 1 : 2.05	42	74
Tank engines......................	1 : 2 or 1 : 2.2	40	71

PRESSURES AND STRESSES IN LOCOMOTIVE PARTS (Brian Reed, *Engr.*, June 17, 1932).—An analysis of a large number of locomotives built since 1919, the majority between 1926–32, was made to determine stresses in the various parts. The locomotives represent British and American practice and also locomotives used in Europe and in other countries. The original article contains many tables of dimensions, stresses, calculations, etc., which are summarized below.

Main and Side Rods.—Most of the main and side rods were of high tensile strength Ni-Cr steel. The vertical and inertia forces were calculated on the assumption that the steam pressure in the cylinder equaled 0.33 normal working pressure, and speed in m.p.h. was equal to the driver diameter in inches. Twenty connecting-rods of material of a yield point of 40 tons (2240 lb.) per sq. in. or over gave the following average stresses per sq. in. Tensile at small end, 10,790 lb.; vertical, 11,350 lb.; horizontal, 12,950 lb. Values of (length/radius of gyration) ranged from 40 to 80 vertically and 120 to 205 horizontally. Inertia stresses are reduced by the lighter sections permissible with alloy steel. Average inertia stress in the alloy steel side rods of locomotives on seven different railroads was 8500 lb. per sq. in., while the average inertia stress in twelve rods of class C carbon steel was 14,100 lb. per sq. in., with a lowest value of 9,600 lb.

Straight Driving Axles and Crank-pins.—Stresses are due to the resultant of spring load and piston thrust, combined with the twisting moment to obtain the equivalent bending moment. An analysis of the stresses in a 4–6–0 locomotive of the Southern Ry. (British) showed a combined bending stress on the axle of 203,840 lb. per sq. in., a bearing pressure due to spring load of 159 lb. per sq. in., and a bearing pressure due to piston load of 1030 lb. per sq. in. The combined bending moment on the crank-pin was 7,765,760 in.-lb., and the equivalent bending moment of the force necessary to turn one wheel was 14,564,480 in.-lb. The stress in the crank-pin was 13,960 lb. per sq. in. and the bearing pressure 2000 lb. per sq. in. The average bearing pressure on the main axle of 30 locomotives built between 1922 and 1932 was 1730 lb. per sq. in., with upper and lower limits of 2120 and 1415 lb.; bending stress averaged 16,650 lb. per sq. in., with upper and lower limits of 22,100 and 7850 lb. per sq. in.

Single Throw Crank-axles for 3-Cylinder Engines.—The stress on the crank-pin is much lower than in a 2-cylinder engine. The pin diameter is much the same, but the piston thrust is usually much greater in a 2-cylinder engine. An analysis of stresses and bearing pressures on the inside crank-pins and journals of 3-cylinder engines on six railroads gave the following values, lb. per sq. in.: Bearing pressure on outside pin, maximum, 1820; minimum, 1495; average, 1667; bending stress on outside pin, maximum, 19,667; minimum, 7840; average, 15,870; bearing pressure on inside pin, maximum, 1140; minimum, 757; average, 999; bending stress on inside pin, maximum, 10,640; minimum, 6832; average, 8769; spring load bearing pressure on journal, maximum, 175; minimum, 150; average, 166; piston thrust load on journal maximum, 1100; minimum, 915; average, 970. Average crank-pin bearing pressure of 20 2-cylinder engines was 1320 lb. per sq. in.

Coupled Axle Boxes.—Bearing pressures per sq. in. of projected area are higher than for main driving axles on the same engine. A study of axle boxes on over 50 engines, covering all classes of service, gave average bearing pressures, lb. per sq. in., of: Fast passenger engines, 177; mixed traffic engines, 195; freight and switching engines, 208.

Truck, Trailer and Tender Axle Boxes.—An analysis of 30 truck axle boxes in fast passenger and mixed traffic showed an average bearing pressure of 150 lb. per sq. in. of projected area, and in freight service of 155 lb. Fifteen trailer axle boxes showed an average bearing pressure of 194 lb. per sq. in., and 15 tender axle boxes showed 230 lb. These correspond closely to the values recommended by the Am. Ry. Assoc. See p. 14-26.

Crosshead Guides.—Stiffness, rather than bending stress, is the criterion of good design. Table 18 shows the proportions, stresses and deflections of the guides in a variety of engines. Deflection has been calculated at point of maximum connecting-rod thrust, and in no case does maximum thrust coincide with mid-point of crosshead travel. The best position for the outer support is near mid-stroke.

Crossheads and Crosshead-pins.—Triple guides have been adopted in a number of locomotives to reduce slipper pressure. In a decapod on the Penna. R. R., triple bearing faces, one above the other, are necessary to take the thrust in the forward direction of 9.85 tons, resulting from a piston load of 81.5 tons. Four-bar arrangements are subject to high stresses and pressures, but the variation between top and bottom limits is not as wide as with 1- and 2-bar designs. Average values of slipper pressure, lb. per sq. in., are:

	One Bar	Two Bars	Three Bars, Forward	Three Bars, Backward	Four Bars
Average.......	69	66	62	101	85
Upper.........	89	85	78	128	93
Lower.........	58	40	55	68	79

Crosshead-pin bearing pressures per sq. in. of projected area may be taken as 5000 lb. In 25 designs the highest value, lb. per sq. in., was 6400; lowest 4050; average, 4800. In these same pins, bending stress, lb. per sq. in., was: highest, 10,600; lowest, 5250; average, 6620. The pin was considered as a beam supported at the ends and uniformly loaded, bending moment being taken as $WL/8$.

Piston-rods.—An examination of 20 class D carbon steel rods showed an average tensile stress at the center of 5800 lb. per sq. in., with 9200 lb. at the smallest cross section. Twelve alloy steel rods showed an average stress of 7325 lb. per sq. in. at the center, and 9850 lb. per sq. in. at the smallest cross section.

6. LOCOMOTIVE DETAILS

The following locomotive details are condensed from the standards and recommended practice adopted by the American Railway Association in the year following each title.

AXLES (1927).—Locomotive Driving Axles. Dimensions are shown in Fig. 9 and Table 19. Axles 9 1/2 in. diam. or over may be hollow or solid, flush with hubs or project 1/8 in. beyond hub faces, in which case increase dimension H by 1/4 in. Dimensions B, D and F apply to main axle; on other axles, length of wheel fits and journals depend on dimensions of main axle. Diameter of axles between journals may equal diameter of journal less limit of wear. Length of journals = length of bearing + 1 in. Engine frames should be so spaced that central plane of journal coincides with central plane of frame. Allowable bearing pressure, lb. per sq. in., are: passenger engines, 175; freight and switch engines, 200.

Formula for Calculating Stress in Main Axle.—Let A = crank radius, in.; B = 1/2 difference in distance between cylinder center lines and frame center lines, in.; $C = \sqrt{A^2 + B^2}$; P = piston thrust, lb.; S = fiber stress, lb. per sq. in.; Z = section

Table 18.—Stresses in Crosshead Guides

(1 Ton = 2240 lb.)

No. of Bars	Cyls.	Thrust on Bar, tons	Length between Supports, in.	Bending Moment, in.-tons	Width of Bar, in.	Depth of Bar, in.	Bending Stress, tons per sq. in.	Deflection of Bar, in.
1	Outside	3.01	58.0	42.8	6.0	4.0	2.68	0.028
1	"	4.07	65.75	66.7	5.75*	6.3	2.04	.0166
1	Inside	3.14	56.0	43.7	6.0	3.5	3.57	.0398
2	Outside	3.12	48.0	24.75	5.5	3.0	3.0	.026
2	"	2.205	39.5	12.7	4.5	2.25	3.35	.0253
2	"	3.45	46.25	24.5	5.5	2.75	3.53	.0295
2	"	3.45	55.75	45.6	5.0	3.75	3.89	.0383
3	Inside	3.38	52.5	43.0	7.5†	2.25	6.8	.098
4	"	4.57	57.0	64.0	3.0	3.0	7.1	.097
4	"	3.78	39.0	32.8	3.25	2.375	5.35	.0402

* I-section bar. † Top bar, forward direction.

modulus of circular or annular section of diameter A, Table 19, minus limit of wear. Then $S = P(B + C)/2\,Z < 23{,}000$ for steel axles.

Table 19.—Dimensions of Locomotive Driving Axles

Bearings, in.	Dimensions, in. (See Fig. 9)										Max. Journal Load, lb.	
	A	*B**	*C*	*D**	*E*, min.	*F**	*H*	*I*†	*L*	*R*	Passenger	Freight and Switch
7 × 9	7	10	7 1/2	7	6 1/2	34	68	....	54	3/4	11,000	12,600
8 × 10	8	11	8 1/2	7 1/2	7 1/2	32	69	....	54	3/4	14,000	16,000
9 × 12	9	13	9 1/2	7 1/2	8 1/2	28	69	....	54	3/4	18,900	21,600
10 × 12	10	13	10 1/2	8	9 1/2	28	70	2	54	3/4	21,000	24,000
11 × 13	11	14	11 1/2	8	10 1/2	26	70	3	54	3/4	25,000	28,600
12 × 13	12	14	12 1/2	8	11 1/2	26	70	3	54	3/4	27,300	31,200
13 × 14	13	15	13 1/2	8	12 1/2	24	70	4	54	3/4	31,800	36,400
14 × 14	14	15	14 1/2	8	13 1/2	24	70	4	54	3/4	34,300	39,200

* Apply to main axles. Length of wheel fits and journals on other axles to be governed by corresponding dimension of main axles. † Axles may be solid or bored to diameter shown.

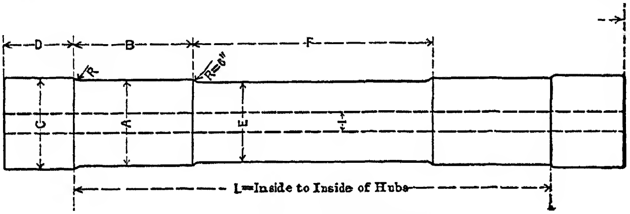


Fig. 9. Locomotive Driving Axle

Engine Truck Axles.—Dimensions are given in Fig. 9 and Table 20. Axles 7 in. diam. and larger may be solid or hollow. Allowable bearing pressures, lb. per sq. in., are: Passenger engines, 160; freight engines, 180. Center plane of journal coincides with center line of frame.

Table 20.—Dimensions of Engine Truck Axles

Bearing Size, in.	Dimensions, in. (See Fig. 9)										Max. Journal Load, lb.	
	A	*B*	*C*	*D*	*E*, min.	*F*	*H*	*I*	*L*	*R*	Passenger	Freight
5 × 9	5	10	5 1/4	7 3/16	4 1/2	32	66 1/8	....	52	1/4	7,200	8,100
6 × 10	6	11	6 1/4	7 3/16	5 1/2	30	66 1/8	....	52	1/4	9,600	10,800
6 1/2 × 11	6 1/2	12	6 3/4	7 3/16	6	28	66 1/8	....	52	1/4	11,400	12,900
7 × 12	7	13	7 1/4	7 3/16	6 1/2	26	66 1/8	....	52	1/4	13,400	15,100
7 1/2 × 13	7 1/2	14	7 3/4	7 3/16	7	24	66 1/8	2	52	1/4	15,600	17,600
8 × 14	8	15	8 1/4	7 3/16	7 1/2	22	66 1/8	2	52	1/4	17,900	20,200

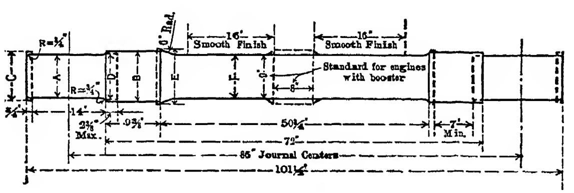


Fig. 10. Trailer Truck Axles

Table 21.—Dimensions of Trailer Truck Axles, Two Wheel Type

Bearing Size, in.	Dimensions, in. (See Fig. 10)						Max. Journal Load, lb.	
	A	*B*	*C*	*D*	*E*	*F**	Passenger	Freight
7×14	7	8 3/8	8 1/4	8 1/4	8 5/8	7 1/2	19,600	20,600
8×14	8	9 3/8	9 1/4	9 1/4	9 5/8	8 1/2	22,400	23,500
9×14	9	10 3/8	10 1/4	10 1/4	10 5/8	9 1/2	25,200	26,500

* Applies to non-booster axles only, rough turned between wheel fits.

Trailer Truck Axles.—Dimensions are given in Fig. 10 and Table 21. Outlines in dotted lines are optional. The enlarged middle section is standard for engines with booster. Allowable bearing pressures, lb. per sq. in., are: passenger engines, 200; freight engines, 210.

AXLE AND CRANK-PIN MOUNTING PRESSURES (1931) recommended for mounting driving and trailing axles and crank-pins are: Cast-iron wheel centers, 10 tons per inch of axle or pin diameter; cast-steel wheel centers, 15 tons per inch of diameter; allowable variation 10% in both cases.

CENTERS AND KEYS.—Dimensions for centers and keys for all axles are given in Fig. 11.

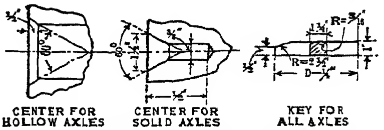


Fig. 11. Axle Center and Key Dimensions

CONNECTING- AND SIDE-RODS (1923).—**Main rods**, Fig. 12, should have tapered fluted section, flanges as narrow as consistent with good design, a setting-up wedge in front of crank-pin, with single-nutted bolts, riveted over, one grease cup, composition bronze bearings at each end, and an approved adjusting feature at front end. Rods, straps and details should be of good material and as light as possible.

Side-rods, Fig. 13, in general, should be of the same construction; the main parallel connection may have a floating bushing. Ample side clearance at knuckle pins, solid oil cups, and single or castle nut for knuckle pins, with cotter or lock nut are essential. Tapped plugs, top and bottom, prevent bushings turning. Knuckle pins should be hollow. Offsets or special shapes are undesirable, except to avoid interference. Collars on inside of bushings are desirable. No welding of any kind may be used to fasten keys, wedges, etc.

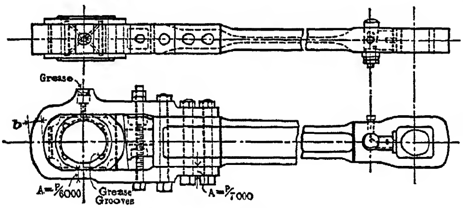


Fig. 12. Main Rod

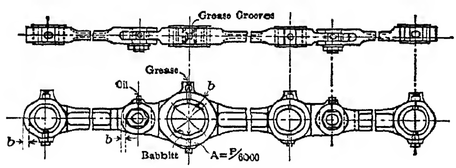


Fig. 13. Side Rods

The following notes on stresses on rods are from *Proc.* Am. Ry. M. M. Assoc., 1911. Stresses in main rods are tension and compression, due to piston pressure and inertia of reciprocating weights, and bending due to centrifugal force acting vertically. Stresses in compression are always greater than in tension, and those due to cylinder pressure are greater than those due to reciprocating weights. Stresses in side-rods are tension and compression, due to part of the piston pressure transferred through the main crank-pin, and to the rod sliding one or more drivers of unequal diameter, or when passing curves. Bending stresses also exist, due to centrifugal force acting vertically. The limit of the force to slide drivers is the coefficient of friction between wheel and rail, which may be taken as 0.3. Each rod is assumed to be able, when starting, to slide the *pairs* of drivers it rotates; at running speed it need slide drivers on one side only. Calculations of rods may be based on 375 r.p.m. for fast freight and passenger engines, and 420 r.p.m. for fast passenger engines. For Mallets a special figure should be used.

Table 22.—Standard Axles for Locomotive Tenders

(American Railroad Association)

Letter	Capacity, lb.	Journal Diameter and Length, in.	Diameter, in.		Length	
			At Wheel Seat	At Center	Center to Center of Journals	Overall
A	15,000	3 3/4 × 7	5 1/8	4 1/4	6′ 3″	6′ 11 1/4″
B	22,000	4 1/4 × 8	5 3/4	4 3/4	6′ 3″	7′ 0 1/4″
C	31,000	5 × 9	6 1/2	5 3/8	6′ 4″	7′ 2 1/2″
D	38,000	5 1/2 × 10	7	5 7/8	6′ 5″	7′ 4 1/2″
E	50,000	6 × 11	7 5/8	6 7/16	6′ 6″	7′ 6 3/4″

Let A, a, = respectively, area, sq. in., and depth, in., of section; b = width of section, in.; B = unit load, lb.; C = maximum unit compressive stress, lb.; C_1, C_2 = respectively, maximum unit compressive stress for transverse and vertical bending; c, c_1, c_2 = coefficients (see Table 23); D = diam. of drivers, in.; d = cylinder diam., in.; E = modulus of elasticity; F = centrifugal force, lb.; G = weight of rod, lb.; l = length of rod, in.; M = bending moment, in.-lb.; m = section modulus = m_h, m_v, axis horizontal and vertical respectively; n = constant = 1 for both ends round; = $^1/_4$ for both ends fixed; p_b = maximum boiler pressure, lb. per sq. in.; p = maximum piston pressure, lb. per sq. in.; P = maximum compressive stress at end of rod, lb. per sq. in.; P_m = maximum end load, lb.; R = length of crank, in.; r = radius of gyration, in., = r_v, r_h for axis vertical and horizontal, respectively; s = offset of rod, in.; S = stress, lb. per sq. in.; T = ultimate tensile strength of steel; W = weight of reciprocating parts.

For fast freight and passenger engines, if $p < 4\,WR$, $4\,WR$ should be substituted for p. The rod body should be considered as a strut with load p or $4\,WR$, and as a beam subject to bending due to whip at high speeds. Main rods usually are tapered, and often fluted. Calculation should be based on the value of A at the center, and $A >$ $(P_m/10{,}000)$. Stresses are computed by the column formula $C = B/\{1-(nB/\pi^2E)(l/r)^2\}$. With C = 10,000, $(l/r) \gtreqless 80$ when neutral axis is vertical, and $(l/r) \gtreqless 160$, with neutral axis horizontal. With neutral axis vertical, $r = b\sqrt{12}$; horizontal, $r = a\sqrt{12}$. Substituting these values in $(l/r) \gtreqless 80$ and $(l/r) \gtreqless 160$, respectively, then l = 23, b = 46 a, whence if l, center to center of pins, < 23, $(l/r) < 80$, and if $l < 46\,a$, $(l/r) < 160$.

In Offset Rods, where the vertical center lines of the bearings are not in the same plane or in the plane of the center of the body, a bending stress is created equal to $P_m\,s/m_v$, which increases stress in the main body.

Knuckle Joints, on rods of 3-coupled and 4-coupled engines, are flexible both horizontally and vertically. If drivers on one side are not in perfect alignment, slight bending stresses occur, in addition to the compression stresses. When knuckle joints are flexible both horizontally and vertically, $l/r \gtreqless 110$ instead of 160.

Bending Stress Due to Whip at High Speed is at a maximum at 0.6 l from crosshead end in main rods, and at the center in side rods. The section at these points, therefore, is the governing section.

Simplified Formulas may be used for checking or for approximate results, within a small percentage of those found by the accurate formulas. These are based on centrifugal force F and on bending moment M. For side rods $M = 0.125\,Fl$, and for main rods $M = 0.064\,Fl$. $G = 0.2833\,Al$, and $GR = 0.2833\,AlR$, where 0.2833 = weight of 1 cu. in. of steel. The following relations hold:

	265	325	375	420
	$2GR$	$3GR$	$4GR$	$5GR$
M (side rods) = ...	$0.071\,Al^2R$	$0.106\,Al^2R$	$0.142\,Al^2R$	$0.177\,Al^2R$
M (main rods) = ...	$0.036\,Al^2R$	$0.055\,Al^2R$	$0.073\,Al^2R$	$0.091\,Al^2R$

The stress due to whip, $S_w = M/m$, should be added to the maximum stress due to end load; the sum should not exceed $^1/_6$ ultimate tensile strength of the steel.

Checking Formulas. For main rods, $A \geqq P/10{,}000 = 0.7854\,d^2pb$; for side rods, $A = 0.15\,WD/R$. For rods with knuckle pins, flexible transversely,

$$C_1 = (P/A)/\{1 - (Pl^2/675{,}000{,}000\,A)\}.$$

For all other rods, $C_1 = (P/A)/\{1 - (Pl^2/1{,}200{,}000{,}000\,Ar_v^2)\}$.

For vertical bending, all rods, $C_2 = (P/A)/\{1 - (Pl^2/300{,}000{,}000\,Ar_h^2)\}$.

For rods without offset, use the value of C_1 or C_2 that is larger as the value of the stress S, lb. per sq. in., in the rod. $S = c_2(l^2r/b) + c_1(P/A)$. $S < {}^1/_6\,T$ if $l < 46\,a$ or $23\,b$, and if $A > P/(^1/_8\,T)$. For offset rods, use the value of C_2, or of $C_1 + (Ps/m_v)$ that is larger, for the value of S. $S = c\,(Al^2r/m_h) + c_1\,P(1/A) + (s/m_h)$.

Allowable stresses on various sections of rod ends are shown in Figs. 12 and 13, except at points marked b. If minimum areas of the two members differ, take A as double the area of the smaller. Minimum areas at points b are: For main rods, $A = PX/30{,}000\,b$; for side rods, $A = PX/60{,}000\,b$, where X = average diam. of eye or spread of jaws.

Table 23.—Values of c, c_1 and c_2

Revolutions per Minute =		265	325	375	420
Main rod	c	0.036	0.055	0.073	0.091
	c_1	.500	.500	.400	.300
	c_2	.22	.33	.44	.55
Side-rod	c	0.071	0.106	0.142	0.177
	c_1	.500	.500	.500	.500
	c_2	.43	.64	.85	1.06

Knuckle Joints for Side-rods (Fig. 14).—Material: For knuckle-pins and rods, medium-grade forged steel A.R.A. specifications; for optional taper key, tire steel; for rod bushing, hard bronze or case-hardened steel. Bronze bushings may float in the rod. Hollow boring of knuckle joint pin is optional.

Bearing Pressures, lb. per sq. in., on knuckle joint pin are: $T/DL \leqq 4000$; $T = 0.3\,WR/r$, where T = thrust, lb.; W = weight on pair of drivers, lb.; R, r = respectively, radius of driving wheel and crank, in.; D and L are pin dimensions. See Fig. 14. Diameter of the threaded part of the pin is:

When D =	T =
3 in.	2 1/2 in.
4 in.	3 in.
5 in.	3 in.
6 in.	3 1/2 in.
7 in.	3 1/2 in.

See also Checking Formulas for Main and Side Rods, p. 14–28.

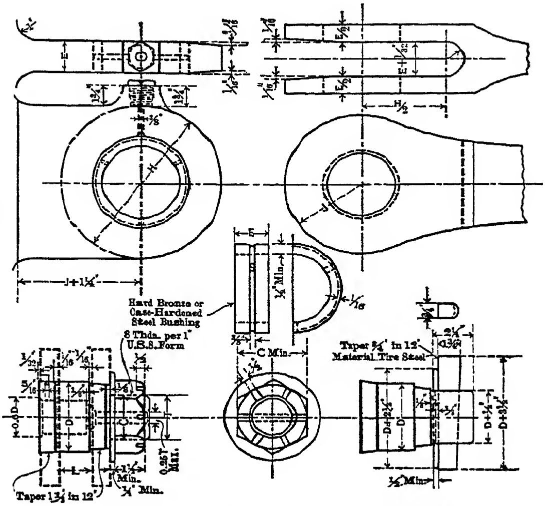


FIG. 14. Knuckle Joints and Knuckle Pins

CRANK-PINS (1928).—The designs of main crank-pins, Fig. 15, and other crank-pins, Fig. 16, are for steel, ultimate tensile strength, 80–85,000 lb. per sq. in. (A.R.A. material specifications for axles, shafts, and forgings). Determining factors of main pin design, depending on size of engine and power required, are piston thrust and location of center line of cylinders in relation to hub face of wheel, *i.e.*, dimensions L, Fig. 15. Following are formulas for fiber stress, bearing pressures, thrust, etc.:

Main Crank-pins.—$f = TL/S < 16{,}000$ lb.; $P = T/(A \times B)$; $p = t/(C \times D)$; where f = fiber stress, lb. per sq. in.; T = piston thrust, lb.; t = side rod thrust, lb.; P = bearing pressure, main rod bearing; p = bearing pressure, side-rod bearing, lb. per sq. in.; S = section modulus = $0.098\,C^3$ for solid pins; = $0.098\,(C^4 - E^4)/C$ for hollow pins; for other symbols see Fig. 15. P and p should not exceed 1600 lb. per sq. in., and $A \leqq B$. Thrust on side-rod bearing equals combined thrust on all coupled crank-pins except main pin. Hollow pins are optional, and of the following dimensions:

C, in.	6–6 1/4		9–9 3/4	10–10 1/2	10
$E_{max.}$,	2	2 1/2	3 1/2	4	4 1/2
F, in.	1	1	2		2

Other Crank-pins.—$f = t \times 0.5\,B/S < 10{,}000$ lb.; $t = 0.3\,WR/r$; $p = t/(A \times B) < 1600$ lb.; $S = 0.098\,A^3$; where W = weight on pair of drivers, lb.; R = radius of driving wheel, in.; r = radius of crank, in.; other notation as for main pins or as in Fig. 16.

Either double or castle nuts may be used on the male threaded end with collar; double nuts are preferred. For front driving wheels or where space is limited, the designs Fig. 16*b* or *c* may be used; the latter is stronger, but the former is more generally used.

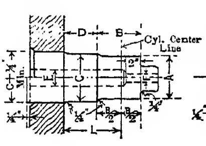


FIG. 15. Main Crank-pin

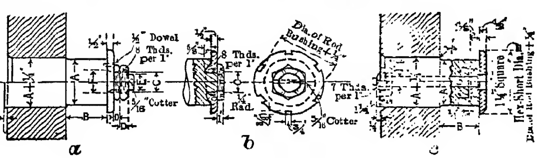


FIG. 16. Coupled Wheel Crank-pins

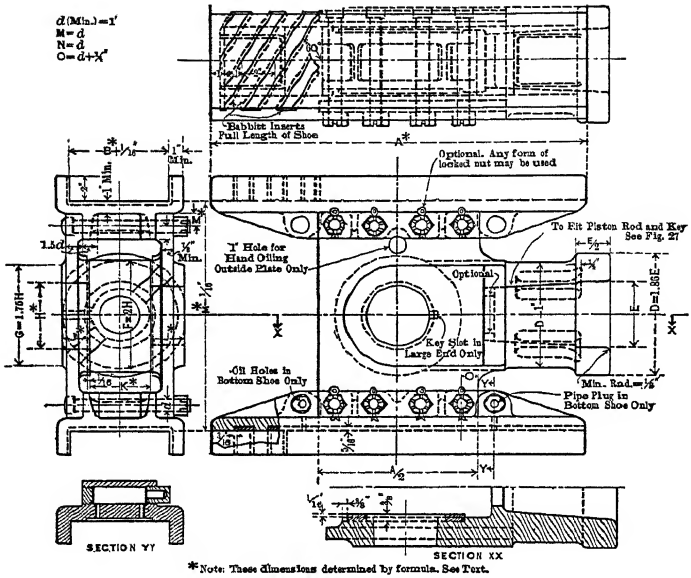


FIG. 17. Alligator-type Crosshead

CROSSHEADS AND GUIDES (1929).—Alligator-type crossheads (Fig. 17) are recommended. Dimensions of crosshead depend mainly on dimensions of the crosshead-pin (see below) and of the guides. Both are determined by allowable bearing pressures; guide dimensions also depend on other dimensions of the engine. Width of guide, B, depends on center line distance between engine cylinders and clearance between crosshead and front crank-pin (minimum clearance = 1 in.). Length of crosshead, A, is such as to make bearing pressure on guide < 100 lb. per sq. in. Let Z = bearing pressure,

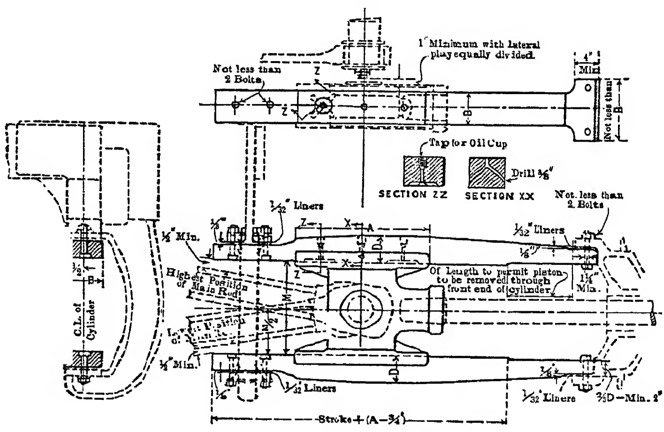


FIG. 18. Crosshead Guides

lb. per sq. in.; P = piston thrust, lb.; T = upward thrust of crosshead, lb.; R = length of main rod, in.; S = stroke, in.; C = vertical distance from plane of driving axle to center line of cylinders, in. Then $Z = T/(A \times B) < 100$; $T = P(^1/_2 S + C)/R$, whence dimension A can be found. Spread of guides, M, depends on clearance between main rod and the end of the guides, at the highest and lowest positions of the rod (see Fig. 18); $^1/_2$ in. is minimum clearance. Dimensions H, J and K (Fig. 17) depend on the dimensions of the crosshead-pin.

Dimension E governs dimensions of piston-rod boss, and depends on design of the rod. See Piston-heads and Rods, p. 14–34. Taper bolts of minimum diameter d = 1 in., taper $^1/_{16}$ in. in 12 in., driven into reamed holes and secured with castle nuts and split cotters, fasten shoes to crosshead. Crossheads are cast steel (A.R.A. specification, grade B), and shoes are cast-iron (A.R.A. specification for cylinders); for heavy fast passenger engines they may be medium bronze without babbitt inserts (A.R.A. specifications for bronze bearings). Bolts are mild steel.

Maximum depth of guide bar, D, $= \sqrt{6\ TI/4\ fB}$, where I = overall length of guide, in.; B = width of guide, in.; f = fiber stress, lb. per sq. in. = 8000 max. Lubrication is through three oil holes in top guide, one at crosshead mid-travel, one 6 in. back of forward edge of crosshead in extreme rear position, and one 6 in. ahead of rear edge of crosshead in extreme forward position; the lower guide is lubricated by oil holes in the lower shoe. Upper and lower shoes are fitted with babbitt inserts, 1 in. wide, dovetailed into the shoe, and set at an angle of 60 deg. to the line of travel of the crosshead. Wear is taken up by shims between guides and supporting lugs on cylinder heads and guide yokes.

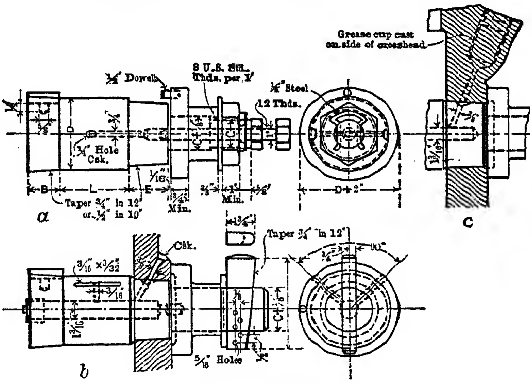


FIG. 19. Crosshead-pins

Crosshead-pins (1929).—Dimensions of pins, Fig. 19, are determined by piston thrust. Let P = piston thrust, lb.; D and L = diameter and length, in., respectively, of crosshead-pin in main rod; S = crosshead-pin bearing pressure, lb. per sq. in. Then $(T/DL) = S < 4800$, D or L being assumed. Dimension C depends on D as follows:

D, in. =	3	5	6
C, in. =	$2\,^1/_2$	3	$3\,^1/_2$

Dimensions B and E are found by formula $T/\{D \times (B + E)\} = S < 5000$; S = crosshead-pin bearing pressure on side plates, lb. per sq. in.; $B = E$, or B (min.) $= E - ^1/_4$ in. Crosshead-pins are forged steel, A.R.A. specification. Pins are held in position by a nut with cotter pin; a taper key, as shown, is an alternative method. Grease lubrication of pins is recommended, but is difficult with a keyed fastening. Fig. 19*b* is a recommended method for oil and Fig. 19*c* a method of grease lubrication for keyed fastenings.

DRIVING AND ENGINE TRUCK BOXES (1928).—Material is cast steel (A.R.A. specification for carbon steel castings, 1923). For driving boxes, dimensions are given in Fig. 20 and Table 24; engine truck boxes in Fig. 21 and Table 25. Details of cellar design and fitting and grooving of brasses have not been standardized (1934). Fig. 21 shows construction for 2-wheel engine trucks using semi-elliptical plate springs. Modified as in Fig. 22 it may be used on 4-wheel front trucks of passenger engines.

Table 24.—Dimensions of Driving Boxes

Journal	Dimensions, in. (See Fig. 20)								
A	*B*	*C*	*D*	*E*	*F*	*G*	*H*	*K*	*L*
7	1 3/4	1 3/8	1 3/4	1 1/4	2*A*–1 in.	2	*A*+1/2 in.	*A*+63/128	9
8	1 3/4	1 3/8	1 3/4	1 1/4	2*A*–1 in.	2			10
9	1 3/4	1 3/8	1 3/4	1 1/4	2*A*–1 in.	2			12
10	2	1 1/2	2 1/8	1 1/2	2*A*–2 in.	2 3/4			12
11	2	1 1/2	2 1/8	1 1/2	2*A*–2 in.	2 3/4			13
12	2 1/4	1 5/8	2 1/2	1 3/4	2*A*–3 in.	3 1/2			13
13	2 1/4	1 5/8	2 1/2	1 3/4	2*A*–3 in.	3 1/2			14
14	2 1/4	1 5/8	2 1/2	1 3/4	2*A*–3 in.	3 1/2			14

Brass plugs *P*, 1 in. diam., taper 1/16 in. in 12 in., driven from inside and riveted.

DRIVING AND TRAILING WHEELS FOR LOCOMOTIVES (1907. Revised 1928). —**Driving Wheels.**—Cast-steel driving-wheel centers should be uncut and shrinkage slots omitted. If total weight of locomotive permits, rims should be cast solid without cores, and have full bearing on tires; sectional area should be approximately 45% that of the tire when new. Hubs and solid rim sections, Fig. 23*b* and *c*, are recommended for all designs where weight and counterbalance limits permit. Cored rim, Fig. 23*f*, *g*, *h*, and elliptical spoke section, Fig. 23*j*, *k*, are alternatives. Fig. 23*j*, *k*, is an optional arrangement of reinforcing webs between spokes. Solid counterweights are recommended, cast integral with driving wheel center wherever the desired counterbalance can be so obtained. Small lead pockets, Fig. 23*f*, permit closer adjustment than solid counterweights. Solid cast counterweights usually are satisfactory on all except main driving wheels. They sometimes can be used on main wheels of relatively large diameter.

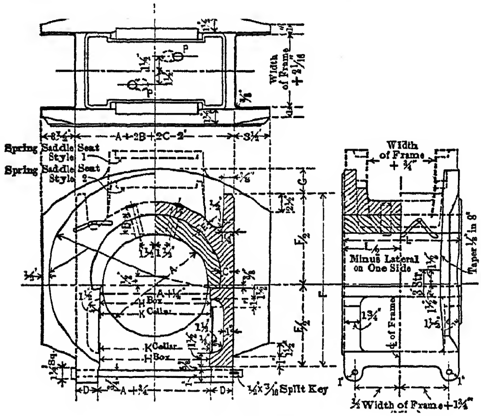


Fig. 20. Driving Box

Trailer Wheel.—Material recommended is cast steel (A.R.A. specification, 1918, for carbon steel castings); hubs, rims and spokes are similar in design to those of driving wheels. The relation of hub bore and journal dimensions, and of whee diameter and number of spokes is as follows (see Fig. 24). Shrinkage tires should be used in all cases.

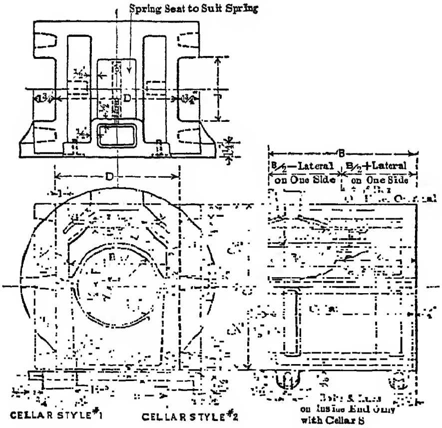


Fig. 21. Engine 2-Wheel Truck Box

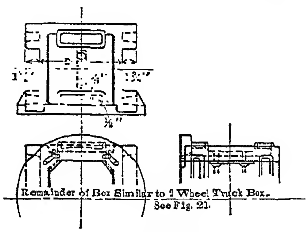
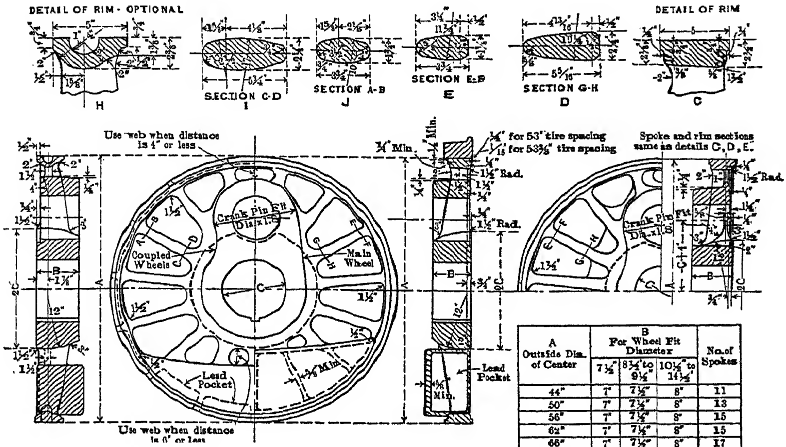


Fig. 22. Engine 4-Wheel Truck Box



A Outside Dia. of Center	B For Wheel Fit Diameter 7½"	8¾" to 9½"	10½" to 14½"	No. of Spokes
44"	7"	7½"	8"	11
50"	7"	7½"	8"	13
56"	7"	7½"	8"	15
62"	7"	7½"	8"	15
66"	7"	7½"	8"	17
72"	7"	7½"	8"	17
74"	7"	7½"	8"	19

FIG. 23. Locomotive Driving Wheels

Referring to Fig. 24, the following relations hold

Wheel Bore, *A*	Journal Size, in.
8 3/8	7 × 14
9 3/8	8 × 14
10 3/8	9 × 14

Wheel Diam. *B*, in.	No. of Spokes
33 3/4	9
37	11
40	11
43	13

ECCENTRIC CRANKS

CRANK ARMS (1931).—Designs in Fig. 25 are for forged medium steel (A.R.A. Standard specification for axles, shafts, and other forgings), but are suitable for electric cast steel. Forged eccentric cranks should be machined all over. Bolts are drive fits in straight reamed holes. Channel milling is optional in forged steel cranks. Dimension *D* is made to suit crank-pin. The bronze ring is made to suit main rod, and may be omitted if not required by main rod design. The eccentric

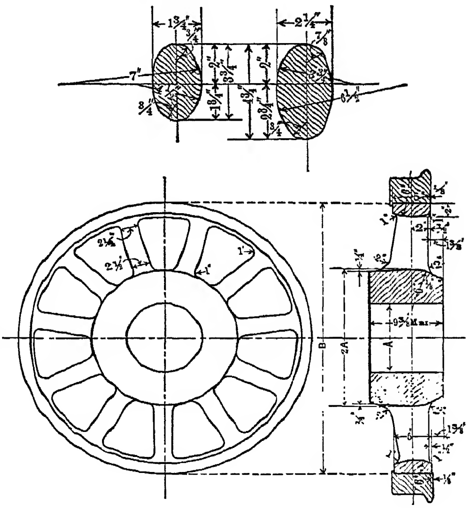


FIG. 24. Locomotive Trailer Wheels

Table 25.—Dimensions of Engine Truck Boxes

Journal *A*	Dimensions, in. (See Figs. 21, 22) *B*	*C*	*D*	*E*, min.	*F*	*G*	*H*	*I*	*J*
5	9	11 1/2	7 1/2	6 1/2	3 7/8	2 5/8	6	1	4 1/16
6	10	12 1/2	9	7	4 1/2	3 5/8	7	1	4 9/16
6 1/2	11	13 1/2	9 1/2	7 1/2	4 3/4	4 1/8	7 1/2	1 1/8	5 1/16
7	12	14 1/2	10	8	5 1/8	4 5/8	8	1 1/4	5 9/16
7 1/2	13	15	11	8	5 3/8	5 1/8	8 3/4	1 1/4	6 1/16
8	14	15 1/2	11 1/2	8	5 3/4	5 5/8	9 1/4	1 1/4	6 9/16

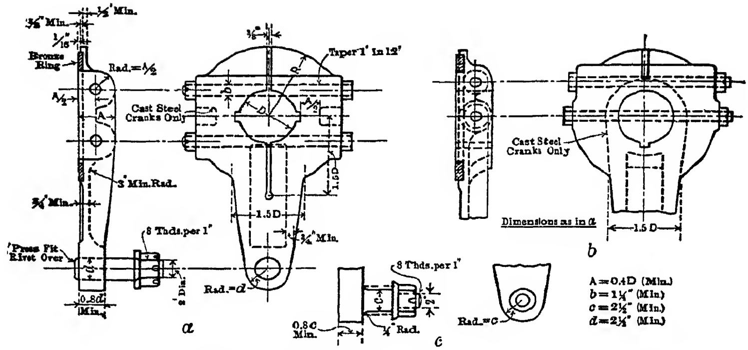


FIG. 25. Eccentric Cranks and Crank Arms

pin is made to suit eccentric rod bearing, and may be forged solid with the crank arm, or made separately, pressed in, and riveted over.

FRONT AND BACK CYLINDER HEADS (1929).—Front and back heads for cylinders up to 32 in. diam., and for boiler pressures up to 300 lb. per sq. in. are shown in Fig. 26. Material is grade B cast steel, A.R.A. specification. Number of studs, $N = A/a$; $A = {}^1/_4\pi D^2Pc/T$; where A = total cross-sectional area at root of thread, of all studs, sq. in.; a = cross-sectional area of one stud; P = boiler pressure, lb. per sq. in.; D = mean diam. of cylinder head joint, in.; c = factor of safety = 8; T = ultimate tensile strength of stud material, lb. per sq. in. Pitch of studs $\leqq 3\,{}^1/_2 \times$ diam. of stud.

PISTON-HEADS AND RODS (1928).—Piston-heads may be either wrought or cast steel (minimum tensile strength, 60,000 lb. per sq. in.). Fig. 27 and Table 26 are designs of piston-head, a solid one-piece head and a two-piece head with the bull ring bolted or riveted to it. Arrangement and style of packing and bull rings, and methods of facing their peripheries with anti-friction metal are optional. Bearing pressures should not exceed 5 lb. per sq. in. for passenger engines, or 6 lb. per sq. in. for freight engines. Bearing pressure = W/V, where W = (weight of piston-head exclusive of packing rings + weight of piston-rod nut + ${}^1/_2$ weight of rod, lb.); V = projected area of head, sq. in., exclusive of packing rings, bearing on bottom of cylinder.

Piston-rods may be of annealed carbon steel, 80,000 lb. per sq. in. minimum tensile strength (A.R.A. specifications, revised 1926), for medium carbon steel forgings. Heat-

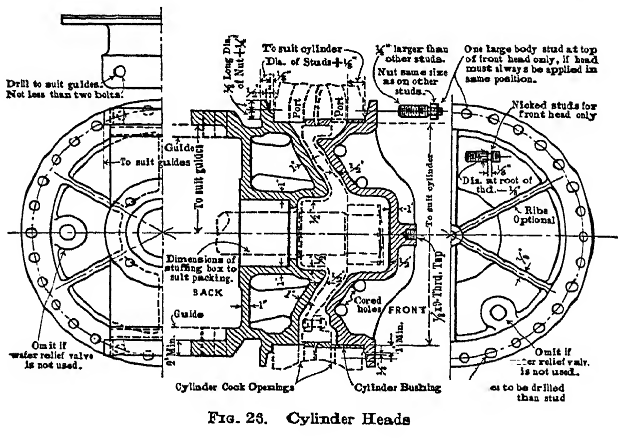


FIG. 26. Cylinder Heads

Table 26.—Dimensions in Inches of Piston-heads (See Fig. 27)

Based on 225 lb. per sq. in. boiler pressure and steel of 60,000 lb. per sq. in. ultimate strength

Cyl. Diam., in.	H	J	K	L
23	6 3/4	5	1 1/4	1
24	6 3/4	5	1 1/4	1
25	6 3/4	5	1 1/4	1
26	7	5	1 1/4	1
27	7	5	1 1/4	1
28	7 1/2	5 1/2	1 7/16	1
29	7 1/2	5 1/2	1 7/16	1
30	8	5 1/2	1 9/16	1
31	8	5 1/2	1 9/16	1

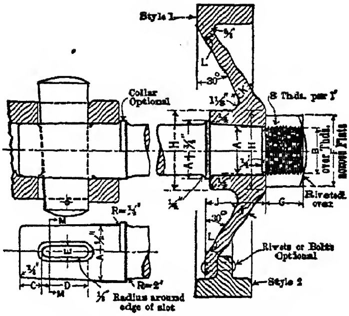


Fig. 27. Piston-head and Rod

treated steel (A.R.A. specifications for forgings of either quenched and tempered carbon, or quenched and tempered alloy steel) is preferable.

Diameter of rod, of threaded end, dimensions of slot and distance from slot to end of rod are functions of the piston thrust. They are determined by the formulas, $A = \sqrt{P/5183}$; $B = 0.163 + \sqrt{P/7068}$; $C = P/(10{,}000 \times \text{mean diam.})$; $D \times E = P/20{,}000$; Area $MM = P/9500$; where P = piston thrust, lb. The other symbols are given in Fig. 27.

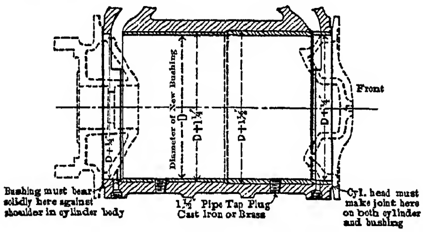


Fig. 28. Cylinder Bushings

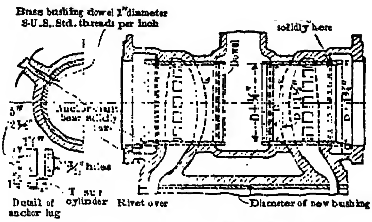


Fig. 29. Steam Chest Bushing

Dimensions of the nuts depend on the value of B as above determined, and are as follows:

B, in. =	2	2 1/4	2 1/2	2 3/4	3	3 1/4	3 1/2	3 3/4	4	4 1/4	4 1/2	4 3/4	5
G, in. =	2	2 1/4	2 1/4	2 1/2	2 1/2	2 3/4	2 3/4	3	3	3 1/4	3 1/2	3 3/4	4
F, in. =	3 1/8	3 1/2	3 7/8	4 1/4	4 5/8	5	5 3/8	5 3/4	6 1/8	6 1/2	6 7/8	7 1/4	7 5/8

Piston-rod and Crosshead Key Tapers, Fig. 27, are respectively: Rod (on diameter), 3/4 in. in 12 in.; key, 1/2 in. in 12 in.; or rod, 1/4 in. in 5 in.; key, 3/8 in. in 12 in.

STEAM CHEST AND CYLINDER BUSHINGS (1930).—New cylinders are fitted with bushings of close-grained gray iron (A.R.A. standard material specification for locomotive cylinders). See Figs. 28–30. If d = diam. of cylinder or steam chest bore, in., and D = outside diam. of bushing, in.,

$$D = d + 0.00025\,d \text{ for cylinders,}$$

$$\text{and } D = d + 0.000375\,d \text{ for steam chests.}$$

Bushings are cold pressed into cylinders and steam chests, and bored in line with each other after insertion. Steam edges of ports in steam chest bushings are machined. Steam ports in cylinder bushings are formed by drilling a series of holes and chipping out bridges between, after bushing is in place, edges being filed smooth. Figs. 29 and 30 show two methods of anchoring bushings in steam chests. Fig. 30 provides for long valve travel with minimum length of steam chest.

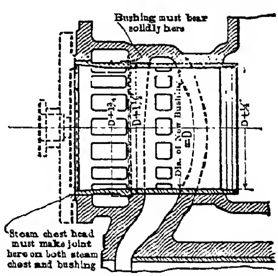


Fig. 30. Steam Chest Bushing

STRESSES IN LOCOMOTIVE BOILERS (1915) APPLYING TO NEW CONSTRUCTION ONLY.—**Longitudinal Barrel Seams and Patches.**—*a.* In figuring net section of plate, use diameter of rivet hole. *b.* In figuring rivet shear, use actual diameter of driven rivet. *c.* In figuring stress in plate and shear in rivet if junction of firebox wrapper sheet and barrel is not cylindrical, use maximum diameter. Surfaces subject to bending under pressure must be adequately braced. *d.* When boiler shells are cut for steam domes or manholes, strength of metal in flange and liner shall equal that of metal removed. If separate flange is used at base of dome, its entire net area is to be assumed as reinforcement. If dome sheet is welded vertically and flanged direct to boiler shell, a vertical distance 2 in. from base of flange is assumed as reinforcement, using net area less rivet holes and 28,500 lb. per sq. in. as ultimate tensile strength.

Longitudinal Gusset Braces and Flat Surfaces.—*a.* In diagonal braces, allow for the angularity of the brace. *b.* Use either sectional area of the brace or strength of attachment of brace to shell, whichever is lowest. *c.* In determining strength of gusset braces for supporting back head and tube sheets, select minimum of: 100% of rivet-bearing area; 80% of rivet shear area; 90% of gusset plate area, measured at right angles to longest edge of gusset sheet. *d.* Calculation of stress in gusset braces covers both the section of the plate and strength of fasteners; use lowest net strength. *e.* Flat stayed surfaces are figured with the boundary of unsupported flat surface located at a distance equal to outside radius of flange measured from inside of shell. *f.* No supporting value shall be assigned to the stiffness of flat plates on flat surfaces. *g.* Allow no staying or supporting value for reinforcing plates, as back head liners, but consider them only as mechanical reinforcements. *h.* 2 in. beyond the outer row of flues on tube sheets is assumed to be self-supporting. *i.* Deduct area of dry pipe hole in calculating area to be stayed on front tube sheet. *k.* T-irons or members subject to bending, are calculated without addition for strength of plate; stress in such beam and abutments not to exceed 12,500 lb. per sq. in. Spacing of rivets over supported surface shall conform to specification for stay bolts. No allowance for value of such beams is made in calculating total areas of longitudinal braces attached thereto. *l.* When a number of diagonal stays support a flat surface, the proportion of area allotted to each brace shall be: (entire net area to be stayed) ÷ (entire net area of braces). If any individual brace appears to receive more than its fair proportion of load, it shall be investigated separately as to the area supported by it. *m.* Patches on barrel of a boiler to be designed with longitudinal and circumferential seams at least equal in strength to main longitudinal and circumferential barrel seams. Patches may be applied to flat stayed surfaces with properly designed single-riveted seams without impairing the strength of the sheet.

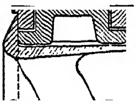


FIG. 31. Piston-valve Ring

Stay Bolts. Radial Stays and Crown Bar Bolts.—*a.* In figuring net area of stay bolts to obtain the stress, area of tell-tale hole shall be deducted. *b.* Area at root of thread must depend on type of thread used. *c.* In determining area for figuring stress on stay bolts, the area of one stay bolt shall be deducted from the rectangle included between any four stay bolts. *d.* In boilers with crown bars supported on firebox side sheets and sling stays, the sling stays are considered as carrying the entire load.

Dry Pipes (1922).—Recommended practice for dry pipes is as follows:

Nominal Size, in.	5	6	7	8	9
Inside Diam., in.	5.047	6.065	7.023	8.071	8.941
Outside Diam., in.	5.563	6.625	7.625	8.625	9.625
Thickness, in.	0.258	0.280	0.301	0.277	0.342

PISTON-VALVE RING shown in Fig. 31, and known as the "L" ring, was recommended by a committee of the Am. Ry. Master Mechanics Assoc. (1916). The spacing, or bull, ring shown with it was also recommended.

EXHAUST NOZZLES usually are circular with a single opening, but for anthracite burning locomotives the double nozzle is favored. For saturated steam the diameter of the single nozzle is about 0.25 × cylinder diameter D. For superheated steam the diameter is about 0.225 D. The combined area of the two openings of double nozzles is made equal to that of the corresponding single nozzle. The net area of opening, with a bridge, is made as large as before the bridge was applied. A nozzle with four projections extending a short distance across the circular opening of the usual type, known as the Goodfellow type, was developed by the Penna. R. R. system at its locomotive testing plant. The projections are of triangular section with the edges turned downward. A considerable increase in locomotive capacity has been effected by means of this nozzle. See *Ry. Age-Gazette*, Apr. 9, 1915.

Multiple-opening exhaust nozzles of enlarged area have had extended application, and provide an effective means of filling a large-diameter stack, thereby reducing the velocity of the gases and the work lost on their discharge to atmosphere. Fig. 32 shows a 6-opening nozzle on the Timken roller-bearing locomotive. The nozzle usually has six thin triangular openings, extending to the center. Width across the top of the nozzle may be as much as 16 in. In this type of nozzle the area of opening may be 40% greater than that of the usual circular nozzle, with no decrease in draft but with considerably lower back-pressure. The decrease in back-pressure gives a greater drawbar pull, but shows no gain in engine efficiency in ordinary service. Two forms of nozzle are used. One has openings extending to the center, the other has a solid center to which a circular plate is bolted. By varying the diameter of this plate the area of the nozzle opening can be adjusted.

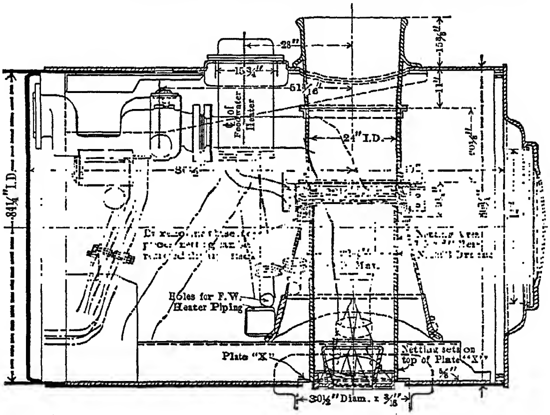


FIG. 32. Front-end Arrangement of Timken Locomotive

LOCOMOTIVE FRAMES (Committee of Am. Ry. Master Mechanics Assoc., 1911). —An approximate formula for the sections of bar frames in either wrought-iron or cast steel is $S = T/C$, where S = sectional area of frame, sq. in.; T = piston thrust = (area of piston $\times$ boiler pressure); C = a constant; see Table 27.

This formula gives sectional areas back of main pedestal, approximately 15% less than similar areas at or ahead of main pedestal. Depth of top rail, ahead of front pedestals, must not be less than that of top rail over front pedestals. The width of frame is made proportional to the weight and power of the locomotive. Very heavy frames, over 5 in. wide, are difficult to make satisfactorily in hammered iron; cast steel seems to be the only suitable material. For locomotives with trailing trucks, a slab spliced to the main frame behind the rear drivers generally is used.

In cross bracing it is recommended that the ties be fastened for the full length of the pedestal jaw, vertically, on the rear pedestal of each driving wheel, or as close an equivalent to this as conditions will permit. The thickness of the bosses on cast-steel cross-tie braces should be not less than $1^1/_2$, preferably 2, times the diameter of the bolt used in fastening.

All bolt heads, where possible, should be next to the castings to insure a full bearing on the bolt. Diameter of frame bolts at thread should be the nearest $^1/_8$ in. to $w/4$, where w = width of frame; the body of the bolt should be made $^1/_{16}$ in. larger. For fastening pedestal jaws together at the bottom, the clip binder or pedestal cap is used more than any other method.

Table 27.—Value of Constant C.

Section at	Sections from Cylinders to Main Pedestals (Including Top Rail Over Main Pedestal)	Back of Main Pedestal
Top of pedestals	2500–2700	2900–3200
Top rail between pedestals	3000–3200	3500–3800
Lower rail between pedestals	4300–4500	5100–5300
Integral single rail at back of cylinder keying lug	1600–1800	....

A cast-steel frame for a 2-10-0 type locomotive built in 1918 for the Penna. Railroad System, has top rail $9\,{}^1/_2$ in. wide and $7\,{}^1/_2$ in. deep between pedestal jaws. Between pedestals the top rail is 7 in. wide, 8 in. deep. The bottom rail is 7 in. wide, 6 in. deep. This frame is 44 ft. 9 in. long; each side weighs 21,225 lb.

VALVE-GEARS.—The Stephenson gear, Fig. 33, is a combination of two eccentrics with a link connecting the extremities of the eccentric rods. By adjusting the position of the link, the valve rod may be put in direct connection with either eccentric or may take intermediate positions between them. The lead decreases as the cut-off is advanced. While it is possible to use the Stephenson as an outside gear, it is not well adapted to such a position, and is generally known as an inside gear.

The Walschaerts gear, Fig. 34, invented by Egide Walschaerts in 1844, allows strengthening the locomotive frame by cross-bracing and is easily accessible from outside the locomotive. It has constant lead and takes its motion from both the crosshead and crank-pin.

The Baker gear, Fig. 35, has a constant lead and variable preadmission.

The Southern gear, Fig. 36, is a modification of the Hackworth type of radial gear. The movement of the valve is obtained from a return crank on the main crank-pin. The eccentric rod is driven from this return crank, and is suspended, near its forward end, from a block which slides in a curved link. The direction of running and point of cut-off are determined by the position of the block in the link.

References.—Valves and Valve Gear Mechanisms, Dalby, Longmans, Green and Co. The Development of Locomotive Valve Gears, J. Dunlop, *Engr.*, London, June 18, July 2 and 16, 1920. Locomotive Valves and Valve Gears, J. H. Yoder and G. B. Waren, D. Van Nostrand Co., 1917. Walschaerts Valve Gear, Amer. Locomotive Co., Bulletin 1018, June, 1916. Method of Laying Out Walschaerts Valve Gear, J. J. Jones, *Ry. Mech. Engr.*, Dec., 1921.

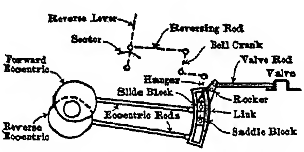


Fig. 33 Stephenson Reversing Gear

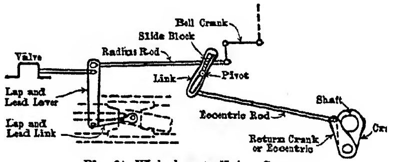


Fig. 34 Walschaerts Valve Gear

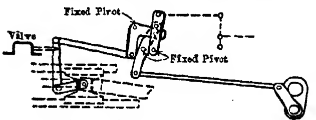


Fig. 35 Baker Valve Gear

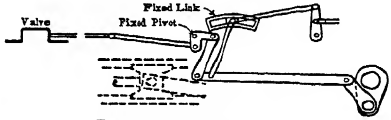


Fig. 36 Southern Valve Gear

Types of Valve Gears Used on American Railroads

POPPET OR LIFTING VALVES for steam admission and exhaust are used extensively on locomotives in Europe. The Lentz and Caprotti are used. A Giesl-Gieslingen (*Ry. Mech. Engr.*, May, 1930) states that the Austrian Railways have over 400 locomotives with poppet valves, and have definitely abandoned the piston valve.

The Lentz valves, Fig. 37, are driven by a light form of Walschaerts valve gear which moves a swinging cam shaft to operate the valves. When the engine is drifting the poppet valves automatically are lifted from their seats and open a connection between the back and front of the piston.

The Caprotti valve, Fig. 38, consists of poppet-type double-seated balanced valves operated by revolving cams driven from one of the driving axles by bevel gears and a central drive shaft. The cams are carried in oil-tight boxes, mounted one above each cylinder. The boxes support the cam mechanism and enclose the oil bath in which cams and rollers operate. The valves are light drop forgings of Cr-Ni steel.

STOKERS (From Committee Report of Am. Ry. Assoc.).—Stokers seldom are used on locomotives of tractive force less than 55,000 lb. Consensus of opinion is that stoker-fired locomotives burn more coal than hand-fired, the percentages of difference ranging from 10 to 40 in favor of hand firing. In these percentages, no account is taken of the advantages resulting from uniform steam pressures and better performance over the division.

In the Street stoker, coal is delivered to the base of a chain-conveyor, which elevates it to a point from which it falls by gravity to distributing nozzles in the back head of the boiler. The Duplex stoker carries the coal up by two screws, on either side of the fire-door. The Hanna stoker elevates the coal by one screw to a point above the fire-door. The Standard stoker feeds coal by a vertical screw through the grate at the rear end of the firebox. These stokers use steam jets to scatter or distribute the coal over the grate. The distributing tubes are fixed in the Street, Duplex, and Standard and movable in the Hanna.

In the Elvin stoker, the coal is distributed by two mechanically-operated shovels or swinging arms operating through the fire-door opening. The Crawford is an underfeed stoker. The coal is fed by pushers operating in two retorts extending the whole length of the firebox at grate level. Coal is pushed forward and over the edges of the troughs onto the grate. A scraper conveyor in the tender delivers the coal in front of the first pusher in each retort.

EXHAUST STEAM INJECTOR (Clarence Roberts, *Ry. Mech. Engr.*, May, 1921). —The exhaust steam injector is used for boiler feeding on about 4000 locomotives in England and the British Colonies and to some extent in France. It is claimed, in England, that the device effects an average saving in fuel of 10%. It has a steam inlet nozzle of much larger cross-sectional area than that of a live steam injector of similar capacity, to provide for the larger volume of exhaust steam.

In a later form (see *Engg.*, Jan. 20, 1922), with 1 lb. of exhaust steam pressure, feedwater can be delivered against a boiler pressure of 150 lb. For boiler pressures above 150 lb., supplementary live steam must be used. With 5-lb. exhaust pressure it can deliver against a boiler pressure of 180 lb. The supplementary live steam amounts to less than 2 1/2% of the evaporation of the boiler. The feedwater is delivered at from 200° to 220° F.

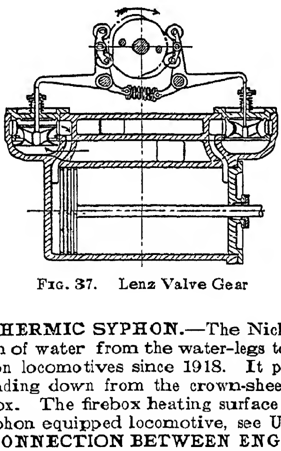


Fig. 37. Lenz Valve Gear

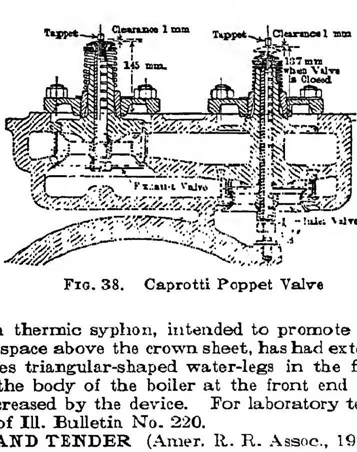


Fig. 38. Caprotti Poppet Valve

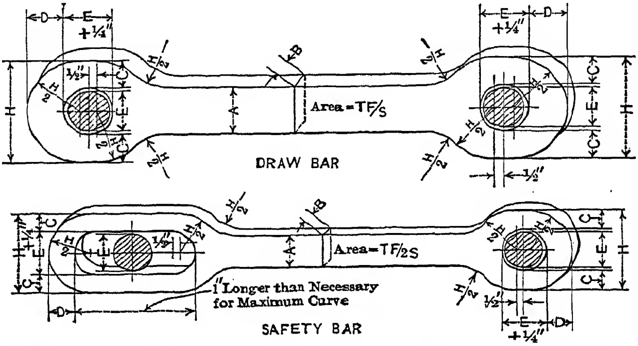
THERMIC SYPHON.—The Nicholson thermic syphon, intended to promote circulation of water from the water-legs to the space above the crown sheet, has had extensive use on locomotives since 1918. It provides triangular-shaped water-legs in the firebox extending down from the crown-sheet to the body of the boiler at the front end of the firebox. The firebox heating surface is increased by the device. For laboratory tests of a syphon equipped locomotive, see Univ. of Ill. Bulletin No. 220.

CONNECTION BETWEEN ENGINE AND TENDER (Amer. R. R. Assoc., 1920).—Drawbars between engine and tender should be straight throughout their full length. Working stress, based on the normal tractive effort of the locomotive, should not exceed 4500 lb. per sq. in.; factor of safety should be not less than 10. If a central safety bar is used, the same limit of stress and safety factor as in the drawbar should be required. A simple formula for designing drawbars and pins is $A = TF/S$, where A = area through body of bar, sq. in.; T = tractive force, lb.; F = factor of safety $\geqq 10$; $S = 45{,}000$ = tensile strength of wrought iron, lb. per sq. in.

Area of each drawbar pin or safety-bar pin = 75% of area through body of bar. The area of the pin is based on the cross-section of a straight bar without bends or offsets and with a depth of pocket not more than 1 in. greater than the thickness of the bar. An offset or bent bar must be of greater cross-section than a straight bar, but it is not necessary to increase the size of pin accordingly. Fig. 39 is a diagram of a satisfactory general form of drawbar. Table 28 gives dimensions of drawbars for various tractive forces. Material in drawbars, safety bars, safety chains and their pins should be refined

wrought iron of ultimate tensile strength not less than 45,000 lb. per sq. in. The use of steel in any of these parts is not recommended.

Safety bars are considered more satisfactory than safety chains for locomotives of 45,000 lb. tractive force, or over. A central safety bar, located immediately beneath the drawbar, is recommended for new locomotives. When a safety bar (at side of drawbar) must be used, Fig. 39 and Table 29 may be used in determining the principal dimensions.



A = Width of Body of Bar
B = Depth of Body of Bar
C = 60% of A
D = 80% of A
E = (Cross Section Area of Pin) = 75% Area of Body of Draw Bar
F = Factor of Safety
H = E + 1/2 + 2C
T = Tractive Force
S = Fiber Stress

FIG. 39. Safety and Draw Bars

A drawbar or safety-bar pin should be considered as a beam fixed at both ends, with a load equal to the rated tractive force of the locomotive. For the condition of bending, $M = WL/8$, and $Z = 0.98\,d^3$. $S = M/Z = (WL/8)/0.98\,d^3 = WL/0.82\,d^3$ or $d = \sqrt[3]{WL/0.82\,S}$, where S = tensile stress, lb. per sq. in.; M = bending moment; W = trac-

Table 28.—Dimensions of Drawbars for Different Tractive Forces at Factors of Safety of 10, 15 and 20

(See Fig. 39. All dimensions in inches)

Tractive Force, 1000 lb.		Fiber Stress 4500 lb. per sq. in. Factor of Safety 10	Fiber Stress 3000 lb. per sq. in. Factor of Safety 15	Fiber Stress 2250 lb. per sq. in. Factor of Safety 20
Over	To	B	B C	
10	20	1 1/2 2 2 1/2 2 1/2	3 1/2 2 2 1/8 2 7/8 2 3/4	3 1/2 2 1/2 2 1/4 3 3
20	30	2 2	2 1/2 2 1/2 3 1/4 3 1/4	4 1/2 3 2 3/4 3 3/4 4
30	45	2 1 2 1/2 3 1/4 3 1/2	5 3 3 4 4	5 3 4 1/2
45	65	3 3	5 1/2 4 3 3/8 4 1/2 4 3/4	6 1/2 4 1/2 4 5 1/4 5 1/4
65	85	1/2 3 1/2 3 1/2 4 1/2 4 1/2	6 1/2 4 1/2 4 5 1/4 1/4	7 1/2 5 4 1/2 6 6
85	110	6 1/2 4 4 5	7 1/2 5 4 1/2 6 6	9 5 1/2 5 1/2 7 1/4 7
110	135	4 1/2 4 1/4 5 3/4 5 1/2	8 1/2 5 1/2 5 1/8 6 3/4 6 3/4	9 6 1/2 5 1/2 7 1/4 7 1/2

Table 29.—Dimensions of Safety Bars (at Side of Drawbar) for Different Tractive Efforts, at Factors of Safety of 10, 15 and 20

(See Fig. 39. All dimensions in inches)

Tractive Force, 1000 lb.		Fiber Stress 4500 lb. per sq. in. Factor of Safety 10	Fiber Stress 3000 lb. per sq. in. Factor of Safety 15	Fiber Stress 2250 lb. per sq. in. Factor of Safety 20
Over	To	D	A E	D
10	20	/8 1 1/4 1 5/8 1 1/2	2 1/2 1 1/2 2	3 1 1/2 7/8 2 1/2
20	30	2 1/2 1 3/8 1 1/2 2 2	3 1 7/8 2 1/2 2 1/4	3 1/2 2 2 1/8 2 7/8 2 3/4
30	45	3/4 1 7/8 2 1/2 2 1/4	3 1/2 2 1/4 2 1/8 2 7/8 2 3/4	4 2 1/2 2 1/4 3 1/4
45	65	3 1/2 2 2 1/8 2 7/8 2 3/4	4 2 1/2 3 1/4 3 1/4	4 1/2 3 1/4 2 3/4 3 5/8 3 3/4
65	85	4 2 1/2 3 1/. 3 1/4	4 1/2 3 1/4 2 3/4 3 3/4 3 3/4	5 3 3/4 3 4 1/4
85	110	4	5 1/2 3 1/2 3 1/4 4 1/2 4 1/4	6 3 3/4 4 7/8 4 7/8
110	135	5 3 3/4	6 3 5/8 4	7 1/2 4 5 3/8

tive force, lb.; L = depth of pocket, in.; Z = moment of resistance; d = diameter of pin, in. Assuming a working stress of 4500 lb. per sq. in., and allowing $^1/_8$ in. on the diameter of the pin for wear, $d = \sqrt[3]{WL/3600} + {}^1/_8$ in. (formula for bending). For the condition of shear, $F = W \times (L - H)/L$, and $S_1 = 0.7854\, d^2$, where S_1 = shearing stress, lb. per sq. in.; F = maximum shearing load, lb. per sq. in.; H = $^1/_2$ depth of bar eye, in. A shearing stress usually is taken as (0.8 × tensile stress) when considering wrought-iron or steel. Then

$$\sqrt{\frac{W \times (L - H)/L}{0.8 \times 0.8 \times 4500}} + {}^1/_8 \text{ in.} \quad \text{(Formula for shear)}$$

Calculations should be made both for shear and bending and the maximum figure taken as the diameter of the pin.

ROLLER BEARINGS FOR LOCOMOTIVES.—The first application of roller bearings on the driving axles of a large locomotive, and their performance in service are described by T. V. Buckwalter. See *Trans.* A.S.M.E., RR-56-1, 1934. The 4–8–4 locomotive, Timken No. 1111, was built in 1930. Roller bearings were used on all axles and also on the crank-shaft and idler gear of the booster. It was built as large and powerful as the clearance limits of the principal U S. railroads would allow. Rated tractive force at starting is 63,700 lb.; boiler pressure, 250 lb.; cylinder diam. and stroke, 27 × 30 in.; driving wheel diam., 73 in.; weight on each driving axle, 66,000 lb. The roller bearings on the driving axles have no adjustable mechanism. Forces due to piston thrust are transmitted and absorbed in a complete train of moving parts of hardened steel, comprising pedestal liner, trunnion guide, hardened wear plates on bearing housing, inner and outer races of the bearing together with the rolls. Completely housing the driving axles and the use of bearings to restrain the axle on a complete circle, eliminates pounding while under steam and while coasting. In 1932, after two years in freight and passenger service on a number of railroads throughout the U. S., 120,000 miles were completed without developing roller bearing troubles of any kind. A study of the bearings then indicated that wheel bearings, excepting driver cones, should have a life expectation of 1,000,000 miles and the driver cones 500,000 miles. Driving and truck axles are lubricated by immersing the bearings in an oil bath, whose level is approximately $^1/_2$ in. below the enclosure level. The lubricant was changed but three times in the two years of operation and little lubrication was added between changes. Thrust reactions due to curvature of track or flange thrusts are taken on the roller bearing surfaces of the tapered bearings. No provision, such as hub liners, for side thrust need be made.

The normal temperature rise of the bearings varied from 15° F. on the engine truck to 40° to 50° on the tender. On the driver bearings it is 15 to 20° above atmosphere. In zero weather service, frost normally adhered to bearing housings and ends of axles. The approximate cost of roller bearing equipment, including bearings and housings and application parts for the 4–8–4 locomotive, not including tender, was $8,000.

LOCOMOTIVE TENDERS.—(Paul T. Warner, *Baldwin Locomotives*, Oct., 1932). Up to 1924, tender frame and tank were made separate, the frame acting as a carriage. With high-capacity tenders, the frame is a substantial steel casting united to the tank to prevent weaving and buckling. Tender frames now are constructed of the water bottom type in a single steel casting to which the tank is riveted or welded. Some tenders are welded throughout. The water capacity of tenders ranges from 18,000 to 25,000 gal., with corresponding fuel capacities ranging from 25 to 30 tons of coal or up to 6,000 gal. of oil. The high-capacity tender usually is carried on two 6-wheeled equalized pedestal trucks. The truck frame is a one-piece steel casting carrying a bolster suspended on swing links. Clasp brakes with shoes on front and back of each wheel are used. Tenders with water capacities from 15,000 to 25,000 gal., when loaded, have weights as follows, in percentage of total locomotive weight: Fast passenger locomotive (4–6–4), 80%; fast freight and passenger (4–8–2, 2–8–4, and 4–8–4), 78%; freight (2–10–2 and 2–10–4), 76%; freight, articulated types, 55%.

COUNTERBALANCING OF LOCOMOTIVES (Recommended Practice Am. Ry. Assoc., 1931).—The following diagrams and formulas illustrate the method recommended for calculating the weight and position of counterbalance to fully balance the out-of-plane action of the revolving weights. Cross balancing of coupled drivers is not considered.

Considering the main driver only, two counterweights are required to completely balance forces set up by revolving overhanging weights, one in the same wheel as the crank-pin, and one in the opposite wheel. Consider the moments produced by the various weights acting at their respective distances from a reference plane through the counterbalance center of the opposite wheel, as shown in Fig. 40. The counterweight required

in the adjacent wheel directly opposite the crank-pin hub and acting at crank-pin radius is

$$W_c = \{(W_1 A + W_2 B + W_3 C + W_4 D)/2B\} + W_t/2$$

where A, B, C, D, E, all in inches, are as in Fig. 40. W_1 = weight of crank-pin hub; W_2 = weight of part of side rod; W_3 = weight of back end of main rod; W_4 = weight of eccentric crank; W_t = total revolving weights = $W_1 + W_2 + W_3 + W_4$. All weights are in pounds and include the included part of the crank-pin.

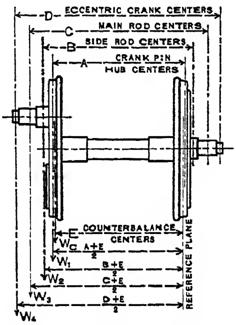


Fig. 40. Counterbalancing

The resulting cross-force F, Fig. 41a, acting in the plane of the counterbalance of the opposite wheel, is balanced in the opposite wheel by a counterweight $(W_c - W_t)$ at crank-pin radius. As the cranks in the wheels are set 90° apart, to combine the counterweight necessary to balance this cross-force with the counterweight opposite the crank-pin, it is necessary to set the final counterweight at an angle θ to the line through the crank and axle centers.

$\theta = \tan^{-1}(W_c - W_t)/W_c$, and the amount of counterweight at angle θ is $\sqrt{W_c^2 + (W_c - W_t)^2}$. If the right crank leads the left, the counterbalance in the right wheel must be advanced through angle θ and set back the same amount in the left wheel. If the left crank leads the right, the reverse is true.

Amount of final counterbalance $= r\sqrt{W_c^2 + (W_c - W_t)^2}/R$, where R = distance from center of gravity of final counterbalance to axle center, in.; r = crank radius, in.

The balancing of 50% of the reciprocating weights is recommended. If W_x = total reciprocating weight, the percentage of reciprocating weight actually balanced is $100\{(W_x/2) - (W_c - W_t)\}/W_x$. The value $(W_c - W_t)$ is the deficiency in revolving balance which must be deducted from reciprocating balance, as the revolving weights first must be satisfied.

Example.—Two-cylinder locomotive, 2–8–2 type; cylinder, 28 in. diam., 30 in. stroke; driver diam., 69 in.; weight on drivers, 257,000 lb.; weight of locomotive, less tender, 376,000 lb. Weights of parts, lb., including included part of crank-pins and bushings: W_3, back end of main rod, 964; W_4, eccentric crank, 153; W_2, side rod at main wheel, 718; W_1, crank-pin hub, 395; side rod, crank-pin and hub on intermediate driver, 730; side-rod, crank-pin and hub, 1st and 4th drivers, 400; W_x, reciprocating parts, including piston, cross-head, wrist-pin, union link and front end of main rod, 1888; W_t, total revolving weight, 2230. Dimensions, in.; C, cylinder centers, 92; B, side-rod centers, 76 3/4; D, eccentric crank centers, 104 3/4; A, crank-pin hub centers, 63 1/2; B, counterbalance centers, 63; R, radius of center of gravity of counterbalance, 15 1/2; r, radius of crank-pin, 15. Right crank leading, 90°.

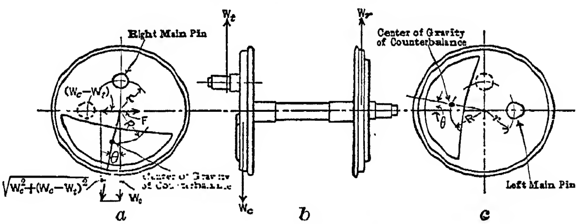


Fig. 41. Counterbalancing

$$W_c = \{(W_1A + W_2B + W_3C + W_4D)/2E\} + W_t/2$$
$$= [\{(395 \times 63\ 1/2) + (718 \times 76\ 3/4) + (964 \times 92) + (153 \times 104\ 3/4)\}/(2 \times 63)] + (2230/2)$$
$$= 2582 \text{ lb.}$$

$$\theta = \tan^{-1}(W_c - W_t)/W_c = (2582 - 2230)/2582 = 0.13632 = 7°46'.$$

Counterweight crank-pin radius $= \sqrt{W_c^2 + (W_c - W_t)^2} = \sqrt{2582^2 + 352^2} = 2606$ lb. Final counterbalance required to balance revolving parts

$$= r\sqrt{W_c^2 - (W_c - W_t)^2}/R = (15 \times 2606)/15\ 1/2 = 2522 \text{ lb.}$$

Percentage of reciprocating balance

$$= 100\{(W_x/2) - (W_c - W_t)\}/W_x = 100\{(1888/2) - 352\}/1888 = 31.4\%,$$

which should be equally distributed among the drivers.

Rules for counterbalancing, adopted by different locomotive builders, are quoted in a paper by Prof. Lanza (*Trans.*, A.S.M.E., x, 302). See also *Trans.*, A.S.M.E., vol. xvi, 305; and *Trans.* Am. Ry. Master Mechanics Assoc., 1897. W. E. Dalby's book Balancing of Engines (Longmans, Green & Co., 1902) contains a very full discussion of this subject. See also Henderson's Locomotive

Operation (*The Ry. Age*, 1904), and Balancing of Locomotives, S. H. Jenkinson, *New Zealand Jour. of Science and Technology*, Jan., 1919. The Balancing and Dynamic Rail Pressure of Locomotives, R. Eksergian, *Trans.* A.S.M.E., RR-51–5, May-Aug., 1929. Lawford H. Fry (Locomotive Counterbalancing, *Trans.* A.S.M.E., RR-56-2, June, 1934) describes a detailed method of cross-balancing, modifying and extending the Am. Ry. Assoc. method given above. See also The Balancing of Three-cylinder Locomotives, F. L. Baxter, *The Engr.*, July 26, 1935.

LOCOMOTIVE WHEEL BALANCING MACHINE.—The Great Western Ry. (England) driving wheel balancing machine comprises a pair of bearings, each supported on four springs, so arranged that the bearing floats with the axle if the wheels are out of balance when spun. A 35-Hp. motor drives the wheels through an extensible shaft with a Hooke joint at each end. A tachometer indicates the speed in m.p.h. For each type of wheel, the calculated revolving and reciprocating masses to be balanced are represented by weights attached to the crank-pins. For inside-cylinder engines, the weights are split and strapped to the crank-pin. To obtain correct results, the weights representing that portion of the reciprocating masses to be balanced should revolve in the plane in which these masses reciprocate. For coupled wheels, the weights representing reciprocating masses are brought into the plane of the connecting-rod, piston, crosshead, etc., by temporary extensions on the crank-pins.

7. POWER BRAKES

The following notes are abstracted from a paper by L. K. Sillcox (*Trans.* A.S.M.E., 1928).

THE AIR BRAKE consists of: Air pump or compressor and main reservoir on the locomotive; engineman's brake valve for controlling brake operations; brake pipe or continuous pipe running throughout the train, connected between cars by flexible hose, for air supply and brake control; triple valve located on each car to charge, apply and release the brakes automatically in response to changes in brake-pipe pressure; auxiliary reservoir on each car; brake cylinder on each car, whose piston is connected, through suitable levers, to the brake shoes.

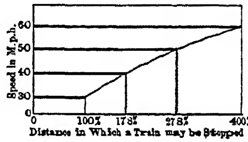


FIG. 42. Comparative Stopping Distances of Train from Various Speeds

Passenger-car brake refinements are: A supplementary reservoir under each car to give high brake-cylinder pressure in emergency application; graduated release, allowing brake-cylinder pressure to be exhausted in steps; quick recharge of auxiliary reservoirs; modifications in valve structure to increase certainty of quick emergency action when desired without possibility of undesired quick action; quick action emergency if brake-pipe pressure falls below a predetermined point; increased certainty and flexibility of brake application and release in service operation.

Freight-car brake refinements for long trains are: Quick-service feature, to make brakes in ordinary service braking much more prompt and positive in application; uniform or restricted recharge feature, to enable auxiliary reservoirs on head and rear ends of long trains to recharge in nearly the same time; uniform or retarded release feature to more uniformly exhaust brake-cylinder pressure on head and rear ends of long trains.

LAWS GOVERNING RETARDATION.—Let A, a, = acceleration, mi. per hr., and ft. per sec., respectively; e = overall efficiency of foundation brake rigging; F, F_1 = retarding or accelerating force, lb. per ton; f = average coefficient of brake shoe friction; g = acceleration due to gravity = 32.2; m = mass = w/g; P = braking effort, lb.; R_c = curve resistance, lb.; R_g = grade resistance, lb.; R_t = train resistance on level tangent, lb.; S = total distance covered, ft.; S_1 = distance, ft., to reduce speed from V_2 to V_1; S_2 = distance traversed, ft., before braking force is effective; t = time, sec.; w = weight, lb.

The fundamental formula is force $= ma = aw/g = 2000\,a/g$ per ton weight;

$a = A \times (5280/3600) = 1.467\,A = (V_2 - V_1)/t.$

$S = \{(V_1 + V_2)/2\}1.467\,t,$ and $t = S/\{(V_1 + V_2)/2\}1.467.$

$A = \{(V_2 - V_1)(V_1 + V_2)/2\,S\} \times 1.467^2 = 1.467^2\,(V_2^2 - V_1^2)/2\,S.$

$F = (2000/32.2)\{(V_2^2 - V_1^2)/2\,S\} \times 1.467 = 67\,(V_2^2 - V_1^2)/S = Pef + R_t + R_c + R_g.$

In train operation F may be increased 5% to provide force for rotational acceleration, whence $F_1 = 70\,(V_2^2 - V_1^2)/S.$

$$S_2 = 1.467\,V_2 t, \text{ and } S_1 + S_2 = 70\{(V_2^2 - V_1^2)/F\} + 1.467\,V_2 t.$$

In stopping a train, V_2 = speed at which brake is applied, and $V_1 = 0$. Then $S = S_1 + S_2 = (70\,V_2^2/F) + 1.467\,V_2\,t.$

Assuming that braking effort is instantaneously effective, $t = 0$ in the above equation,

and $S = 70\,V_2^2/F$. The distance in which a train may be stopped with a constant retarding force F, thus varies with the square of the initial speed. See Fig. 42.

The distance S may be decreased by increasing the retarding force F, or decreasing the time t in which braking effort becomes effective. The actual braking force F obtainable at the wheel rim is reduced by low efficiency of foundation rigging, e, and the uncertain and varying coefficient of friction, f, between shoe and wheel. See Table 30 for values of factor ef. The maximum value obtainable of $F = Pef$ is 180% × 0.141; the minimum value is 125% × 0.70 = 0.0875. To prevent wheel sliding, maximum braking coefficient must not exceed adhesion between wheel and rail. Fig. 43 shows stopping distances with 100% retardation. Distances in practice are about 4 times those shown in Fig. 43, allowing an adhesion factor of 25%.

FIG. 43. Stopping Distances with 100% Retardation

TYPES OF FOUNDATION BRAKE RIGGING are shown in Fig. 44. Let F = force of brake cylinder-pressure × piston area; B = brake shoe pressure per pair of shoes; N = number of pairs of brake shoes per car, whence NB = total brake shoe pressure; R = total lever ratio = NB/F; W = weight of car empty; P = braking ratio = NB/W; a, b, c, etc. = length of lever arms as shown in Fig. 44. Then for the various arrangements of rigging, the following relations hold:

Arrangement A:

$B = F\{a(c + d)/(bd)\}$

$N = 4$

$R = 4\{a(c + d)/bd\}$

$P = (4F/W)\{a(c + b)/bd\}$

Arrangement B:

$B = Fa/b$

$N = 6$

$R = 6\,a/b$

$P = (6\,F/$

Arrangement C:

$\{$

$N = 6$

$R = 6\,ac(e + f)/bf(c + d)$

$P = (6\,F/W)\{ac(e + f)/bf(c + d)\}$

Arrangement D:

$\{$

$= 4$, or 6

$+ f)/bdf\}$.

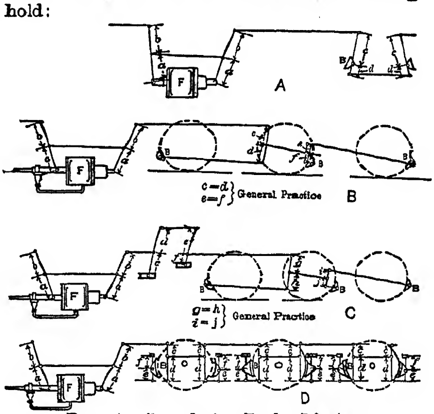


FIG. 44. Foundation Brake Riggings

BRAKING RATIOS, the total brake shoe pressure against the wheels of a locomotive or car divided by the weight, are given by S. G. Downs (*Ry. Mech. Engr.*, May, 1930). Steam railway passenger-train cars for service operation are braked at a ratio of 90% of the light weight; for emergency operation, 150%. The standard freight car for service operation is braked at 60% of its light weight; for emergency operation, 65%. Electrically-propelled cars in rapid transit service normally are braked at 100% of the light weight; for emergency, 140%. Single-unit street cars are braked at ratios of 100 to 120%.

Table 30.—Efficiency Factors (Values of *ef*)

Speed, m.p.h.	Braking force, percent	Clasp brake		Single shoe	
		Plain	Flanged	Plain	Flanged
30	125	0.141	0.169	0.108	0.112
	150	.129	.154	.099	.103
	180	.118	.141	.090	.094
60	125	.103	.122	.074	.090
	150	.094	.112	.068	.082
	180	.086	.102	.062	.075
80	125	.092	.109	.070	.074
	150	.084	.100	.064	.068
	180	.077	.092	.059	.062

An extreme variation in gross to tare ratio on freight cars is found on the coal cars of the Virginian R. R., weighing 75,000 lb. with a capacity of 120 tons. They are equipped with a special form of empty-and-load brake which involves the use of three brake cylinders. The braking ratio is 40%, empty and loaded.

The clasp form of brake is most desirable for heavy cars, as it permits the heavy brake work to be distributed over two shoes per wheel, thereby keeping shoe temperature low with resultant higher coefficient of friction. It also eliminates longitudinal thrust on the journal bearings and the tendency to force the journal brass out of its normal position.

LOCOMOTIVE STANDARDS AND RECOMMENDED PRACTICE.—A manual issued by the Am. Ry. Assoc., 1922, gives specifications for materials for locomotives. Standards and recommended practice for parts, as wheels, tires, axles, etc., rules for inspection and testing of locomotive boilers and rules for fuel economy on locomotives are given. See also specifications of Am. Soc. Test. Matl.

8. LOCOMOTIVE PERFORMANCE

LOCOMOTIVE TESTS OF THE PENNA. R. R.—Eight locomotives were tested in the dynamometer testing plant built by the P. R. R. Co. at the St. Louis Exposition in 1903. The principal results obtained are shown in Table 31.

The conclusions derived are: The coal used in these tests was a semi-bituminous, containing 16.25% volatile combustible, 7.00% ash and 0.90% moisture. The maximum boiler capacity ranged from 8 1/2 to more than 16 lb. of water evaporated per hour per sq. ft. of heating surface. Little or no advantage was found in the use of Serve or ribbed tubes. The boiler efficiency decreases as the rate of power developed increases. Furnace losses due to excess air are no greater with large grates properly fired than with smaller ones. The boilers with small grates were inferior in capacity to those with large grates. No special advantage is derived from large firebox heating-surface. The tube heating-surface is effective in absorbing heat not taken up by the firebox. The steam consumption of simple locomotives operating at speeds and cut-offs commonly used ranges from 23.4 and 28.3 lb. per I.Hp.-hour; of compound locomotives, between 18.6 and 27 lb.; with superheating the minimum steam consumption was reduced to 16.6 lb. of superheated steam. The simple engine used 40% more steam than the compound at 40 r.p.m., 27% more at 80 r.p.m. and only 7% more at 160 r.p.m.

The lowest figures for dry coal consumed per dynamometer Hp. were approximately:

		40	80	160	240
Revs. per min		40	80	160	240
Compound freight engine	lb. (		2.25	3.25	....
	dyn		800	800	....
Compound passenger engine	lb. coal		2.8	2.3	3.0
	dynamometer Hp.		600	900	1000

The frictional resistance of the engines showed an extreme variation ranging from 6 to

Table 31.—Locomotive Tests at St. Louis Exposition

Engine Performance				
Maximum I.Hp., four freight locomotives	1041	1050	1098	1258
Maximum I. Hp., four passenger locomotives	816	945	1622	1641

	Simple Freight	Compound Freight	Compound Pass.
Minimum water per I.Hp.-hr., lb.	23.67	20.26	18.86
Water per I.Hp.-hr., maximum load, lb.	23.83	22.03	21.39
Water per I.Hp.-hr., maximum consumption, lb.	28.95	25.31	24.41

Boiler Performance

Coal per sq. ft. of Grate per hr., lb.	Equiv. Evap. per sq. ft. of Heat. Surface per hr., lb.	Coal per sq. ft. of Heating Surface per hr., lb.	Equiv. Evap. per lb. of Dry Coal, lb.	Equiv. Evap. per sq. ft. of Heat. Surface per hr., lb.	Equiv. Evap. per lb. of Dry Coal, lb.
20	3–5	0.6	10–11.5	4	9.7–12.1
40	5–7.5	0.8	9–10.5	6	8.8–11.3
60	7–10	1.0	8.2–9.7	8	7.8–10.5
80	8.2–12	1.2	7.7–9.1	10	6.8–9.6
100	10.4–14	1.4	7.1–8.5	12	5.8–8.8
120	11.4–15.3	1.6	6.6–8.1	14	5.5–8.
		1.8	6.2–7.7		

38% of the indicated horsepower. Frictional losses increased rapidly at speeds in excess of 160 r.p.m. It appears that machine friction is closely related to lubrication. With oil lubrication a stress at the drawbar of approximately 500 lb. is required to overcome the friction of each coupled axle, while with grease the required force is from 800 to 1100 lb.

The Penna. R. R. later removed the testing plant to Altoona and continued to operate it. Some test results with freight and passenger locomotives are given in Table 32.

ECONOMY OF HIGH PRESSURES (*Bull.* No. 26, Univ. of Ill. Expt. Station).—Tests of a locomotive with cylinders 16 × 24 in. at the Purdue University locomotive testing plant gave results as follows:

Boiler pressure, lb. per sq. in.......	120	140	160	180	200	220	240
Steam per I.Hp.-hour, lb..........	29.1	27.7	26.6	26	25.5	25.1	24.7
Coal per I.Hp.-hour, lb...........	4	3.77	3.59	3.50	3.43	3.37	3.31

In the same tests boiler economy at different rates of driving and different pressures was determined, the results leading to the formula $E = 11.305 - 0.221\ H$; E = lb. of water evaporated from and at 212° F. per lb. of Youghiogheny coal; H = equivalent evaporation per sq. ft. of heating surface per hour, with an average error not exceeding 2.1% for any pressure.

LOCOMOTIVE SUPERHEATER TESTS ON PENNA. R.R. LOCOMOTIVE (C. D. Young, *Proc.* Franklin Inst., 1914).—A Schmidt fire tube superheater, or an altered form of it, was used in all the tests. The normal superheater consisted of tubes arranged in 32 groups in large flues in the boiler. (See Fig. 45.) Each superheater element comprised four seamless steel tubes, outside diam. 1 7/16 in., 0.148 in. thick. To obtain a range of superheat, superheaters of 1/4-, 1/2-, 3/4- and full-length, and short returns at the header were used. The shortened superheaters were not expected to develop an arrangement suitable or desirable for use in regular service; their purpose was to obtain wide variations in superheat.

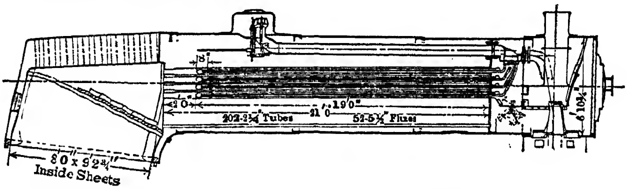


Fig. 45. Locomotive Boiler Arrangement for Superheater Test

Considering only the economy in steam consumption resulting from superheat, Figs. 46 and 47 give authentic data on economy due to superheat, unobscured by other variables.

Fig. 46 indicates that for every increase in superheat at every cut-off there is a saving in steam. At 15% cut-off, at the speed of these tests (240 r.p.m.), for every 20° rise in superheat the water is reduced 1 lb. At 50% cut-off this increases to a requirement of about 40° rise for the same reduction in water rate. With a cut-off of 15% and a superheat of about 70° a water rate of 19 lb. per I.Hp.-hr. can be obtained, while if the cut-off is extended to 50% at the same speed, the superheat must be increased to 300° if the water rate is to remain 19 lb. The importance of the length of cut-off is apparent.

Table 32.—Penna. R. R. Locomotive Tests

Engine Performance

Maximum I.Hp.: 2–10–2 freight locomotive, 3486; 4–6–2– passenger locomotive, 3184

Using Superheated Steam	Freight, Limited Cut-off		Passenger	
	Coal, lb.	Water, lb.	Coal, lb.	Water, lb.
Minimum consumption per I.Hp.-hr.....	2.0	14.9	1.5	15.0
Consumption at maximum load........	3.4	16.6	3.7	20.5
Maximum consumption................	4.3	21.6	3.8	21.2

Boiler Performance, 4–6–2 Passenger Locomotive

Coal per sq. ft. of Grate, per hr., lb.	Equiv. Evap. per lb. of Dry Coal, lb.	Equiv. Evap. per sq. ft. of Heat. Surf. per hr., lb.	Equiv. Evap. per lb. of Dry Coal. lb.	Coal per sq. ft. of Grate, per hr., lb.	Equiv. Evap. per lb. of Dry Coal, lb.	Equiv. Evap. per sq. ft. of Heat. Surf. per hr., lb.	Equiv. Evap. per lb. of Dry Coal, lb.
20	12.6	4	12.9	120	9.1	14	9.2
40	11.9	6	12.1	140	8.4	16	8.4
60	11.2	8	11.4	160	7.6	18	7.7
80	10.5	10	10.6	170	7.3		
100	9.8	12	9.9				

Coal used—Penna. bituminous: Volatile matter, 34.9%; ash, 7.7%; moisture, 1.2%; calorific value, 14,467 B.t.u. per lb. Coal was hand-fired.

EVAPORATION IN LOCOMOTIVE BOILERS.—Table 33 from Merriman's Civil Engineer's Handbook, gives the average evaporation in locomotive boilers per pound of coal from feedwater at 60° F. into steam at 200 lb. per sq. in. In bad-water districts, 10% should be deducted from the figures in the table for each $^1/_{16}$ in. of accumulated scale, and 1% for each grain per gallon of foaming salts. Heating-surface used in computing Table 33 includes only water-evaporating surface, and does not include superheater surface.

DISTRIBUTION OF HEAT OF COMBUSTION IN LOCOMOTIVE BOILERS.—J. T. Anthony (*Jour.* A.S.M.E., Sept., 1917) states that the firebox furnishes only 5 to 10% of the total heating-surface, but is responsible for 25 to 50% of the total evaporation, due to the fact that its heating-surfaces are ideally disposed for the absorption of radiant heat. It is common practice to design locomotives to deliver their rated tractive effort when burning 120 lb. of coal per sq. ft. of grate per hour. Of the heat contained in the coal, 53% is absorbed by the boiler and 47% lost. Of the heat lost, 27% is chargeable to the furnace,   , to the heating-surfaces, and 14% is unavoidably lost in the front end gases.

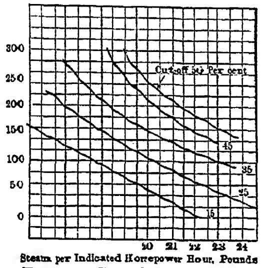


FIG. 46. Superheat and Steam Consumption, Simple Locomotive

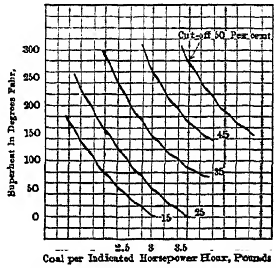


FIG. 47. Superheat and Coal Consumption, Simple Locomotive

BRICK ARCH TESTS.—Tests of a Mikado type locomotive made by the Penna. R. R. locomotive testing plant at Altoona in 1916, shows the value of the brick arch in the firebox. The arch equipment was of the Security sectional type, supported on water tubes, Fig. 48. A high-volatile bituminous coal, sized by being passed over a screen, and hand-fired, was used. The advantages which could be ascribed to the arch were: Maximum evaporation when using high-volatile coal was increased 15.5%; a lower smoke density was observed; increased evaporation per pound of coal, which for ordinary rates of working represents an economy in coal of from 6 to 8%; higher boiler efficiency at all rates of evaporation was obtained.

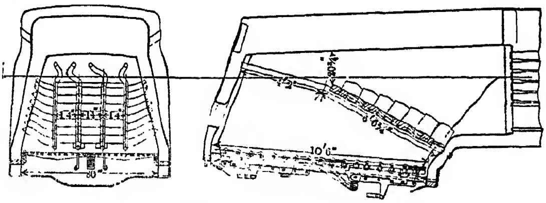


FIG. 48. Firebrick Arch Locomotive Furnace

LOCOMOTIVE BOOSTER TESTS (*Ry. Age*, Sept. 16, 1922).—Tests of a booster on a stationary testing plant, with steam slightly superheated, show that with 200 lb. steam pressure, a maximum drawbar pull of about 11,000 lb. can be obtained with trailing wheels 45 in. diameter. This pull decreases to about 10,500 lb. at 5 m.p.h. and to about 9300 lb. at 10 m.p.h. The mechanical efficiency was between 90 and 95%. The maximum B.Hp. of the booster engine was 316 at 17.8 m.p.h., and the steam consumption ranged between 6000 lb. at 5 m.p.h. and 15,000 lb. at 20 m.p.h., or between 40.8 lb. and 43.4 lb. per B.Hp.-hr. The booster had a 2-cylinder double-acting engine, with cylinders 10 × 12 in. and cranks set at 90°. The gear ratio between engine shaft and trailer axle was 14 to 36.

Table 33.—Average Evaporation in Locomotive Boilers

Thermal Value of Coal, B.t.u.	Pounds of Coal Fired per Hour per sq. ft. of Heating-Surface											
	0.8	1.0	1.2	1.4	1.6	1.8	2.0	2.2	2.4	2.6	2.8	3.0
	Evaporation, lb. of Steam per lb. of Coal											
10,000	5.24	4.87	4.55	4.25	3.98	3.74	3.51	3.31	3.13	2.96	2.80	2.66
12,000	6.29	5.85	5.46	5.10	4.78	4.49	4.22	3.98	3.75	3.55	3.37	3.19
14,000	7.34	6.82	6.37	5.95	5.57	5.24	4.92	4.64	4.38	4.14	3.93	3.73

LOCOMOTIVE FEEDWATER HEATING (Thomas C. McBride, *Trans.* A.S.M.E., xlii, p. 307, 1920).—Waste heat to heat the feedwater may be obtained from the waste gases in the smokebox or stack and the exhaust steam. Waste gas heaters cannot be used with cold water, as accumulations of condensation and soot affects their efficiency. The exhaust-steam heater may be either: 1. Open or injection; the exhaust steam comes in direct contact with the cold feedwater, and water condensed from the exhaust steam is mixed with the feedwater; 2 Closed or surface; heat from the exhaust steam passes to the feedwater through thin sheets of metal, generally thin brass tubes. Tables 34 to 36 give data regarding the performance of locomotive feedwater heaters.

A report of a committee of the Am. R. R. Assoc. (1920) on feedwater heaters for locomotives, states that a test of a Worthington open-type heater on a 2–8–2 locomotive on the Penna. Railroad showed that feedwater entering the heater at temperatures between 39° and 41° F. is heated to between 178° and 211° F.

Table 37 shows the weight and heat of the steam used by the engines and boiler feed pump, and the heat recovered by the heater. The exhaust steam drawn from the exhaust passages and condensed in the heater, weighed from 1735 to 7722 lb. per hr., or from 12 to 15% of the feedwater used by the boiler. The heater locomotive was operated at rates of boiler feeding of from 14,480 to 52,475 lb. per hr., the latter rate being close to the possible maximum with this locomotive. A closed-type heater tested on the same 2–8–2 locomotive gave typical test results as shown in Table 38. The heater contained 192 brass tubes, 5/8 in. outside diam., 46 in. long. Total heating-surface was 121 sq. ft.

In tests with the closed heaters the condensed exhaust steam was not recovered. Later designs return this condensation to the tender tank.

WEIGHT OF EXHAUST STEAM CONDENSED.—With the open-type feedwater heater, the exhaust steam used to heat the water is condensed and becomes a part of the boiler feedwater supply. While it would be possible to weigh this condensation by

Table 34.—Heat Saving Due to Locomotive Feedwater Heater

(McBride)

Assumed steam pressure in branch pipe, lb. per sq. in.	200	200	200	200
Assumed superheat, deg. F.	None	None	150	150
Assumed temperature water in tender, deg. F.	40	70	40	70
Heat content per pound of steam, B.t.u.	1199.1	1199.1	1284.6	1284.6
Heat content per lb. of water, B.t.u.: at 40° F.	8.0		8.0	
at 70° F.		38.0		38.0
Heat to generate one pound of steam, B.t.u.	1191.1	1161.1	1276.6	1246.6
Heat saving, B.t.u., water heated from: 40° to 215° F.	175.0	....	175.0	
70° to 215° F.		145.0		
Heat saving, percent		12.5		11.6
Total heat required with feed pump as compared to injector operation, percent	102	102	101.75	101.75
Heat saving in locomotive with heater as compared to injector operation, percent	15.0	12.75	13.94	11.80
Heat required, percent	87.0	89.25	87.81	89.95
Heat saving as compared to injector operation, percent.	13.0	10.75	12.2	10.05

Table 35.—Exhaust Steam Required for Heating

Assumed tender water temperature, deg. F.	40	70
B.t.u. required to heat one lb. of feedwater to 215° F. from: 40° F.	175	....
70° F.	....	145
Heat of exhaust steam assumed at 7 lb. pressure, B.t.u. per lb.	1157	1157
Heat of water at 215° F., B.t.u. per lb.	183	183
Heat given up by exhaust steam in condensing from 7 lb. pressure to water at 215° F., B.t.u. per lb.	974	974
Exhaust steam required to heat one lb. of cold feedwater, percent	18	15
Exhaust steam to heater from feed pump, percent	2	2
Exhaust steam from exhaust ports required by surface heater, percent	16	13
Exhaust steam from exhaust ports required by open heater, percent	13.8	11.5

Table 36.—Water Saving Due to Feedwater Heater

Steam for locomotive cylinders, percent	100	100
Steam required for feed pump, percent	2	2
Water required by locomotive with surface heater discharging water condensed from exhaust steam to track, as compared to injector operation, percent	102	102
Exhaust steam condensed in heating feedwater, percent: 0.18×102	18.4	.....
0.15×102	....	15.3
Water required by locomotive with open heater or surface heater saving condensation from exhaust steam, as compared to injector operation, percent	83.6	86.7
Water saving, percent	16.4	13.3

drawing off the water from the heater and then returning it, this method would be impracticable, as heat would be lost and the regular action of the heater interfered with. Instead of weighing this condensation, its weight is calculated from the observed temperature of the steam and water by the formula $S = \{W(h_1 - h_0)\} \div (H - h_1)$, where S = exhaust steam condensed and added to feedwater, lb. per hr.; W = weight of feedwater from tank, lb. per hr. (deducting losses); h_0 = heat in feedwater from tank, B.t.u. per lb. (cold water); h_1 = heat in feedwater from heater, B.t.u. per lb. (hot water); H = heat in exhaust steam, B.t.u. per lb. (at heater). Weight of steam condensed as found by this formula, is added to weight of feedwater from tank; total is water fed to boiler.

PISTON VALVE LEAKAGE. (*Engg.* July 4, 1919).—In an investigation by the Penna. R. R. of the leakage of locomotive piston valves, a full size valve was operated under 200 lb. per sq. in. pressure with saturated and superheated steam. Results were as follows: A 12-in. diam. valve with two $1/2 \times 5/8$-in. rings at each end, operated in a plain bushing with saturated steam, leaked from 171 to 183 lb. per hr. at each end. With steam superheated 100° F., the leakage ranged from 194 to 210 lb.; at 200° F. superheat, from 181 to 197 lb.; at 300° F. superheat, from 122 to 132 lb. Length of valve travel (2 to 6 in.) and speed (strokes per min.) had little effect upon leakage. No difference in leakage was found between the step and diagonal form of ring joint. Rings with $1/16$ in. side clearance showed somewhat less leakage than the same rings closely fitted in the grooves. The steam pressure in the ring grooves back of the rings was found to be the same as the steam chest pressure when the valve was at rest. Pressure did not fall under the first or steam side ring, when in motion, but under the second or exhaust side ring, the pressure ranged from 50% to 80% of steam chest pressure.

CONTROL LOCOMOTIVES FOR ROAD TESTS (R. P. Wagner, *Trans.* A.S.M.E. RR–51–4, 1929).—Instead of a train, a specially arranged locomotive under steam is coupled to the rear end of the dynamometer car. The valve gear is placed in full reverse; the exhaust nozzle is shut off to prevent drawing in gases from the front end. A port in the exhaust pipe admits air from outside, which is compressed in the cylinders and blown off at variable pressure through a hand-controlled valve. To protect the cylinders from heating beyond 850°, hot water is admitted into the exhaust (or in this case, inlet) pipe. This water, which is ready for evaporation, rapidly absorbs the heat developed during the compressing process and prevents the temperature rising.

The engineer of the brake locomotive adjusts his engine to maintain the speed under all circumstances. The locomotive under test maintains an even drawbar pull. The weight of the imaginary train thus is adjusted in accordance with the varying grades. If the dynamometer car permits indicating, in addition to reading, of drawbar pull, speed, work performed, all temperatures by automatic resistance thermometers, together with a continuous analysis of combustion gases, this method probably is as efficient as that of a stationary testing plant.

Table 37.—Performance of Open Type Feedwater Heater on Locomotive Test Plant

Test No.	Weight of Steam, Pounds per Minute, Going to				Heat in Steam, B.t.u. per Minute, Going to				Heat Recovered from Exhaust Steam by Heater		Speed, Miles per Hour	Indicated Horsepower		Coal per I.Hp.-hr., lb.		Coal Saving by Feed Heating, Percent
	Engines	Feed Pump	Safety Valve	Feed Heater	Engines	Feed Pump	Safety Valve	Total	B.t.u. per Minute	% of Total Heat in Steam		Without Heater	With Heater	Without Heater	With Heater	
1	314.5	7.4	0.8	42.3	402,749	8,876	960	412,585	50,726	12.3	14.6	990	965	2.2	2.0	9.1
2	492.8	10.5	0.7	71.4	639,063	12.594	840	652,497	85,694	13.1	14.6	1549	1534	2.3	2.0	13.1
3	599.1	12.5	2.1	90.0	780,867	14,994	2519	798,380	108,063	13.5	22.0	2001	1949	2.3	1.9	17.4
4	757.3	14.6	6.5	113.7	996,531	17,393	6597	1,020,521	136,736	13.4	22.0	2388	2373	2.9	2.2	24.7

Table 38.—Performance of Closed Type Feedwater Heater on Locomotive Test Plan

Test No.	Speed, Miles per hr.	Indicated Horsepower		Coal per Indicated Hp.-hr., lb.		Coal Saving by Feed Heating, Percent
		Without Heater	With Heater	Without Heater	With Heater	
1	15	876.3	957.4	2.24	2.19	2.2
2	15	1536.4	1480.1	2.33	2.02	13.3
3	22	1963.2	1916.4	2.54	2.03	20.1
4	22	2382.3	2376.4	2.84	2.16	23.9

Table 39.—Dimensions of Rails Used by Principal Railroads in the United States

Section	Wt.	Dimensions, Inches. (See Fig. 51) A	B	C	D	E	F	G	H	I	J	K	L	N	M	P	Slope of Head	Splice Bar Angles *	Per-cent Head	Per-cent Web	Per-cent Base
R.E.	130	6	6 48/64	2 60/64	42/64	30/64	14	8/16	6/16	12/16	1/16	14	3 3/8	1 14/64	3 44/64	1 54/64	1 to 16	4 to 1	36.4	23.8	39.8
R.E.	120	5 48/64	6 32/64	2 56/64	40/64	30/64	14	6/16	6/16	10/16	1/16	14	3 1/4	1 12/64	3 34/64	1 50/64	1 to 16	4 to 1	37.1	22.7	40.2
R.E.	110	5 32/64	6 16/64	2 50/64	38/64	28/64	14	6/16	6/16	10/16	1/16	14	3 1/8	1 8/64	3 26/64	1 46/64	1 to 16	4 to 1	37.4	23.0	39.6
R.E.	100	5 24/64	6	2 44/64	36/64	25/64	14	6/16	6/16	10/16	1/16	14	2 31/32	1 4/64	3 18/64	1 42/64	1 to 16	4 to 1	38.2	22.6	39.2
R.E.	90	5 8/64	5 40/64	2 36/64	36/64	23/64	14	6/16	6/16	6/16	1/16	14	2 29/32	1	3 10/64	1 30/64	1 to 16	4 to 1	36.2	24.0	39.8
A.R.A.-A.	100	5 32/64	6	2 48/64	36/64	24/64	14	6/16	6/16	6/16	1/16	14	2 15/16	1 4/64	3 24/64	1 36/64	1 to 16	4 to 1	36.9	23.4	39.7
A.R.A.-A.	90	5 8/64	5 40/64	2 36/64	36/64	24/64	14	6/16	6/16	6/16	1/16	14	2 29/32	1	3 10/64	1 30/64	1 to 16	4 to 1	36.2	24.0	39.8
A.R.A.-A.	80	4 40/64	5 8/64	2 32/64	33/64	24/64	14	6/16	6/16	6/16	1/16	14	2 18/32	62/64	2 46/64	1 28/64	1 to 16	4 to 1	38.8	21.0	40.2
A.R.A.-B.	100	5 9/64	5 41/64	2 42/64	36/64	31/64	12	5/16	6/16	5/16	1/16	12	2 65/128	1 5/64	2 55/64	1 45/64	3°	13°	40.2	19.2	40.6
A.R.A.-B.	90	4 49/64	5 17/64	2 36/64	35/64	31/64	12	5/16	6/16	5/16	1/16	12	2 11/32	1 2/64	2 40/64	1 39/64	3°	13°	40.1	19.2	40.7
A.R.A.-B.	80	4 28/64	4 60/64	2 28/64	35/64	31/64	12	5/16	6/16	5/16	1/16	12	2 15/64	1	2 30/64	1 30/64	3°	13°	38.8	19.5	41.7
A.S.C.E.	100	5 48/64	5 48/64	2 48/64	36/64	19/64	12	4/16	5/16	4/16	1/16	12	2 65/128	62/64	3 5/64	1 45/64	None	13°	42.0	21.0	37.0
A.S.C.E.	90	5 24/64	5 24/64	2 40/64	36/64	19/64	12	4/16	5/16	4/16	1/16	12	2 45/128	59/64	2 55/64	1 38/64	None	13°	42.0	21.0	37.0
A.S.C.E.	85	5 12/64	5 12/64	2 36/64	36/64	19/64	12	4/16	5/16	4/16	1/16	12	2 17/64	57/64	2 48/64	1 35/64	None	13°	42.0	21.0	37.0
A.S.C.E.	80	5	5	2 32/64	35/64	19/64	12	4/16	5/16	4/16	1/16	12	2 3/16	56/64	2 40/64	1 32/64	None	13°	42.0	21.0	37.0
A.S.C.E.	75	4 52/64	4 52/64	2 30/64	34/64	18/64	12	4/16	5/16	4/16	1/16	12	2 15/128	54/64	2 35/64	1 27/64	None	13°	42.0	21.0	37.0
Penn. System	130	5 32/64	6 40/64	3	44/64	34/64	12	8/16	7/16	12/16	1/16	16	3 1/4	1 14/64	3 26/64	2	None	18° T; 14° B			
Penn. System	125	5 32/64	6 32/64	3	42/64	34/64	12	8/16	7/16	12/16	1/16	16	3 1/4	1 14/64	3 26/64	1 56/64	None	18° T; 14° B	38.4	22.4	39.2
Penn. System	100	5	5 44/64	2 43/64	36/64	34/64	10	5/16	7/16	5/16	1/16	10	2 31/64	1 6/64	2 50/64	1 52/64	2 1/2 at top	15° T; 13° B	40.4	19.1	40.5
Penn. System	85	4 40/64	5 8/64	2 32/64	34/64	30/64	10	4/16	7/16	4/16	1/16	10	2 15/64	1	2 30/64	1 42/64	None	15° T; 13° B	42.3	18.5	39.2
L.V.	136	6 32/64	7	2 60/64	42/64	28/64	10	8/16	7/16	12/16	1/16	14	....	1 16/64	3 56/64	1 56/64	4°	4 to 1	35.4	23.7	40.9
L.V.	110	5 32/64	6	2 56/64	38/64	24/64	10	6/16	7/16	6/16	1/16	14	2 15/16	1 4/64	3 4/64	1 56/64	4°	4 to 1			
Dudley Ren'f.	105	5 32/64	6	3	40/64	18/64	14	8/16	5/16	1	1/16	14	3 1/8	62/64	3 26/64	1 40/64	1 to 16	4 to 1	40.2	26.2	33.6
Dudley	100	5 32/64	6	3	38/64	18/64	14	8/16	5/16	5/16	1/16	14	3 1/8	62/64	3 26/64	1 40/64	1 to 16	4 to 1	41.2	24.7	34.1
Dudley Ren'f.	90	5	5 32/64	2 42/64	36/64	22/64	14	8/16	5/16	1	1/16	14	3	62/64	3 2/64	1 32/64	1 to 16	4 to 1	37.9	25.7	36.4
Dudley	85	5	5 16/64	2 44/64	34/64	16/64	14	8/16	5/16	5/16	1/16	14	2 5/8	56/64	2 48/64	1 40/64	1 to 16	4 to 1	45.5	20.8	33.7
D.L. & W.	105	5 24/64	6	2 48/64	40/64	26/64	10	4/16	7/16	4/16	1/16	8	2 31/32	1 2/64	3 16/64	1 46/64	4°	13°	39.2	23.2	37.6
D.L. & W.	101	5 24/64	5 28/64	2 48/64	40/64	26/64	10	4/16	7/16	4/16	1/16	8	2 3/8	1 2/64	2 44/64	1 46/64	4°	13°	40.6	20.2	39.2
C. of N.J.	135	6	6 32/64	3 10/64	48/64	....	....	....	....	....	....	....	....	1 14/64	3 18/64	2	4°	14°			
C. & N.W.	100	5 9/64	5 45/64	2 36/64	36/64	35/64	12	5/16	6/16	5/16	1/16	12	2 29/128	1 9/64	2 61/64	1 39/64	2 15/32 at top	13°	36.1	19.7	44.2
C. & N.W.	90	5 6/64	5 34/64	2 32/64	32/64	31/64	12	5/16	6/16	5/16	1/16	12	2 33/64	1 2/64	2 62/64	1 34/64	2 13/32 at top	13°			
G.N.—1918	90	5	5 24/64	2 40/64	38/64	27/64	14	6/16	10/16	10/16	1/16	14	2 21/32	1 2/64	2 56/64	1 30/64	5°	13°	35.7	22.7	41.6
G.N.—1920	90	5	5 24/64	2 40/64	36/64	27/64	12	7/16	7/16	10/16	1/16	14	2 53/64	1 2/64	2 56/64	1 30/64	1 to 16	13°	36.6	21.8	41.6
A.T. & S.F.	90	5 12/64	5 40/64	2 36/64	36/64	22/64	14	6/16	6/16	6/16	7/32	14	2 29/32	1	3 10/64	1 30/64	1 to 16	4 to 1	36.1	24.0	39.9
A.T. & S.F.	85	4 56/64	5 24/64	2 32/64	36/64	28/64	14	6/16	6/16	6/16	1/16	14	2 23/32	1	2 58/64	1 30/64	1 to 16	4 to 1	37.0	22.7	40.2
C.P.R.	85	5	5 8/64	2 32/64	36/64	24/64	8	6/16	4/16	6/16	4/16	8	2 11/32	1	2 44/64	1 28/64	2 7/16 at top	4 to 1	36.4	22.5	41.1
C.N.R.	80	5	5	2 36/64	35/64	21/64	8	6/16	4/16	6/16	4/16	8	2 1/4	58/64	2 44/64	1 26/64	2 1/2 at top	13°	39.2	21.3	39.5
Harriman	75	4 28/64	4 60/64	2 28/64	33/64	24/64	14	6/16	6/16	6/16	1/16	14	2 1/2	60/64	2 40/64	1 24/64	1 to 16	4 to 1	38.5	21.7	39.8

* T = Angle at top; B = Angle at base.

POWDERED FUEL FOR LOCOMOTIVES.—Attempts to fire locomotives with powdered fuel have met with difficulties which have prevented its extensive use, but a successful application in Germany is described by R. Roosen (*Trans.* A.S.M.E. RR–52–8, 1930). The burner consisted of a hollow truncated cone, Fig. 49, whose large end was closed by a spray plate perforated with about 1900 nozzle-shaped holes. At the smaller end is a device to mix the coal dust and air. Two of these burners were placed below the firebox back-head on a 3-cylinder 2–8–2 German State Railways locomotive (Fig. 50) with 2100 sq. ft. heating-surface, and 42 sq. ft. grate area. Each spray plate injects about 3300 lb. of fuel per hour. An intensity of combustion of 316,000 B.t.u. per hour per cu. ft. of furnace volume was attained. The steam generated was 20.5 lb. per hour per sq. ft. of heating surface.

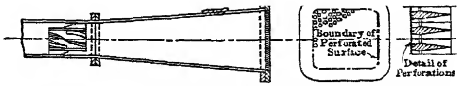


Fig. 49. Arrangement of Burner for Powdered Coal

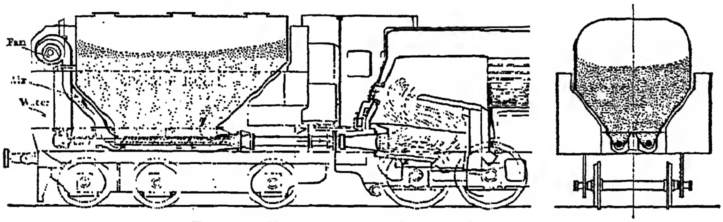


Fig. 50. Layout of Stugg Locomotive

9. TRACKS, GRADES AND CURVES

RAILS USED ON STEAM RAILROADS (C. W. Gennet, Jr., *Ry. Age*, Mar. 15, 1921).—Table 39 and Fig. 51 show the dimensions of the rails used by the principal railroads in the United States in 1921. The abbreviations are: R.E. = Am. Ry. Engg. Assoc.; A.R.A. = Am. Ry. Assoc.; A.S.C.E. = Am. Soc. of Civil Engrs.; P.S. = Pennsylvania System; L.V. = Lehigh Valley; D.L. & W. = Delaware, Lackawanna and Western; C. of N. J. = Central Railroad of New Jersey; C. & N.W. = Chicago & Northwestern; G.N. = Great Northern; A.T. & S.F. = Atchison, Topeka & Santa Fe; C.P.R. = Canadian Pacific Railway; C.N.R. = Canadian National Railway.

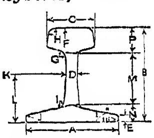


Fig. 51. Rail Sections (See Table 39)

The Penna. R. R. 152-lb. rail described by T. J. Skillman (*Ry. Age*, Aug. 8, 1931) is shown in Fig. 52. It is the heaviest and stiffest rail used on any railway, and was designed to safely sustain an axle load of 100,000 lb. at 100 m.p.h. It is 8 in. high, 6 $^3/_4$ in. base and weighs 152 lb. per yd. The 130-lb. rail, standard since 1916, was designed for axle loads of 80,000 lb. at 80 m.p.h. The 152-lb. rail is 75% stiffer than the 130-lb. rail. Chemical composition is: C, 0.70–0.85; Mn, 0.70–1.00; P, 0.04 max.; Si, 0.15–0.30.

VELOCITY GRADES (Amer. Ry. Master Mech. Assoc. *Proc.*, 1914).—If a ruling grade can be approached at a speed of from 25 to 45 m.p.h., it will be possible to haul heavier trains over the grade than those calculated for a constant speed or dead pull. The formula to be used is $G_m = 3.5 \times (V_1^2 - V_2^2)/L$, where G_m = percent of grade to be deducted from actual grade; V_1 = initial speed, m.p.h.; V_2 = speed at top of grade; L = length of grade, ft. The equivalent grade is found by deducting G_m from the actual grade, and the tonnage rating is calculated for the equivalent grade.

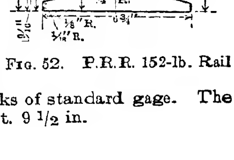


Fig. 52. P.R.R. 152-lb. Rail

GAGE OF TRACK ON CURVES (Am. Ry. Eng. Assoc. Manual, 1915).—Curves of 8° or under should be standard gage. The gage should be widened $^1/_8$ in. for each 2° or fraction thereof over 8°, to a maximum of 4 ft. 9 $^1/_4$ in. for tracks of standard gage. The gage, including widening due to wear, should never exceed 4 ft. 9 $^1/_2$ in.

ELEVATION OF CURVES (Am. Ry. Eng. Assoc. Manual, 1915).—The approximate formula $E = 0.00066\,DS^2$ will give essential theoretically correct elevation for ordinary practice, where E = elevation, in., of the outer rail at gage line; D = degree of curve; S = speed, m.p.h. Ordinarily an elevation of 8 in. should not be exceeded.

CURVES IN RELATION TO WHEEL BASE (*Locomotive Data*, Baldwin Locomotive Works, 1920).—The sharpest curve to which two pairs of flanged wheels will adjust themselves depends upon their distance apart, the diameter of the wheels and the size and shape of the flanges. Assuming the M.C.B. standard for flanges and rails, and that the gage is not widened on the curve, a sufficiently accurate formula for all practical purposes is: $R = (W/2)\sin\alpha$, where R = radius of sharpest curve that can be passed; W = wheel base; α = angle the flanged wheels make with the rails. The value of $\sin\alpha$ for various diameters of wheels is:

Diam. of wheels, in.	20 to 24	25 to 30	31 to 40	41 to 50	51 to 60
$\sin\alpha$	0.117	0.107	0.09	0.08	0.075

When a truck is used, the swing must be sufficient to allow the locomotive to pass the curve. An approximate formula then is $R = WT/2S$, where W = distance from center pin of truck to rear of rigid wheel base; T = distance from center pin of truck to front of rigid wheel base; S = one-half of the total swing of the truck; R = radius of the sharpest curve which can be passed. All dimensions must be in the same unit. Figs. 53 and 54

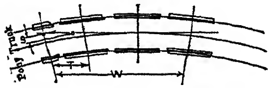


FIG. 53. Two-wheel Truck

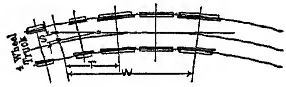


FIG. 54. Four-wheel Truck

show how these dimensions are taken for 2-wheel and 4-wheel trucks. For an extended discussion of this subject, see *Trans.* A.S.M.E., xlii, 1920. Also *Ry. Mech. Engr.*, April and May, 1931.

10. MISCELLANEOUS DATA

According to a publication of the Committee on Public Relations of the Eastern Railroads there were in 1931 on the Class I railroads of the U. S., 55,400 locomotives, 2,200,000 freight cars and 51,750 passenger-cars, operating over 430,000 miles of tracks. Since 1911 the average tractive force of steam locomotives has been increased 62%, to 45,800 lb. The capacity of freight cars has been increased 27%, to 47 tons, in the same period. The average 1931 freight train carried 733 tons of freight in a train of 48 cars, at an average speed of 14.8 m.p.h. In 1920 the fuel required in freight service per 1000 gross ton-miles was 197 lb.; in 1931 it was 137 lb., a decrease of 30%. In the same period the fuel per passenger train-mile has been reduced from 18.8 lb. to 14.5 lb., or 23%.

AIR CONDITIONING.—Air conditioning applied to railway passenger cars means so controlling temperature and moisture of the air that atmospheric conditions in the car are those most suitable for the comfort of passengers. In general practice, temperature only is directly controlled, but there is an incidental precipitation of moisture on the cooling coils of the apparatus, which carries away odors and dust from the air. Provision sometimes is made for adding moisture to the air as needed. Air conditioning requires that the car windows be kept closed. As outside air that is introduced passes through filters, the air in the car is cleaner than when conditioning is not used.

Three general methods of obtaining the necessary refrigeration for air cooling are used:

1. Ice Water Circulating System.—Water is sprayed over ice in a bunker beneath the car floor, or sometimes in the vestibule. The cooled water then is circulated through pipe coils in the upper deck of the car. Air from the car, together with some outside air, is driven by a fan through the cooling coils and discharged into the body of the car.

2. The Mechanical Compressor System uses a closed refrigeration cycle for cooling, as in the ice system. The handling of the air and the cooling coils are the same.

3. The Steam-jet or Water-vapor System.—Refrigeration is produced by spraying water into a partially evacuated, heavily insulated chamber. The sprayed water evaporates and lowers the temperature of the remaining water, which then is circulated in the air-cooling units. Cooling coils and air circulation are the same as in the other systems.

Refrigerants.—The refrigerants used are water vapor in the steam-jet system, and Freon in the compressor systems. A temperature of 50° F. is obtained with water vapor

at a vacuum of 29.637 in. of mercury. Freon (F 12) is an inert gas with a boiling point of −22° F. at atmospheric pressure.

The Main Units of the air conditioning apparatus usually are located beneath the car floor; the air-cooling coils are placed at the top of the car, either at the ends or at the center.

Power for compressor, pumps, air-circulating fans and control, is obtained either directly from the car axle or from an axle-mounted generator and storage battery unit.

The air conditioning apparatus has the following approximate ratings. Water pump motors, 1/4 to 3/4 Hp.; fan motors for air circulation, 1/2 to 1 Hp., depending on whether bulkhead or duct air distribution is used; compressor motors, 5 to 11 1/2 Hp.; axle generators, 7 1/2 to 15 kw.; Storage battery, 400 to 1000 amp.-hr.

Distribution of Air.—Two general systems for distributing the cooled air are in use: 1. The bulkhead system; air is discharged into the car through grilles located near the ceiling, either at one or both ends of the car, or in some installations, at the center. 2. The duct system; air is carried in ducts which have openings designed to distribute the air uniformly throughout the interior of the car.

It is usual to install a steam radiator coil in front of the cooling coil, so that the system can be used as an aid to heating in cold weather and the advantages of filtered, circulated air obtained at all times.

The air handled per car ranges from 1200 to 2400 cu. ft. per min., and the fresh air continuously added from outside may be up to 50% of the air circulated through the car.

WEIGHING PLANTS FOR LOCOMOTIVES (C. C. Bailey, *Ry. Age*, Sept. 3, 1921). —A large scale for weighing locomotives at the Eddystone plant of the Baldwin Locomotive Works has a capacity of 450 tons. The main scale platform is in six sections, each designed to carry a concentrated load of 150,000 lb.; 24 individual wheel scales are mounted on this platform scale, each of 50,000 lb. capacity. The weigh beam of the platform scale is graduated to 895,000 lb. by increments of 5000 lb., with an auxiliary beam of 5000 lb. capacity by increments of 50 lb., giving a total of 900,000 lb. The scale is mounted on a concrete foundation 110 ft. long, 11 ft. wide and 3 ft. deep. Tests of the scale show that the sum of the wheel loads obtained by means of the individual scales varies 0.5% from the total weight recorded by the platform scales.

MILL-TYPE CARS.—A special gondola car is used by several railroads to transport long structural steel shapes. One of these on the Penna. R. R. is 70 ft. 3 in. long over couplings, 65 ft. 6 in. long inside and 7 ft. 7 1/2 in. wide. The sides are 3 ft. 6 in. high. The car weighs 62,000 lb.; normal capacity is 140,000 lb., and load limit 148,000 lb. The cubic capacity is 1738 cu. ft. It is carried on two 4-wheel trucks, each with a wheel base of 5 ft. 8 in.

Table 40.—Gondola Type Cars of Large Capacity Used in Coal Traffic

Railroad	Penna.	C. & O.	N. & W.	Penna.	Virginian
Date of construction	1919	1921	1921	1919	1920
Capacity of car, lb	140,000	200,000	200,000	220,000	240,000
Capacity of car, cu. ft	2,545	3,703	3,636	4,000	4,450
Weight of car, empty, lb	53,400	68,300	53,500	74,600	78,800
Weight of car, loaded, lb	193,400	268,000	253,500	294,600	318,800
Length coupled, ft. and in	44–5 1/2	47–1	46–2	52–5 1/2	53–3 1/2
Width outside, ft. and in	10–2	10–3 5/8	10–1 1/4	10–2	10–3 1/4
Height, rail to top of side, ft. and in.	10–6	11–0	11–0	11–6	11–0
Total wheel-base, ft. and in	37–10	39–7 1/2	40–2	45–6	45–6 1/2
Trucks, number	2	2	2	2	2
Axles, number	4	6	6	6	6
Wheels, diameter, in	33	33	33	33	33
Journals, size, in	6×11	5 1/2×10	5 1/2×10	6×11	6×11
Weight on rail per axle, lb	48,350	44,717	42,250	49,100	53,100
Weight per ft., coupled length, lb	4,350	5,695	5,490	5,616	5,985
Revenue load, % of gross weight	72.4	74.6	78.9	76.4	75.3

AUTOMOTIVE VEHICLES

By Ralph A. Richardson

1. GENERAL INFORMATION

Most passenger automobiles, trucks and buses have 4-cycle engines. Some trucks and buses use high-speed oil or Diesel engines. Cylinders vary from 4 to 16 per engine; some buses use two 6- or 8-cylinder engines. Passenger cars in the low price field have 6 or 8 cylinders; the upper price classes have from 8 to 16. A majority of trucks and buses use 6 cylinders. The brake-horsepower (B.Hp.) output ranges from 40 to 200, the engines peaking at 3000 to 4000 r.p.m. Over a period of years the tendency has been to increase power output and engine speed.

PERFORMANCE FACTORS.—Several useful performance factors may be derived if an automobile engine is assumed to be a gas pump, that at a given time has approximately the same output per cubic inch of displacement and the same specific fuel consumption as all other engines. Then from known specifications, factors representing car performance, economy and wear may be determined, by which to compare one car with another.

Engine r.p.m. per Car-mile per Hour (m.p.h.) is useful for comparing the ratio between engine speed and vehicle speed. It is also useful in deriving the other performance factors. The equation is: Engine r.p.m. per m.p.h. $= E_r = (5280 \times R)/(C \times 60)$, where $R =$ rear axle gear ratio; $C =$ rear wheel rolling circumference, ft., $= \pi \times$ tire diam. (approx.). E_r varies between 48 and 58, averaging 53. A high value usually indicates a high-speed engine.

Cubic Feet per Car-mile (Q_c) is an economy factor showing how much mixture the engine pumps per car mile traveled. It assumes all engines to have the same carburetion, breathing capacity and distribution charactertistics, a safe assumption when comparing most automobile engines over a period of a few years. This factor represents gasoline used per mile of car travel. The equation is: $Q_c = (5280\,R/C) \times (D/2)$. $D =$ engine displacement, cu. ft.; R and C are as before. In present (1935) passenger cars, Q_c depends on the size of engine, and varies as follows in the several price fields: Low from 135 to 220; lower medium, from 190 to 250; upper medium, from 230 to 350; high, from 270 to 380. A low value indicates probable low fuel consumption, in miles per gallon. As a rule, small European cars are designed for high economy, and Q_c may be as low as 63. This value was obtained on a Morris Minor Coach of 51.7 cu. in. displacement.

Cubic Feet per Ton-mile, (Q_t) is a performance factor showing the amount of mixture pumped per ton of vehicle moved per mile, and represents the power-weight ratio. It assumed that all engines develop the same brake M.E.P. at the same road speeds. This is sufficiently accurate for purposes of comparison. The equation is

$$Q_t = (5280\,RD/2C) \times (2000/W),$$

where $W =$ car weight, lb., loaded with passengers, gas, oil, water and equipment; R, D and C are as before. In American passenger cars, Q_t ranges from 100 to 135; average, 112. A high value indicates high performance as to acceleration high gear, hill climb and maximum speed. In European small cars, Q_t may be as low as 63, the value for the Morris Minor.

Feet of Piston Travel per Car-mile, T_p, is a factor giving wear in the engine. The equation is $T_p = (5280\,R/C) \times 2\,S$, where $S =$ stroke, ft.; R and C are as before. T_p ranges from 1750 to 2725 for American cars. European cars have values of about the same range.

COEFFICIENT OF FRICTION BETWEEN TIRES AND ROAD SURFACE is the force required to cause tires to slide, divided by normal pressure between tires and road. It has two values: F, when sliding is impending; f, when sliding is under way. See Table 1. If $P_s =$ force required to start sliding, $P_u =$ force required for uniform sliding, $w =$ normal pressure between tire and road, then $F = P_s/w$ and $f = P_u/w$. With hard-packed snow on pavement, F may be from 0.17 to 0.20; f from 0.12 to 0.15. Ice and sleet on pavement reduce F to as low as 0.08, and f to 0.07.

THE RESISTANCE OF MOTOR VEHICLES on level road is made up of rolling resistance and air resistance. Grade resistance is an additional factor on hills. Knowing

Table 1.—Average Coefficients of Friction between Tires and Road Surfaces, Sliding in the Line of Travel

(*Bulletin* No. 88, Iowa State College)

Surface	Wet Road Surface		Dry Road Surface	
			F	
Portland cement concrete, 2 years old........	0.89	0.81	0.96	0.85
Portland cement concrete, 5 years old, greasy.	.96	.89	.64	.54
Asphaltic concrete........................	.87	.79	.86	.82
Bitulithic.................................	.69	.61	.73	.72
Wood block...............................	.82	.75	.81	.60
Brick-monolithic..........................	.91	.82	.60	.54
Brick, sand filled.........................	.87	.79	.62	.53
Brick, asphalt filled.......................	.85	.75	.81	.75
Gravel....................................	.75	.65	.79	.68
Earth......................................	.63	.65		

total resistance, the horsepower required to propel the vehicle at any speed may be calculated.

Rolling Resistance depends on road surface characteristics, type and condition of tires and friction of wheel bearings. It ranges from 12 to 30 lb. per 1000 lb. of vehicle on smooth roads. An average value for smooth concrete is from 17 to 20 lb. On rough roads, gravel, or dirt, it may be 100 lb. or more. It is fairly constant throughout the speed range.

Air Resistance depends on the aerodynamic characteristics of the body. It varies as (speed)^2, and is, therefore, of great importance at high speed; at speeds above 50 m.p.h., air resistance is the major part of total resistance. Air resistance is directly proportional to the projected frontal area, the shape of the body and (wind velocity)^2. Let R_a = air resistance, lb., K = coefficient of resistance, A = projected frontal area, sq. ft., V = air velocity past body, m.p.h. Then $R_a = KAV^2$. K varies with the shape of the body, being greatest for shapes with sharp corners and large flat areas normal to the wind, and least for streamline shapes. Lay gives the value of K as 0.00155 for a 1930 sedan as found from model tests. A good average for sedans as found by tests on full size bodies is 0.0019 to 0.0020. Material reductions are possible by using large radii for all corners.

Frontal areas range from 26 sq. ft. for small cars to 33 sq. ft. for the largest. This factor is determined largely by the passenger seating capacity and can be changed but little.

If R_t = total resistance, lb., W = total weight of car and passengers, lb., r = rolling resistance, lb. per 100 lb. of car, K, A, V as before, then $R_t = (Wr/1000) + KAV^2$.

Power Required to Drive the Car at any speed is Hp. $= R_t(V/375)$.

2. ENGINE DETAILS

THE COOLING SYSTEM dissipates waste heat. Several types are possible. Water-cooling, with either thermosyphon or pump circulation has been most used. Direct air cooling has been successful. An evaporative cooling system, in which the water surrounding cylinders is always at boiling temperature also is possible. In this system the radiator is a condenser for the steam.

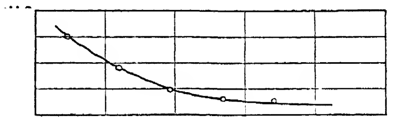


Fig. 1. Heat Dissipation from Typical Passenger Car Radiator Section

In design of the water-cooled system the most severe load is used, *viz.*, that imposed by air temperature of 110° F., and car speed of 15 to 25 m.p.h. (1000 engine r.p.m.) with wide-open throttle. The quantity of heat to be dissipated varies with engine design, being greatest for L-head, and least for valve-in-head engines. Specific heat of cooling water varies between 45 and 75 B.t.u. per B.Hp. per min. at 1000 r.p.m. Design of the system depends on variables associated with the fan, water pump and radiator section. Fig. 1 shows heat dissipation from a typical passenger car radiator section. A temperature rise of 80° F. is the maximum possible with existing sections, and is the difference between air and upper radiator tank water temperatures. For best operation, upper tank temperatures should be approximately 165° F., although under most severe load 210° F. may be tolerated.

A propeller fan delivers the proper amount of air through the radiator. Fan horse-

power varies as (speed)^3, while delivery, cu. ft. per min., varies directly with speed. Fig. 2 shows test results of three typical fans mounted behind a radiator section.

Thermostats to obtain quick warming of cylinder walls, especially in winter, prevent crankcase dilution. They begin to open at 135° F. and are fully open at 140° F. Radiator shutters, thermostatically operated, also are used to obtain rapid warming.

ANTI-FREEZE.—Denatured alcohol solutions, to which there are two principal objections, are most generally used. The solution evaporates, especially on heavy runs; unless it is tested periodically and sufficient replacement alcohol added, both motor and radiator are apt to be damaged by freezing. Alcohol solution or vapor will damage car finish. The strength of solution necessary to protect against freezing depends on the temperature. The following are the proportions of 90% (180° proof) denatured alcohol and water required at various temperatures.

Freezing Point......	+10° F.	0° F.	−10° F.	−20° F.	−30° F.
Parts of Alcohol....	2 1/2	3			
Parts of Water.....	5 1/2	5	4 1/2	4	3

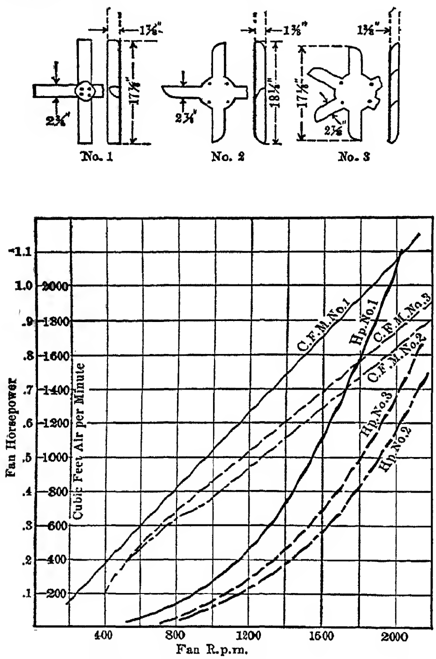


FIG. 2. Fan Tests. Horsepower and Air Delivery

Distilled glycerine and ethylene glycol solutions cost more than alcohol, but do not evaporate; only water need be added to replace evaporation. Solution lost mechanically, by leakage, foaming, etc., must be replaced by fresh solution. These solutions, under ordinary conditions, do not injure car finish. The principal objection to glycerine and ethylene glycol solutions is their tendency to loosen iron rust and scale in the water passages of the engine, thus making difficult the maintenance of leakproof connections. The entire cooling system must be cleaned and flushed before glycerine or ethylene glycol is used, and cylinder head gaskets and pump packing must be tightened or replaced. Cylinder head gaskets must be tight to prevent solution leaking into the crankcase, where it may gum and cause sticking of moving parts. Pump packing must be tight to prevent air being drawn into the cooling system, otherwise, foaming and other difficulties may result.

Salt solutions, as calcium or magnesium chloride, sodium silicate, etc., honey, glucose and sugar solutions and oils are unsatisfactory for automobile radiators.

CYLINDER ARRANGEMENT AND NUMBER are usually determined by the economics of manufacture, and by vibration characteristics. Fig. 3 compares the forces in engines of most of the usual cylinder combinations. The analysis was made on engines of the same displacement and the forces are plotted to scale.

Cylinders of passenger cars usually are cast iron, cast en block. In truck and bus engines, steel cylinder liners often are used. The minimum thickness of section recommended for cast iron is $^3/_{16}$ in., but up to $^1/_4$ in. may be used for large-bore cylinders for passenger cars and trucks. Average thickness for passenger cars is $^7/_{32}$ in. Minimum thickness of steel sleeves with radial stiffening ribs is $^1/_8$ in.; inserted steel sleeves usually are $^3/_{16}$ in. thick. Cylinder blocks should have a Brinell hardness of 187 to 202 in the cylinder bore and at the valve seats. If these parts are so located in the casting as to cool slowly, they will be hard at the edges, corners and thin sections, and of hard machinability if made of unalloyed iron giving above hardness values. Additions of Ni and Cr

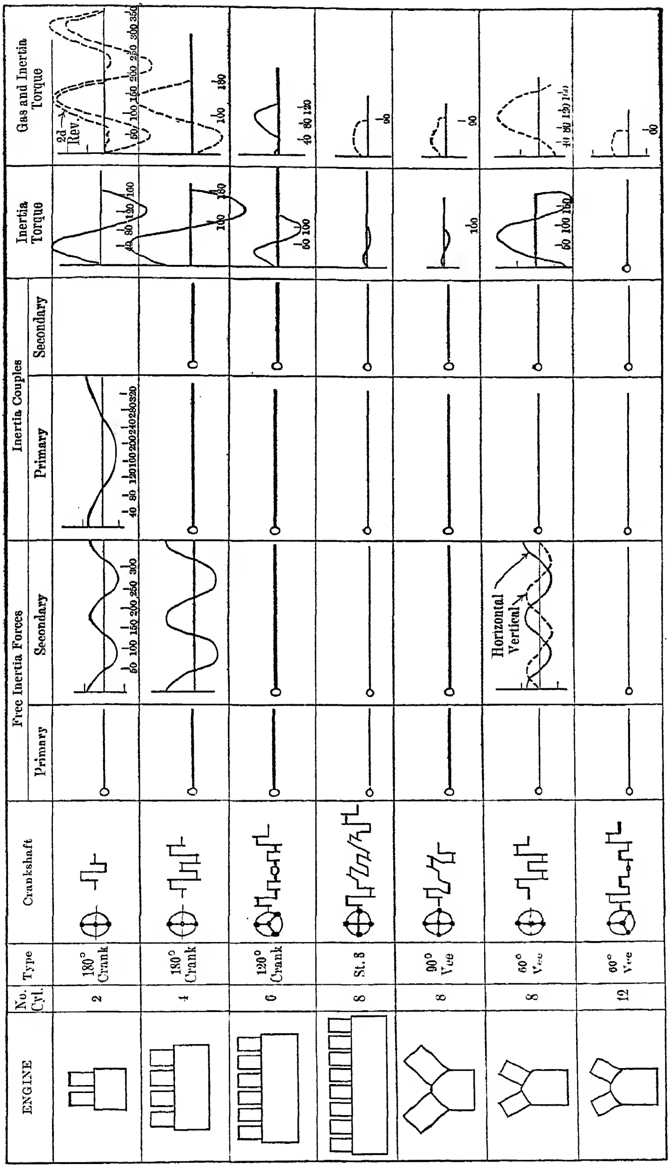


Fig. 3. Forces Acting in the Engine

Table 2.—Compositions of Cast Irons for Automotive Engine Cylinders

(A. L. Boegehold, Regional Meeting, A.S.T.M., Detroit, March, 1930)

Cylinder Iron No.	C, total	Mn	P, max.	S, max.	Si	Cr	Ni
1	3.15–3.30	0.55–0.75	0.20	0.12	2.00–2.40	0.06–0.16	0.03–0.08
2	3.15–3.30	0.55–0.75	0.20	0.12	2.00–2.40	0.05–0.07	0.25–0.35
3	3.15–3.50	0.50–0.80	0.20	0.12	1.75–2.25	....	1.00–1.50

counteract this tendency. An addition of 1 1/2 % Ni also lowers the coefficient of expansion. Tensile strength should be 35,000 lb. per sq. in. when cast in a 1/2-in. section. Table 2 gives composition of cast irons much used for cylinders.

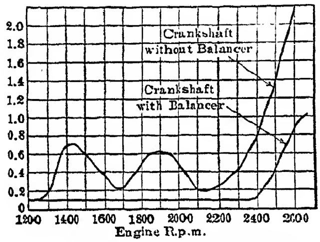


Fig. 4. Torsional Vibration of Crankshaft

CRANK-SHAFTS for automobile engines are forgings of plain carbon steel (S.A.E 1045). Alloy steels usually are unnecessary, since bearing sizes limit the diameter to a value where plain carbon steel is strong enough. A main bearing between each crank throw, between each 2 throws or each 4 throws all are in successful use. Limiting the number of bearings reduces center distances between cylinders and makes possible a shorter engine.

Crank-shaft Vibration.—Crank-shafts are counterweighted to reduce centrifugal forces and bearing loads. In practice, 50% of the centrifugal forces should be balanced by the counterweights to obtain satisfactory smoothness. Static and dynamic balance is required. The limits vary from 1/4 to 1 oz.-in. In long crank-shafts (6 or more cylinders), torsional vibration at resonant speeds produces vibration which must be eliminated. 4-cylinder or V-8 engines usually have a short stiff crank-shaft with a natural frequency above normal driving speeds. Torsional vibration in an automobile engine crank-shaft is produced by a resonance of the torque impulses of the engine with the natural frequency of torsional vibration of the crank-shaft; they are excited by the gas pressure on the piston and by inertia forces of reciprocating parts. Gas pressures recur every two engine revolutions and inertia forces every revolution. The forces are periodic and may be repre-

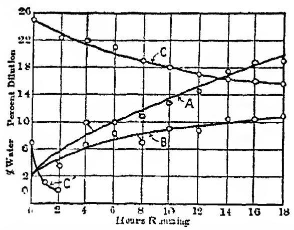


Fig. 5. Dilution of Oil vs. Hours of Running. *A*. Fresh oil, standard operation. *B*. Fresh oil, 1/3 of carburetor air drawn through crank-case. *C*. Polluted oil, 1/3 of carburetor air drawn through crank-case. *C'*. Water in oil, 1/3 of carburetor air drawn through crank-case. Fuel used 70% gasoline, 30% kerosene.

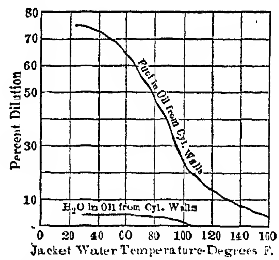


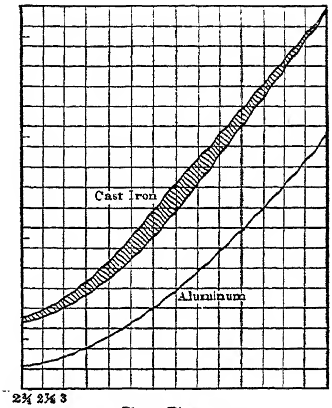
Fig. 6. Effect of Jacket Water Temperature on Dilution of Oil from Cylinder Walls

sented by harmonic sine functions, $f \times$ r.p.m., where f = frequency = 1/2, 1, 1 1/2, 2, 2 1/2, 3, 3 1/2, 4, 4 1/2, 5, 5 1/2, 6, etc. The crank-shaft responds to excitation by the gas or inertia forces with a vibration whose frequency equals that of the exciting forces. The natural frequency of torsional vibration of a crank-shaft is determined by the shaft stiffness and inertia of the shaft, connecting-rods and pistons. It is proportional to (stiffness ÷ inertia). Since long crank-shafts cannot be built stiff enough to make the natural frequency of torsional vibration occur above normal driving range, a balancer is necessary. A torsion balancer or Lanchester damper are two types generally used. Fig. 4 shows the amplitude of vibration of a crank-shaft with and without a torsion balancer.

CRANK-CASE AND CRANK-CASE VENTILATION.—The upper half of the crank-case may be either cast iron or aluminum alloy, cast with the cylinders or separately. It must

be sufficiently rigid to prevent deflection, due to engine operation, misaligned crank-shaft bearings, camshaft bearings and camshaft drive mechanism. The lower half of the crank-case is an oil reservoir, holding 5 to 10 qt., depending on size of engine. American practice provides ventilation through the crank-case to remove water and light ends of the fuel which blow past the pistons. This water is washed from the walls by the oil and deposited in the crank-case. During choking, especially in cold weather, large quantities of fuel also may be blown past the pistons into the oil. Water in the crank-case tends to corrode bearing surfaces of the crank-shaft, piston-pins, cylinder walls and valve gear. Corrosion is most severe with gasolines high in sulphur. A blast of air through the crank-case holds oil dilution and water content to a minimum. See Fig. 5.

Fig. 6 shows tests of fuel dilution and water dilution on cylinder walls of an automobile engine which had stood out all night before the test. Initial jacket water temperature was 5° F.



G. 7. Relative Weight of Cast-iron and Aluminum Pistons

PISTONS.—The most common piston material is cast iron or aluminum alloy; magnesium also has been used. Cast iron gives long life with little wear; it can be fitted to closer limits and distorts less than aluminum. Aluminum has the advantage of light weight and higher heat conductivity. Fig. 7 compares the weights of typical cast-iron and aluminum pistons. The weight of the piston assembly is held within limits of $^1/_{16}$ to $^3/_{16}$ oz. to insure good engine balance. Clearance for cast-iron pistons may be taken as from 0.00075 to 0.001 in. per inch of cylinder bore, and approximately double this for aluminum. No exact rule is possible, as clearance depends on piston design and the duty to which it is subject. Some manufacturers plate cast-iron pistons with tin to prevent scoring and seizing. The piston then may be fitted as closely as 0.0005 to 0.001 in. or have a light drag fit in the bore.

The piston is fitted with three or four piston rings. The upper rings are plain; the lower one or two are oil control rings. The plain rings are from $^3/_{32}$ to $^5/_{32}$ in. wide; the oil rings from $^1/_8$ to $^3/_{16}$ in.

VALVES AND VALVE MECHANISM.—Fig. 8 shows the most common arrangement of valves. The I-head, commonly known as valve-in-head or overhead valve, gives a

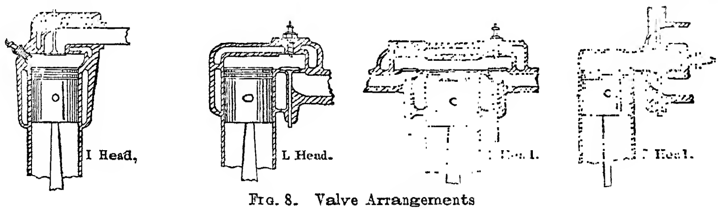


Fig. 8. Valve Arrangements

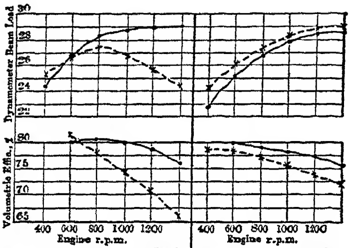
simple combustion chamber with a minimum heat loss to cooling water due to the small water jacketing. The L-head gives simplified valve action. The T-head requires two camshafts, but permits large valves and low lifts. It requires the greatest amount of water jacketing. The F-head has the intake valve over the exhaust valve, and is a combination of overhead and L-valves.

Valve Material must withstand high temperature without corrosion. For intake valves, which operate cooler than exhaust valves, S.A.E. 3140 or chrome-nickel steel is used. Exhaust valves are made of Silchrome-, chrome-nickel- or tungsten-steel.

Valve Mechanism Design depends on many variables. The valve timing selected can give power either at high or at low engine speeds. To obtain best performance at high speed entails a sacrifice at low speed and *vice versa*. A compromise must be made to obtain the results desired. Fig. 9 shows the effect of varying intake timing on a single-cylinder

test engine, 3 1/8 in. bore, 4 1/2 in. stroke, with overhead valves. Fig. 10 is a timing diagram for a typical passenger car engine.

Cam shape has much to do with the operation of the valves. Moving parts should be of light weight to reduce inertia forces. Good valve follow should be sought. This requires: 1. A maximum of acceleration in as short a time as rigidity of the system will permit; 2. A minimum of deceleration continued over as long a time as is consistent with (1); 3. A minimum of reversals of forces in the acceleration curve. A cam contour so designed should give maximum volumetric efficiency and optimum valve follow. See Fig. 69.



Inlet	Opens	Closes	Total Opening	Opens	Closes	Total Opening
x---x	14°BTC	28°ABC	222°	14°BTC	68°ABC	262°
o——o	14°BTC	68°ABC	262°	5°BTC	77°ABC	262°
Exhaust	77°BBC	32°ATC	289°	77°BBC	32°ATC	289°

G. 9. Intake Timing and Engine Performance

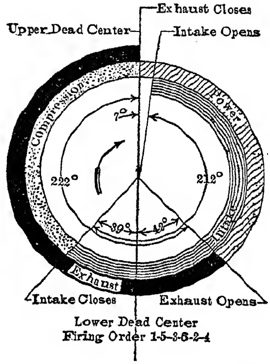


FIG. 10. Typical Valve Timing Diagram

Valve Springs.—Valves are closed by helical coiled springs. One of the greatest difficulties with such a spring is its tendency to vibrate or "surge" at definite speeds, depending on the natural vibration frequency of the spring. This frequency can be calculated by Ricardo's formula for helical springs, $n = 531\sqrt{R/W}$, where n = frequency, vibrations per min., R = scale of the spring = load, lb., necessary to deflect it 1 in., W = weight of active part, lb. If (cam r.p.m. × an integer) = n, a resonant vibration results. At low speeds, the amplitude is small but at some operating speeds it becomes large enough to cause the coils to close and the springs will produce a sound.

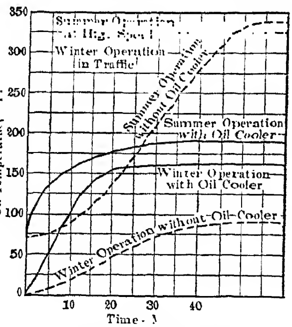


FIG. 11. Effect of Water-cooled Oil-cooler on Oil Temperature

Formulas used to calculate round wire springs are:

$$P = \pi S d^3/8d = fGd^4/8\,D^3N; \quad R = GD^4/8\,D^3N;$$

$$S = fGd/\pi D^2N; \quad f = \pi D^2NS/Gd, \quad n = S/4.78f,$$

where P = pressure of spring, lb.; R = scale of spring (see above); S = stress, lb. per sq. in.; G = modulus of elasticity in torsion; f = deflection of spring, in., under load P; d = diameter of wire, in.; D = pitch diameter of coil, in.; N = number of active turns of spring; n = frequency, cycles per min.

High-carbon steel, low in manganese, or chrome-vanadium spring steels are used.

AIR FLOW THROUGH VALVES.—Valve port size, timing and lift must be proportioned correctly to obtain high volumetric efficiency. Valve lift usually is about 1/4 port diameter; in slow speed engines a material gain in air flow can be obtained by lifts of 1/3 port diameter. The air flow through a valve port with 30° seats is greater than through one with 45° seats. Gas velocity through the port is given by the approximate formula, $V = D^2sN/8qdh$, where D = cylinder diam., in.; V = gas velocity, ft. per sec.; s = piston stroke, in.; d = port diam. in.; h = mean valve lift (in. from valve lift diagram); q = period of opening, degree of crankshaft rotation. This formula assumes the valve to be open for the duration shown on the diagram, and the valve

annulus to be a cylindrical surface. If the valve is tapered a correction factor should be applied as follows:

For 30° seats, $A_1 = A_2\{0.866 + (0.375)(h/d)\}$

for 45° seats, $A_1 = A_2\{0.707 + (0.353)(h/d)\}$

$V_2 =$ [illegible]

where V_2 = velocity corrected for seat angle; V_1 = velocity, assuming a flat seat; A_1 = area of conical annulus, sq. in.; A_2 = area of cylindrical annulus, sq. in.

ENGINE LUBRICATION must prevent metal-to-metal contact and remove excess heat from the bearings. Oil is fed to bearing surfaces by force or splash. Dippers on the connecting-rods pick up the oil from troughs kept full by a circulating pump, and splash it to the bearings. When pressure is used, a gear pump forces oil through holes in the crank-shaft to the main and connecting-rod bearings. The connecting-rod may be drilled to allow oil to flow to the piston-pin. A pressure of 30 to 40 lb. per sq. in. at 30 m.p.h. car speed is usual.

Oil Coolers may be used to control oil temperature. In summer, they keep the temperature low and in winter they provide a quick oil warm-up and a higher stabilized temperature. Fig. 11 shows the effect on summer and winter oil temperatures of a water-cooled type oil cooler.

Oils for Lubricating automobile engines are classified by S.A.E. standards of viscosity. S.A.E. 30 is used for normal driving conditions, S.A.E. 20 and 10 for winter temperatures, and S.A.E. 40 to 60 for high-speed driving in warm weather. 10-W and 20-W are special oils for sub-zero winter driving. Viscosity is the most important property of oil from the automobile standpoint. It affects starting friction, oil circulation, oil consumption, oil pumping, smoking in exhaust, fouling of spark plugs, carbon formation in the engine, oil pressure, crank-case dilution and noise. Volatility also is important at high speeds. Fig. 12 shows the effect of high speed on oil consumption of two oils of different volatility.

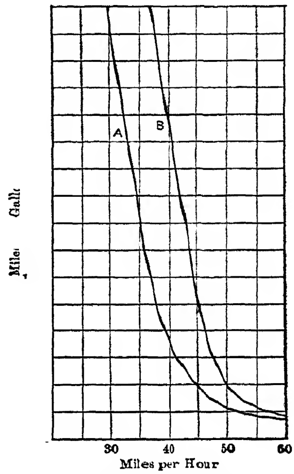


Fig. 12. Effect of Car Speed on Oil Consumption

3. ENGINE DESIGN

KINEMATICS OF THE ENGINE (Air Service Information Circular, No. 421, H. Caminez and C. W. Isler.).—To determine the principal forces acting in the engine, the motion of the pistons and the connecting-rods relative to the crank-shaft is found; the angular velocity is taken as constant because of the large inertia of the flywheel and uniformity of torque in multi-cylinder engines. In Fig. 13*a*, which is a diagram of the usual arrangement of pistons, connecting-rod, and crank-shaft, let L = connecting-rod length, center to center in., $= BD$; R = crank radius, in. $= OD$; θ = crank angle from top center position, deg.; ϕ = angle of connecting-rod with center line of cylinder, deg.; s = piston travel, in. $= AB$.

Piston- and Connecting-rod Position for various crank angles can be found by a diagram drawn to scale or by mathematical analysis. With the arrangement of Fig. 13*a*,

$$s = (R - R\cos\theta) + (L - L\cos\phi) \quad [1]$$

and $\phi = \sin^{-1}(R/L)\sin\theta. \quad [2]$

Combining [1] and [2], piston travel =

$$s = R\left[1 - \cos\theta + (L/R) - \sqrt{(L/R)^2 - \sin^2\theta}\right] \quad [3]$$

and percent of piston travel

$$= {}^1/_2\left[1 - \cos\theta + (L/R) - \text{[illegible]}^2 - \sin^2\theta\right] \quad [4]$$

Equation [4] has been solved at every 10° of crank angle for the usual ratios of found in automobile engines. See Table 3.

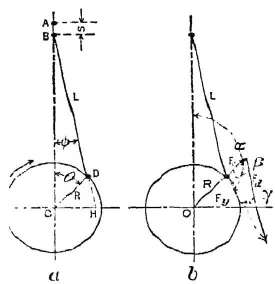


Fig. 13. Piston, Crankshaft and Connecting Rod Relations

Table 3.—Percent of Piston Travel for Various Ratios of L/R.

Crank Angle, deg.		Values of L/R							
		3.0	3.2	3.4	3.6	3.8	4.0	4.2	4.4
		Percent of Piston Travel							
0	360	0.0	0.0	0.0	0.0	0.0	0.0	0.0	0.0
10	350	1.0	1.0	1.0	1.0	0.9	0.9	0.9	0.9
20	340	4.0	3.9	3.8	3.8	3.7	3.7	3.7	3.6
30	330	8.8	8.7	8.6	8.5	8.4	8.3	8.2	8.1
40	320	15.2	15.0	14.8	14.6	14.4	14.3	14.2	14.1
50	310	22.8	22.5	22.3	22.1	21.8	21.6	21.4	21.2
60	300	31.4	31.0	30.6	30.3	30.0	29.7	29.5	29.3
70	290	40.5	40.0	39.6	39.2	38.8	38.5	38.2	37.9
80	280	49.6	49.1	48.7	48.2	47.8	47.5	47.2	46.9
90	270	58.6	58.0	57.5	57.1	56.7	56.4	56.1	55.8
100	260	67.0	66.5	66.0	65.6	65.2	64.8	64.5	64.2
110	250	74.7	74.2	73.8	73.4	73.0	72.7	72.4	72.1
120	240	81.4	81.0	80.6	80.3	80.0	79.7	79.5	79.3
130	230	87.1	86.8	86.5	86.3	86.0	85.8	85.6	85.3
140	220	91.8	91.6	91.4	91.2	91.0	90.9	90.8	90.7
150	210	95.4	95.3	95.2	95.1	95.0	94.9	94.8	94.7
160	200	98.0	97.9	97.8	97.8	97.7	97.7	97.7	97.7
170	190	99.5	99.5	99.4	99.4	99.3	99.3	99.3	99.3
180	180	100.0	100.0	100.0	100.0	100.0	100.0	100.0	100.0

Table 4.—Crank Angle Factor for Piston Velocity

Crank Angle, deg.		Values of L/R							
		3.0	3.2	3.4	3.6	3.8	4.0	4.2	4.4
0	360	0.0	0.0	0.0	0.0	0.0	0.0	0.0	0.0
10	350	0.231	0.227	0.224	0.221	0.218	0.216	0.214	0.212
20	340	.450	.443	.437	.432	.427	.423	.419	.415
30	330	.646	.637	.629	.622	.615	.609	.604	.599
40	320	.811	.799	.790	.782	.775	.768	.762	.756
50	310	.936	.924	.915	.906	.898	.891	.885	.880
60	300	1.017	1.007	.998	.990	.983	.977	.971	.966
70	290	1.053	1.045	1.038	1.032	1.027	1.022	1.018	1.015
80	280	1.045	1.041	1.037	1.034	1.031	1.029	1.027	1.025
90	270	1.000	1.000	1.000	1.000	1.000	1.000	1.000	1.000
100	260	0.925	0.929	0.932	0.935	0.938	0.941	0.943	0.945
110	250	.827	.835	.841	.847	.852	.857	.861	.865
120	240	.715	.725	.734	.742	.749	.755	.761	.766
130	230	.596	.608	.617	.626	.634	.641	.647	.652
140	220	.475	.486	.495	.503	.511	.518	.524	.530
150	210	.354	.363	.371	.378	.385	.391	.396	.401
160	200	.234	.241	.247	.252	.257	.261	.265	.269
170	190	.117	.120	.123	.126	.129	.131	.133	.135
180	180	0.0	0.0	0.0	0.0	0.0	0.0	0.0	0.0

Table 5.—Crank Angle Factors for Piston Acceleration.

Crank Angle, deg.		Values of L/R							
		3.0	3.2	3.4	3.6	3.8	4.0	4.2	4.4
0	360	1.333	1.313	1.294	1.278	1.263	1.250	1.238	1.227
10	350	1.298	1.279	1.261	1.246	1.232	1.220	1.209	1.198
20	340	1.195	1.179	1.165	1.153	1.142	1.131	1.122	1.114
30	330	1.033	1.022	1.013	1.005	0.998	0.991	0.985	0.980
40	320	0.824	0.820	0.817	0.814	.812	.809	.807	.806
50	310	.585	.589	.592	.595	.597	.600	.602	.604
60	300	.333	.344	.353	.361	.368	.375	.381	.386
70	290	.087	.103	.117	.129	.140	.151	.160	.168
80	280	− .139	− .120	− .103	− .087	− .073	− .061	− .050	− .040
90	270	− .333	− .313	− .294	− .278	− .263	− .250	− .238	− .227
100	260	− .486	− .467	− .450	− .435	− .421	− .409	− .397	− .387
110	250	− .597	− .581	− .567	− .555	− .544	− .534	− .524	− .516
120	240	− .667	− .656	− .647	− .639	− .632	− .625	− .619	− .614
130	230	− .701	− .697	− .694	− .691	− .688	− .686	− .684	− .682
140	220	− .708	− .712	− .715	− .718	− .720	− .723	− .725	− .727
150	210	− .699	− .710	− .719	− .727	− .734	− .741	− .747	− .753
160	200	− .684	− .700	− .714	− .727	− .738	− .749	− .757	− .765
170	190	− .672	− .691	− .708	− .724	− .738	− .750	− .761	− .771
180	180	− .666	− .688	− .706	− .722	− .737	− .750	− .762	− .773

Velocity.—Let v_c = crank-pin velocity, ft. per sec.; v_p = piston velocity, ft. per sec.; N = r.p.m.; R = crank radius, in.; $(d\theta/dt)$ = angular velocity of connecting-rod; f_v = crank angle factor for piston velocity. Then, referring to Fig. 13*b*, and considering linear velocity of the crank-pin as

$$v_c = R(d\theta/dt) = (2\pi N/60) \times (R/12), \qquad [5]$$

$$\frac{d\theta}{dt} = \frac{R}{L}\,\frac{\cos\theta}{\cos\phi}\,\frac{d\theta}{dt} = \frac{\cos\theta}{\ } = v_- =$$

$$= v_c f_v; \qquad [7]$$

$$f_v = \sin\theta + \cos\theta\tan\phi = \sin\theta\,[1 + \{\cos\theta/\sqrt{(L/R)^2 - \sin^2\theta}\}] \qquad [8]$$

The crank angle factor for piston velocity, f_v, may be found graphically. In Fig. 13*a* it is OH/OD. Table 4 gives values of f_v for ratios of L/R generally used in automobile engines. Since ϕ, the connecting-rod angle, is small, the resulting error from substitution of $\sin\phi$ for $\tan\phi$ in [7] and [8] will be small. Making this substitution the approximate equations are:

$$v_p = v_c\{\sin\theta + {}^1/_2\,(R/L)\sin 2\theta\}; \qquad [9]$$

$$f_v = \sin\theta + {}^1/_2\,(R/L)\sin 2\theta. \qquad [10]$$

Acceleration.—The exact expression for piston acceleration for the arrangement in Fig. 13*a* is complex and not commonly used. The approximate equation in general use is

$$a = (v_c^2/R)\{\cos\theta + (R/L)\cos 2\theta\} = (v_c^2/R) \times f_a, \qquad [11]$$

where a = linear acceleration of piston; f_a = crank angle factor for piston acceleration.

$$f_a = \cos\theta + (R/L)(\cos^2\theta/\cos^3\theta) - \sin\theta\tan\phi \quad \text{(Exact)} \qquad [12]$$

$$f_a = \cos\theta + (R/L)\cos 2\theta \quad \text{(Approx.)} \qquad [13]$$

Table 5 gives values of f_a for ratios of L/R generally used in automobile engines. Table 6 is a comparison of exact and approximate values.

INERTIA AND CENTRIFUGAL FORCES are caused by the motion of the pistons and connecting-rods. In a fixed-cylinder engine, piston motion is linear; the resulting forces are purely inertia forces. The motion of the connecting-rods is more complex. For an exact solution of the forces produced by the rod, it is necessary to consider the components of its motion. The motion of the rod may be analyzed as a translation of its center of gravity, with the linear velocity and acceleration of the piston combined with an angular velocity and acceleration about the piston-pin. This motion of the rod sets up in each of its elements three separate forces: *a*, an inertia force due to linear acceleration in the direction of the cylinder axis; *b*, a centrifugal force due to angular velocity about the piston-pin; *c*, an inertia force due to angular acceleration about the piston-pin. These forces are very closely approximated by assuming the mass of the connecting-rod to be divided between the piston-pin and the crank-pin in inverse

Table 6.—Comparison of Values of Factors for Piston Velocity and Acceleration by Approximate and Exact Formulas

Crank Angle, deg.	Factors for Piston Velocity for $L/R = 4$		Factors for Piston Acceleration for $L/R = 4$	
	Exact Equation	Approximate Equation	Exact Equation	Approximate Equation
0	0.0	0.0	1.2500	1.2500
10	0.2164	0.2164	1.2202	1.2197
20	.4226	.4224	1.1336	1.1312
30	.6091	.6083	0.9950	0.9910
40	.7676	.7659	.8139	.8094
50	.8914	.8891	.6026	.5994
60	.9768	.9742	.3572	.3750
70	1.0223	1.0200	.1468	.1505
80	1.0289	1.0276	− .0682	− .0613
90	1.0000	1.0000	− .2583	− .2500
100	0.9407	0.9420	− .4153	− .4085
110	.8571	.8594	− .5372	− .5335
120	.7552	.7578	− .6248	− .6250
130	.6406	.6429	− .6380	− .6862
140	.5180	.5197	− .7182	− .7226
150	.3909	.3917	− .7370	− .7410
160	.2614	.2616	− .7458	− .7482
170	.1308	.1308	− .7494	− .7499
180	0.0	0.0	− .7500	− .7500

proportion to the distance of the respective pins from center of gravity of the rod. The crank-pin portion produces a centrifugal force, the piston-pin portion produces an inertia force. To obtain the weight at either end, support the rod on knife edges directly over the center line of the bearings, the axis of the rod being horizontal. The knife edge under the end to be weighed rests on scales. Results can be verified by comparing the sum of the weights of the two ends with the total weight of the rod. The centrifugal force acting at the crank-pin in the direction of the crank-throw is

$$F_c = 0.0000284 W_c RN^2, \qquad [14]$$

where F_c = centrifugal force, lb.; W_c = weight of lower end of connecting-rod, lb.; R = crank radius, in.; N = r.p.m. The inertia force acting in the direction of the cylinder is

$$F_i = -0.0000284 W_i RN^2 f_a \qquad [15]$$

where F_i = inertia force, lb.; W_i = reciprocating weight = weight of piston assembled plus upper end of connecting-rod, lb.; f_a = crank angle factor for piston acceleration.

Resultant Forces on Piston.—The gas pressure acting on the piston is obtained at various crank angles from the indicator diagram. Fig. 14 is a typical indicator card for an automobile engine. Two values of indicated mean effective pressure are given. If an experimental indicator card is not available, a theoretical card can be calculated. The total gas force is this pressure multiplied by piston area. Equation [15] determines the inertia force at various crank angles. Then if F_a = resultant force along cylinder axis, F_g = force on piston due to gas pressure, F_i = inertia force,

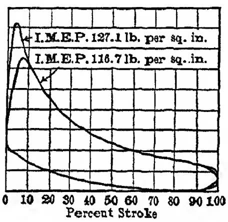


FIG. 14. Typical Indicator Card

$$F_a = F_g + F_i \qquad [16]$$

Piston Side Thrust due to the force acting along the cylinder axis is

$$F_s = F_a \times \tan\phi, \qquad [17]$$

where F_s = piston side thrust, lb.; F_a = resultant force along cylinder axis, lb.; ϕ = connecting-rod angle, deg. For the arrangement of Fig. 13*a*, F_s may be expressed in terms of crank angle θ by

$$F_s = F_a \{\sin\theta / \sqrt{(L/R)^2 - \sin^2\theta}\}. \qquad [18]$$

Piston side thrust is found throughout a complete cycle and is plotted *vs.* crank angle and *vs.* piston travel. The area of the latter curve is proportional to the piston friction loss if the coefficient of friction remains constant. From the average height of this curve, the average side thrust during the power stroke and during the complete cycle is determined.

Piston side pressure, lb. per sq. in., equals (total piston side thrust) ÷ (projected bearing area). Since the piston diameter usually is relieved above the lower piston ring, only that portion below the lower ring is considered in determining effective bearing area.

Torque.—The torque due to forces acting in the cylinder at any instant during the cycle is

$$T = F_a \times R \times f_v, \qquad [19]$$

where T = torque, lb.-ft.; F_a = resultant force along cylinder axis, lb.; R = crank radius, ft.; f_v = crank angle factor for piston velocity. The resultant torque in a multi-cylinder engine is the algebraic sum of the instantaneous torques of the individual cylinders. To find this resultant, the angular relation of the cycles in the various cylinders must be considered. This analysis gives the indicated torque, as frictional forces are not considered.

The mean torque can be obtained from a curve of torque *vs.* crank angle by using a planimeter. The ratio of the maximum to the mean torque of the engine is determined and used in later calculations.

Resultant Force on Crank-pin is found by combining graphically the resultant force along the connecting-rod axis with the centrifugal force due to the weight of the lower end of the connecting rod. Use equation [14]. Let F_d = resultant force along the connecting-rod axis; F_a = resultant force along cylinder axis; ϕ = connecting-rod angle. Then

$$F_d = F_a / \cos\phi. \qquad [20]$$

In equation [20] a plus (+) force denotes a force producing compression in the rod, a negative (−) force one producing tension. With the arrangement shown of Fig. 13*a*, F_d can be expressed in terms of the crank angle θ by

$$\overline{n^2\,\theta}. \qquad [21]$$

Fig. 13*b* shows a graphic determination of the resultant force on the crank-pin. F_c is laid off to scale along the center line of the crank-throw. F_d represents a positive force acting along the connecting-rod and is drawn parallel to it. The resultant force, F_r, represents, to scale, the magnitude of the force acting on the crank-pin at the angle shown. The direction of F_r with respect to engine axis, crank-throw and connecting-rod is given by angles α, β and γ respectively. F_r is obtained at various crank angles throughout the cycle, and plotted on a polar diagram with respect to the engine axis and crank throw, to determine maximum and mean forces. These forces give connecting-rod bearing loads.

Resultant Force on Crank-shaft Bearings.—The load distribution on the crank-shaft bearings depends on the rigidity of the crank-shaft and the crank-case, the alignment of the bearings, and the clearances between the journal and the bearings. These factors cannot be predetermined, and an exact analysis of the load distribution between the various crank-shaft bearings is not possible where more than two main bearings are used.

An empirical method for computing the forces acting on the main bearings of a crank-shaft where there is a main bearing at each side of the crank-pin is as follows: The forces acting on the crank-shaft bearings are obtained by assuming the force at the crank-pin, together with the centrifugal force resulting from the weight of the crank-pin and crank-cheeks, to be equally divided between the two crank-shaft bearings at each side of the crank-pin. End bearings are loaded on only one side, by half of the force of the crank-pin combined with half the centrifugal force due to the crank-pin and crank-cheeks of the crank-throw. The same loading is applied on both sides of the center and the intermediate bearings.

If more than one crank-pin bearing or crank-throw is located between each pair of main crank-shaft bearings, the load distribution on the main bearings is found by treating each section of the crank-shaft between two main bearings as a uniform beam rigidly supported at the center line of the main bearing bolts. The reactions on the main bearings A and B due to a force F_r on a crank-pin located between them is

$$R_a = F_r\{b^2(3a + b)/(a + b)^3\}, \qquad [22]$$

$$R_b = F_r\{a^2(a + 3b)/(a + b)^3\}, \qquad [23]$$

where R_a and R_b = bearing reactions at bearings A and B, respectively; F_r = resultant force on crank-pin; a and b = distance between center line of crank-pin and center line of main bearings, respectively. The resultant force on a crank-shaft bearing is the vector sum of the reactions due to the crank-pin loads. In determining the resultant force, consideration must be given to the direction relative to the engine axis of each separate reaction. Both the relative positions of the crank-pins and their cyclic relation also must be considered.

BEARING ANALYSIS.—Maximum and mean bearing pressures are determined by dividing the maximum and mean resultant forces acting on the bearing by the projected area of the bearing. In determining projected bearing area, only the straight portion of the bearing length is considered as effective. The resultant pressures are expressed in pounds per square inch.

Table 7.—Bearing Data of Typical Automobile Engines

	Eight-cylinder Passenger Car Engine. Full pressure lubrication	Six-cylinder Heavy-duty Truck Engine
Cylinder dimensions, in..	$1/16 \times 4\,5/8$	$4\,7/8 \times 5\,1/2$
Maximum engine speed..	4250 r.p.m.	2100 r.p.m.
Con. rod, length, in.	$9\,3/4$	$12\,1/8$
" " bearing diam., in.	$2\,3/16$	2.5
" " " width, "	$1\,5/16$	2.25
" " " Babbitt, width, in.	$1\,1/16$	2.375
" " " av. load, lb. per sq. in.	1,600	608
" " " max. " " " "	2,500	993
" " " av. PV	63,000	16,600
" " " max. PV	98,700	27,100
At r.p.m..	4,140	2,500

Main Bearings	Front	Front Center	Center	Rear Center	Rear	Front	Front Center	Center	Rear Center	Rear
Length, in.	1.605	1.183	1.681	1.183	2.386	2.281	1.562	2.812	1.562	3.250
Diameter, in.	2.311	2.374	2.436	2.499	2.561	2.750	2.750	2.750	2.750	2.750
Average load..	450	590	730	560	280	277	466	477	466	202
Maximum load.	900	1,180	1,460	1,120	560	462	826	653	826	336
Average PV	18,300	24,800	31,400	24,800	12,750	8,310	13,980	14,310	13,980	6,060
Maximum PV	36,600	49,600	62,800	49,600	25,500	13,900	24,780	19,590	24,780	10,800
At r.p.m.	4,140	4,140	4,140	4,140	4,140	2,500	2,500	2,500	2,500	2,500

The rubbing factor PV on the bearing, lb. per sq. in. $\times$ ft. per sec., is the product of the rubbing velocity of the journal and the mean bearing pressure. The rubbing velocity in ft. per sec. is

$$V_r = \pi(D/12) \times (\text{r.p.m.}/60), \quad [24]$$

where V_r = rubbing velocity of bearing, ft. per sec.; D = bearing diameter, in. Table 7 gives bearing data for two typical automobile engines.

CONNECTING-ROD STRESS ANALYSIS.—Stress, S_c, in connecting-rod shank, due to direct compression, is calculated by Rankine's formula,

$$S_c = (F_r/A) + m\,(L^2/I_x)\,F_r, \quad [25]$$

or

$$= (F_r/A) + m\,(L_1^2/4I_y)\,F_r, \quad [26]$$

where F_r = resultant force along connecting-rod axis; A = area of shank cross section, sq. in.; I_x and I_y = moment of inertia of area about x–x and y–y axes, respectively (see Fig. 15); L = length of connecting-rod, in.; L_1 = length of shaft as in Fig. 15, in.; m = constant for steel column = 0.000526. For equal compressive strength of the shank cross-section in both planes, $I_y = (L_1/L)^2 \times (I_x/4)$.

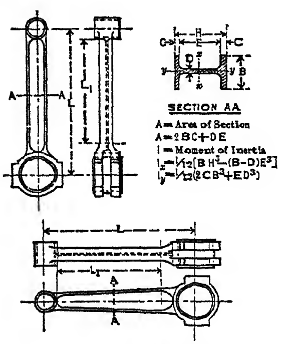


Fig. 15. Connecting Rod

Considering the rod as a uniform section extending from piston-pin center to crank-pin center, the *whipping force* that results from transverse acceleration of the rod, is approximately

$$F_w = (wAL/2g) \times (\pi N/30)^2 \times (R \sin\theta/12)$$
$$= 0.00000402ALN^2R \sin\theta, \quad [27]$$

where F_w = whipping force, lb.; A = area of shank cross-section, sq. in.; L = length of connecting-rod, center to center, in.; N = r.p.m., R = crank radius, in.; θ = crank angle, deg.; w = weight of material, lb. per cu. in. = 0.283; g = acceleration due to gravity = 32.2.

F_w is considered as a distributed load varying uniformly from zero at the piston-pin to a maximum at the crank-pin end of the rod. Such a loading is assumed to produce a reaction of 1/3 total load at the piston-pin and 2/3 total load at the crank-pin. The reactions are small and are neglected in determining bearing loads. The bending moment produced by F_w in a section of the connecting-rod shaft at distance x from the piston pin is

$$M_w = (F_w x/3)\{1 - (x^2/L^2)\}. \quad [28]$$

Maximum bending moment, M'_w, occurs at the section where $x = 0.577L_1$ and is $M'_w = 0.1283\,LF_w$. The stress set up in the shank cross-section due to the whipping section is $S_b = M_w/Z$, where S_b = fiber stress due to whip, lb. per sq. in., Z = section modulus of shank area about axis parallel to piston-pin.

PISTON-PINS are made of a carburizing steel, usually from bar stock; seamless steel tubing sometimes is used. S.A.E. steels, 1015, 1020, 2520 and 6115 are used. The case depth of carburization is 0.030 in. to 0.045 in. Pins are ground and lapped to a fine satin finish. Hardness is about Rockwell C–60.

Oil is supplied to the pin either by splash from the connecting-rod bearings or by pressure through gun drilling in the connecting rod. Clearance in the bearing surface is a light push fit. That in the locking part of the pin is at least 100-lb. forced fit. Pins usually are locked either in the piston or in the upper end of the connecting-rod; sometimes the pin floats both in piston and rod. Bearing pressures due to explosion pressure vary between 2000 and 3500 lb. per sq. in. of projected area. Bearing pressure = F/DB lb. per sq. in., where F = force on piston, lb., D = outer diameter of the pin, in., B = length of bearing in connecting-rod, in.

To determine stresses, consider the pin as a beam supported at the center of the piston-pin bosses, and loaded uniformly at the connecting-rod upper bearing. The resulting maximum stresses are:

$$S_t = (32/\pi)D/(D^4 - d^4) \times F\,(2L - B)/8, \quad [29]$$

$$S_s = F/2A = F/\{\pi/2)(D^2 - d^2)\}, \quad [30]$$

where S_t = tensile stress, lb. per sq. in.; S_s = shearing stress, lb. per sq. in.; d = inner diameter of the pin, in; L = distance between centers of piston bosses, in.; other notation as above.

4. AUTOMOBILE FUELS AND COMBUSTION

HEAT CONTENT.—Gasoline, either cracked or straight run, is the ideal fuel for automobile engines. Its heat content is high. See Table 8. Columns 4 and 5 show the total, gross or high heating value, *i.e.*, B.t.u. liberated by a unit amount of gasoline burned with oxygen in a constant volume enclosure, the products of combustion CO_2, SO_2 and H_2O being cooled to the initial temperature and H_2O condensed to a liquid. Columns 6 and 7 show the net or low heating value, *i.e.*, B.t.u. liberated at constant pressure. This value is the most significant in calculating engine efficiencies, as, in most practical applications, combustion occurs at constant pressure and the H_2O formed is not condensed. The average specific gravity at 60° F. of commercial gasoline is 0.74.

VAPOR LOCK.—The 10% point of the A.S.T.M. distillation curve indicates the approximate temperature of the gasoline at which vapor lock troubles in the fuel line may occur. Summer gasolines range from 125° to 150° F. and winter gasolines from 105° to 140° F. A more accurate indication of the tendency to vapor lock is the Reid vapor pressure.

Table 9 gives measured temperatures in the liquid gasoline flowing through the fuel system in a typical automobile.

STARTING CHARACTERISTICS.—For easy starting, especially in cold weather, volatility is the most important fuel property. In winter, all of the gasoline will not evaporate in the manifold; an excess must be supplied to enable the portion which does evaporate to form an explosive mixture. The excess is supplied by the choke, which changes the mixture ratio from the normal of about 13 : 1 to as low as 0.3 : 1. Fig. 16 shows the air-fuel ratios required to produce an explosive mixture. The 10% points also are shown on the curve.

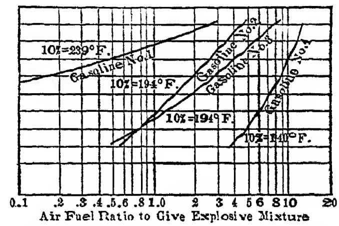


FIG. 16. Variation, with Temperature, of Air-fuel Ratios for Explosive Mixtures

Table 8.—Heating Value and Properties of Gasoline

(U. S. Bureau of Standards Miscellaneous Pub. No. 97)

Gravity		Density, lb. per gal.	Heat of Combustion at Constant Volume, Q_v		Heat of Combustion at Constant Pressure, Q_p	
Deg. A.P.I. at 60° F.	Specific at 60°/60° F.		B.t.u. per lb.	B.t.u. per gal.	B.t.u. per lb.	B.t.u. per gal.
55	0.7587	6.326	20,140	127,400	18,810	119,000
56	.7547	6.292	20,170	126,900	18,830	118,500
57	.7507	6.258	20,190	126,400	18,850	118,000
58	.7467	6.225	20,210	125,800	18,870	117,500
59	.7428	6.193	20,230	125,300	18,880	116,900
60	.7389	6.160	20,260	124,800	18,900	116,400
61	.7351	6.128	20,280	124,300	18,920	115,900
62	.7313	6.097	20,300	123,700	18,930	115,400
63	.7275	6.065	20,320	123,200	18,950	114,900
64	.7238	6.034	20,340	122,700	18,960	114,400
65	.7201	6.004	20,360	122,200	18,980	113,900
66	.7165	5.973	20,380	121,700	18,990	113,400
67	.7128	5.943	20,400	121,200	19,010	112,900
68	.7093	5.913	20,420	120,700	19,020	112,500
69	.7057	5.884	20,440	120,200	19,040	112,000

Table 9.—Gasoline Temperatures in Automobile Fuel Systems

(From paper by W. G. Lovell, Local Meeting of S.A.E., Pittsburgh, March, 1930)

	At 60 m.p.h.	Idling, after slowi from 60 m.p.h.
Atmospheric temperature.	79 deg. F.	79 deg. F.
Gasoline, rear tank......	100	111
Gasoline line from tank..	113	135
Entrance to fuel pump...	122	140
Exit to fuel pump.......	129	158
Carburetor float chamber.	136	160
Water in radiator........	189	207

ıder 5 mg. per 100 cc.

OCTANE NUMBER.—The essentials to a uniform method ... engine accessories; 2. Com-

effectively

and characteristics of the

. is affected by the engine and conditions

Bore, in....		Piston-pin, diam., in.	$1\,^1/_4$
Stroke, in..	4	Connecting-rod length, in.	10
Cylinder displacement, cu. in.	37.4	Number of piston rings	5
Intake-valve diam., in.	$1\,^3/_{16}$	Timing-gear face-width, in.	1
Exhaust-valve diam., i		Carburetor flange	S.A.E.
Connecting-rod bearing diam., in.	$2\,^1/_4$	Carburetor flange diam., in.	1
Connecting-rod bearing length, in.	$1\,^5/_8$	Exhaust manifold, pipe-tap size, in.	$1\,^1/_4$
Front main-bearing diam., in.	$2\,^1/_4$	Spark-plug size, mm.	18
Front main-bearing length, in.	2	Indicator-opening diam., in.	$^7/_8$
Rear main-bearing diam., in.	$2\,^1/_4$	Indicator-opening threads per in.	18
Rear main-bearing length, in.	$4\,^1/_4$	Approximate weight, lb.	450

Bouncing Pin Indicator (Fig. 18) indicates the intensity of knock. It comprises a steel rod which rests on a diaphragm exposed to pressures in the combustion chamber. Knock causes the pin to bounce up, closing the contact points. These points, and either a gas-evolution burette or a knockmeter, are connected in series in an electric circuit. The burette. Fig. 19, is filled with 10% H_2SO_4 solution, and the gas generated by electrolysis of this solution is measured. The knockmeter is a device to integrate current, resulting from knock, flowing in the bouncing pin circuit.

Reference Fuels and Octane Number Scale.—The common reference scale, expressed in terms of octane number, has been adopted by both

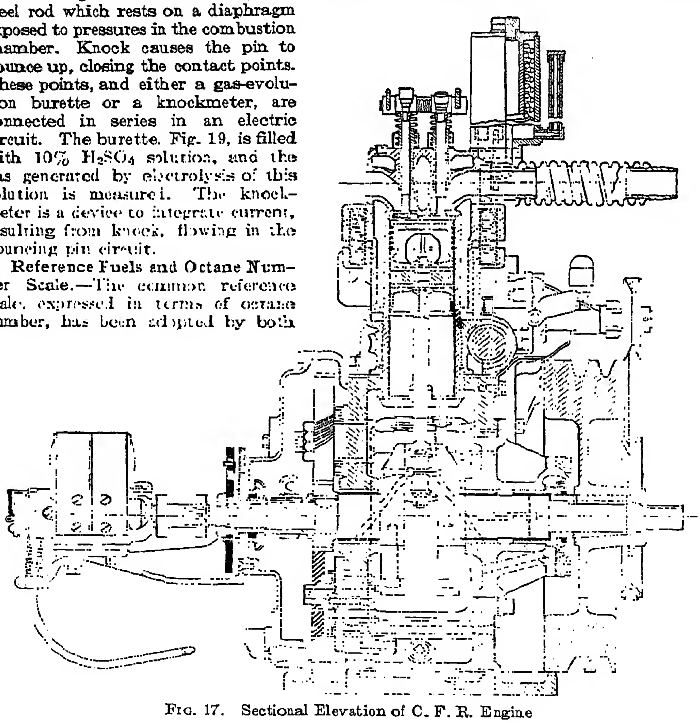


FIG. 17. Sectional Elevation of C. F. R. Engine

the oil and automobile industries. Two pure hydrocarbons, both readily available and with a constant antiknock value, are the basis of the octane number scale. One, normal heptane, has very low antiknock value. Its source is the sap of the Jeffrey pine, grown on the Pacific coast. The second, iso-octane (2,2,4-trimethyl pentane) of high antiknock quality, is made synthetically from tertiary butyl-alcohol.

The octane number of any gasoline is the percentage by volume of octane in the octane-heptane mixture that just matches the gasoline in antiknock quality, as determined in the C.F.R. engine by the standard procedure. Thus, if a gasoline is matched by a mixture of 60 parts octane and 40 parts heptane, its octane number is 60. The octane number of gasolines ranges from 40 to 87. Premium fuels are 78; average grade, from 60 to 70; fighting grade aviation gasoline, 87.

The test data, Fig. 20, show the increase in power, brake M.E.P., octane number requirements, and the decrease in specific fuel consumption (lb. per B.Hp. per hr.) when compression ratio is increased. These curves indicate trends only and show performance in a test engine.

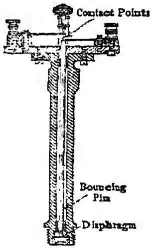


FIG. 18. Bouncing Pin Element

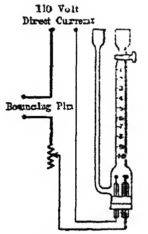


FIG. 19. Gas Evolution Burette

Reference.—Co-operative Fuel Research Apparatus and Method for Knock Testing, T. A. Boyd; presented at Twelfth Annual Meeting, American Petroleum Institute, Nov. 11, 1931.

5. FUEL SYSTEM

CARBURETORS.—Carburetors act as metering devices and should: *a.* Properly proportion the air-fuel ratio at different loads; *b.* Correct for temperature changes; *c.* Provide suitable accelerating mixtures; *d.* Have low air resistance, to maintain high volumetric efficiency. Commercial carburetors embodying these principles may be classified according to metering principle as plain tube and air valve. The former uses a Venturi tube to obtain a nearly constant mixture ratio throughout the major portion of its range. The latter uses the depression of an air valve for the same purpose. Other devices to obtain the desired characteristics over the entire speed and load ranges are: Metering pins, idle jets, compensating jets, double Venturis, pressure bleeds, and numerous others. Many of these are common to both plain tube and air valve carburetors.

MIXTURE RATIOS at full throttle should be approximately 13 : 1 (lb. of air per lb. of fuel), and about 15 : 1 at part-throttle level-road operation to give greater economy. For all other loads, the ratio should be between these two. Part-throttle economizers and full-throttle enriching devices are added to carburetors to obtain this double range feature. For accelerating, a rich mixture is necessary and is provided by pressure on the float chamber or a fuel pump. For starting, especially in cold climates in winter, the carburetor must supply a very rich mixture. Only about 6% of the fuel can be evaporated to form a combustible mixture at 0° F. A choke to provide mixture ratios of 0.3 : 1 to 0.8 : 1 at 0° F. is necessary for starting at low cranking speeds.

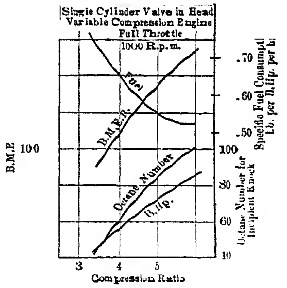


FIG. 20. Power, Fuel Consumption and Octane Number Requirements at Various Compression Ratios

MANIFOLDING.—An ideal manifold should supply an equal charge of identical fuel-air ratio established by the carburetor to all cylinders at all speeds, loads and rates of acceleration. It is desirable that the manifold have equal demand intervals on a branch, established either in the firing order or in the manifold design.

Distribution of mixture to the cylinders is complicated by less than half the liquid being vaporized by the carburetor. To obtain better vaporization heat is supplied in the manifold. Heat reduces volumetric efficiency and should be used as little as possible. Hot spots surrounding the throttle are effective, especially at part-throttle. To vaporize liquid in suspension at full throttle, it must strike a hot spot at 90° to the path of travel.

Liquid distribution concerns not only the manifold but also the carburetor, intake ports and valves. Turbulent or spiraling air flow tends to upset liquid distribution.

Violent spiraling tends, by centrifugal force, to throw all liquid drops to the walls of the manifolds. If spiraling is mild, distribution may be bettered by air straighteners. Manifolds of simple design are, in general, the best. In multi-cylinder engines, especially above 6 cylinders, multiple carburetors are necessary. Pronounced bumps or cavities at the tee, against which the mixture impinges, tend to increase turbulence and are undesirable. A straight section of at least (2 × manifold diam.) between elbow and siamese port is desirable to permit the mixture to resume normal flow.

FUEL FEED.—A constant supply of gasoline must flow to the carburetor from the tank during operation of the engine. Fig. 21 shows a typical positive engine-driven pump used for this purpose. Diaphragm *a* draws fuel from the tank and feeds it to the carburetor under pressure of the spring *c*. Diaphragm *a* is pulled down positively by lever *f* to draw fuel, but link *e* allows it to remain down if the carburetor float-valve is closed.

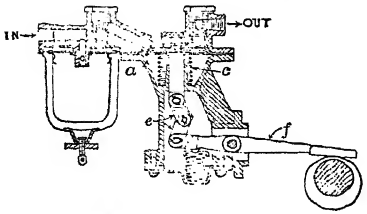


Fig. 21. Fuel Pump

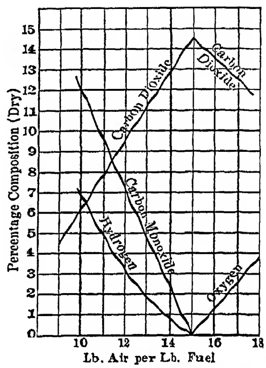


Fig. 22. Relation of Exhaust Gas Composition and Mixture Ratio

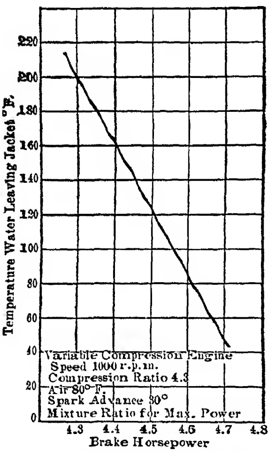


Fig. 23. Relation of Power and Jacket Water Temperature

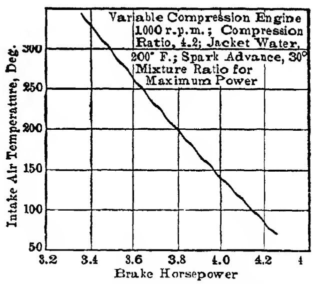


Fig. 24. Relation of Power and Carburetor Air Temperature

EXHAUST GAS COMPOSITION.—Fig. 22 shows the approximate relationship between exhaust gas composition and mixture ratio in the engine. This curve may be used to determine the mixture ratio of an entire engine or a single cylinder by measuring the CO content of the gases. Distribution in a multi-cylinder engine can be studied by making an analysis of the gases from each cylinder.

EFFECT OF ENGINE VARIABLES ON POWER OUTPUT.—Figs. 23–26 respectively show the effect of jacket water temperature, carburetor air temperature, spark advance and mixture ratio on power output of a single-cylinder variable-compression test engine. Fig. 20 shows the effect of changes in compression ratio, brake M.E.P., fuel consumption and octane number requirements for the fuel under stated conditions. Increase in compression ratio, jacket water temperature, carburetor air temperature

and spark advance increases the tendency of the engine to knock. Carburetor mixture ratio also affects this tendency, which decreases with a mixture leaner or richer than about 13 : 1, which ratio varies with different fuels. Low jacket water temperature has greater effect than low intake air temperature in reducing knock.

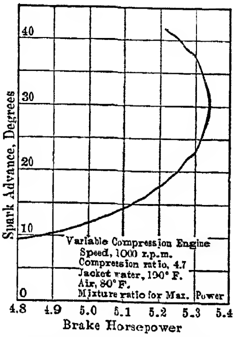


FIG. 25. Relation of Power and Spark Advance

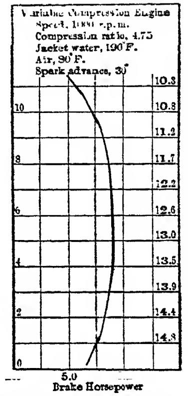


FIG. 26. Relation of Power and Mixture Ratio

The curves show relative effect of the variables and cannot be applied directly to all engines. They are useful in showing trends. In Fig. 24, the loss of power is directly due to the loss in volumetric efficiency with increased air temperature. Fig. 25 shows the characteristic of an optimum spark advance for given engine speed, compression ratio, mixture ratio and temperature conditions. Fig. 26 shows the characteristic of an optimum carburetor setting for specific engine conditions for maximum power.

6. ELECTRICAL SYSTEM

Electrical systems in American passenger cars operate at 6 volts; in European cars at 12 volts. The electrical circuit diagram, Fig. 27 is typical of 1935 practice on a medium-price 6-cylinder engine. The ignition coil converts the low voltage from the storage battery to a high voltage which will jump the spark plug gap. Fig. 28 shows the effect

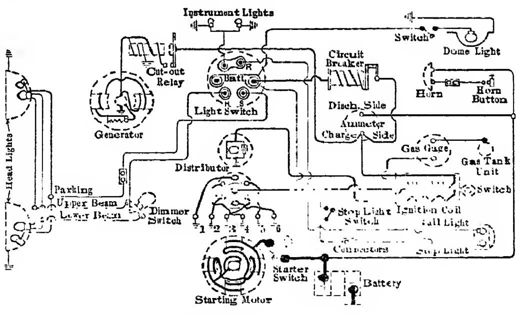


FIG. 27. Electrical Circuit Diagram

of a primary voltage on secondary voltage in a typical 6-volt coil. The spark plug gap should be from 0.015 to 0.018 in. for high-compression engines and from 0.018 to 0.022 in. for low-compression engines.

In a 6-volt generator, the field generally is shunt wound with 3d brush regulation of current. A cut-out, mounted in the generator allows current to flow to the battery when the generator speed has reached a point where generator voltage is greater than battery voltage. Fig. 29 shows current-speed characteristics for a typical shunt-wound generator with 3d brush regulation.

Engagement of the starter pinion with the fly-wheel gear is made by a manual shift lever. Two other systems are used, a semi-automatic and an automatic engaging mechanism. The motor is series-wound to give *high starting torque*. The starter will crank a warm engine at about 165 r.p.m. and a cold engine (0° F.) at about 35 r.p.m.

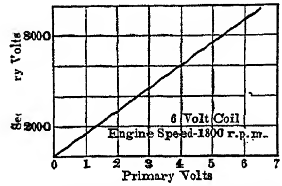


FIG. 28. Effect of Primary Voltage on Secondary Voltage

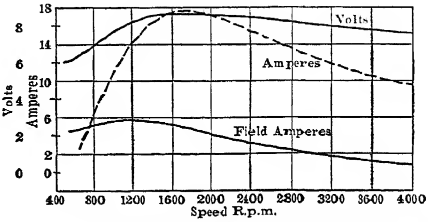


FIG. 29. Third Brush Generator Charging a Half-charged 13-plate Battery

7. CHASSIS

FRAMES AND SPRINGS.—The body, fenders, radiator, engine, transmission and wheels are attached to the motor vehicle frame. Strength and rigidity are the chief requirements of the frame. Usually, the two side members are channel section, with tubular or channel cross members riveted to them. To obtain greater torsional rigidity, X cross members sometimes are used. Fig. 30 shows a so-called double drop frame which uses X cross bracing. The dimensions of frame section range as follows: Depth, 5 to 9 in.; width, 1 $^3/_4$ to 3 in.; thick- to $^3/_{16}$ in. The smaller values refer to light cars and the larger to the heaviest.

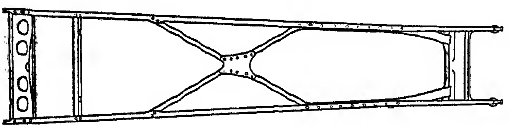


FIG. 30. Double Drop Frame with X Cross Bracing

Methods of Drive.—The three main methods of drive are: Hotchkiss (Fig. 31), torque tube (Fig. 32), and independent spring suspension (Fig. 33). In the Hotchkiss, driving and braking torque is taken through the front and rear springs. In the torque tube, the drive is taken through a hollow tube enclosing the propeller shaft and firmly connected at either end to the rear axle and transmission. In independent rear spring suspension, the rear axle housing is mounted on the frame. Universal joints, between the wheels and drive gears, are necessary to allow for the motion between the frame and wheels. The vertical motion of each rear wheel is therefore independent of the other. Front wheels may be similarly mounted so their motion is independent.

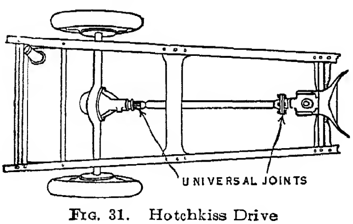


FIG. 31. Hotchkiss Drive

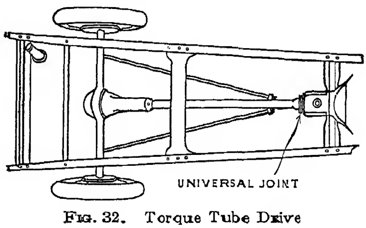


FIG. 32. Torque Tube Drive

Deflection of Semi-elliptic Springs may be found from the beam formula $D = WL^3/48EI$, where D = deflection, in.; W = weight, lb.; L = length, in.; E = modulus of elasticity; I = moment of inertia. The initial deflection so found determines the period of vibration. An approximate formula for the period of semi-elliptic springs is $V = \sqrt{35{,}300/D}$, where D = initial deflection, in.; V = period, vibrations per min. For rear springs, $V < 90$ gives good riding characteristics. In semi-elliptic front springs V should be more than double the rear to prevent resonance between the front and rear springs causing a pitching motion. In individual front wheel suspension, the front and rear springs can be about the same, which eliminates pitching. To control the springs, shock absorbers are used, some of which may be adjusted or are self-adjusting for different types of roads.

Factors other than springs contributing to good riding quality are: 1. Low unsprung weight; 2. Low center of gravity; 3. Long wheel-base; 4. Soft springs; 5. Good shock absorption; 6. Rebound control; 7. Large section tire; 8. Proper weight distribution; 9. Adjustable seats; 10. Proper seat cushions and springs.

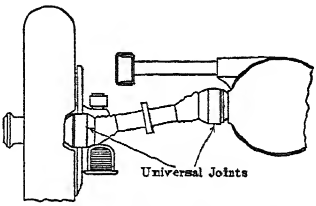
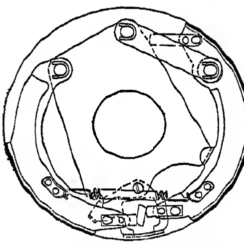


FIG. 33. Independent Rear Spring Suspension Drive

BRAKES must be able to stop the car quickly from any speed at which it may be driven, with good control and without undue change in the rate of deceleration. They must be able to withstand continuous operation to keep the speed low on long grades and to hold a parked car stationary. Legal requirements call for two brakes, a hand-lever-



G. 34. Two-shoe Brake

FIG. 35. Three-shoe Brake

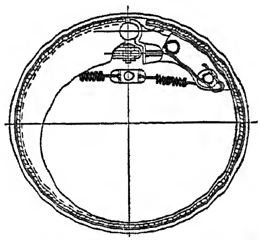


FIG. 36. Internal Band Brake

operated parking and emergency brake, and a foot-pedal service brake. Brakes on all four wheels always are used. As a rule, automobile brakes are (1935) of the internal type. Details of design vary greatly. Fig. 34 shows a two-shoe, and Fig. 35 a three-shoe type. Fig. 36 is an internal band type. Fig. 37 is an internal hydraulic brake.

The brakes must dissipate the kinetic energy of the moving car by converting the energy into heat, which must be radiated by the drum surface. The energy at maximum speed ranges from 600,000 ft.-lb. in a small car to 2,000,000 ft.-lb. in a large one. A stop from maximum speed generates from 0.70 to 0.90 B.t.u. per sec. per sq. in. of drum surface. Fig. 38 shows the stopping distances from various speeds at various rates of deceleration. Fig. 39 shows desirable limits of pedal pressure and deceleration. Requirements for a good brake are: 1. Minimum variation under conditions of operation; 2. Low pedal pressure; 3. Pedal pressure proportional to retardation; 4. Smooth and quiet action; 5. Permanent equalization; 6. Long life and infrequent adjustment; 7. Accessibility and simplicity of adjustment.

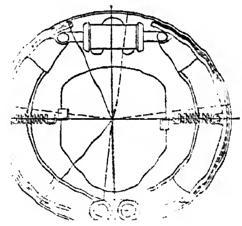


FIG. 37. Internal Hydraulic Brake

Brake Linings have much to do with meeting the above requirements. Three general types are used: Woven asbestos, impregnated with oxidizing oils; folded and rubberized asbestos cloth; molded homogenous mixture of asbestos fiber with other materials to

act as a binder and govern the coefficient of friction. The characteristics of these types are: *Woven Linings.* Wear well; easy to install; coefficient of friction gradually increases as lining wears. *Folded Linings.* Generally increase and decrease in friction value as wear progresses through the layers of cloth and compound. *Molded Linings* (flexible) usually have a higher friction coefficient than hard, dry linings and vary more with temperature changes. Hard, dry linings use a phenolic or bakelite type of binder instead of volatile oils, rubber or gums; wearing characteristics are excellent. Fig. 40 shows variation in coefficient of friction with speed, load and temperature of several lining materials.

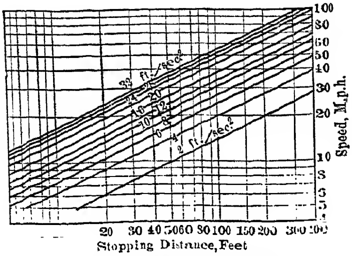


Fig. 38. Stopping Distances at Various Deceleration Rates

Drum material and design materially affect brake operation. With severe service, scoring is a problem. Cast iron, or steel with a centrifugally-cast inner layer of cast iron, is much used to resist scoring. Deep drawing steel (S.A.E. 1010 or 1025) also is used.

The division of applied effort between the front and the rear wheels depends on wheelbase, height of center of gravity, distribution of car weight on front and rear wheels and coefficient of friction between tires and road. The weight transfer, due to deceleration, also should be considered for which Fig. 41 may be used.

Example.—Assume static weight distribution to be: Front wheels, 40%; rear wheels, 60%; r = (height of center of gravity) ÷ wheel-base = 0.20; deceleration = (25 ft. per sec.)². *Solution.*—From the intersection E of the 40% line and line r = 0.20 for front wheels, lay off on the 40% line a distance EF = (25 ft. per sec.)², according to scale shown. At F erect a perpendicular to EF, intersecting r = 0.20 (front wheels) at G. Project G horizontally and read the new percentage of weight on front wheels as 55.5%. Percentage of weight on rear wheels = 100 − 55.5 = 44.5%. If the coefficient of friction between tires and road is such that (25 ft. per sec.)² is the maximum deceleration possible, the distribution of brake effort should be 55.5% front and 44.5% rear in order to obtain the maximum deceleration.

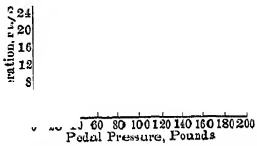


Fig. 39. Relation of Deceleration and Pedal Pressure

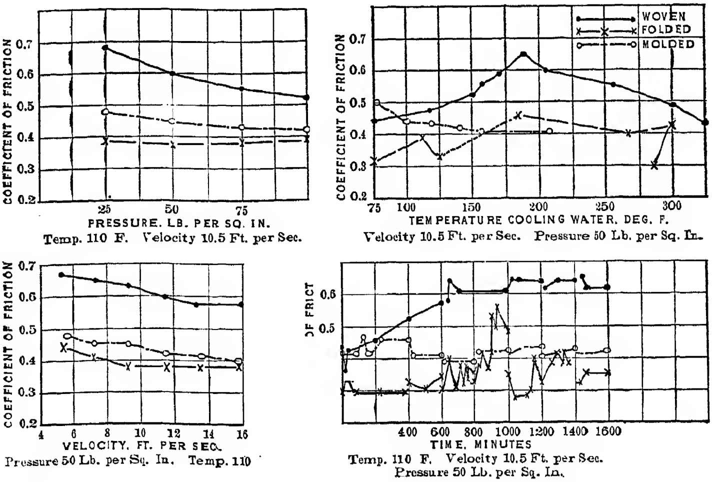


Fig. 40. Relation of Coefficient of Friction, Speed, Load and Temperature of Various Brake Linings

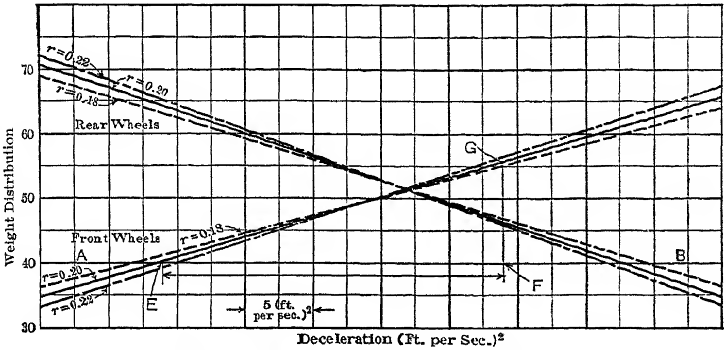


FIG. 41. Weight Transfer Due to Deceleration

The Internal Brake depends on self-actuation for part of the low pedal pressure necessary. The self-actuating brake produces a retarding force greater than the product of the normal pressure due to the applied force and the friction coefficient of the lining, due to a wedging action between shoe and drum. Variations in other parts of the brake limit self-actuation to about 50%.

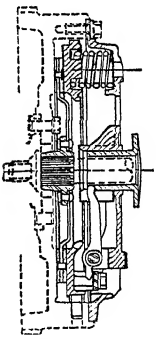


FIG. 42. Single Disc Clutch

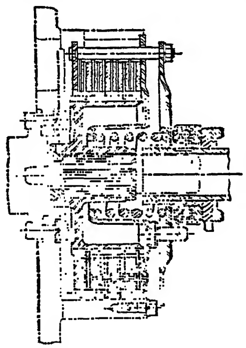


FIG. 43. Multiple Disc Clutch

Pedal Pressure is transmitted to brake shoes by either a mechanical or a hydraulic system. Both should have high efficiency to obtain low pedal pressure. Retardation should be proportional to pedal pressure, with an initial pressure of not over 25 lb. and preferably 10 to 15 lb. In the larger and heavier passenger cars, trucks and buses, a servo or auxiliary mechanism is sometimes desirable to obtain low pedal pressures. Compressed air, manifold vacuum or a mechanical device, driven from the engine or drive shaft, variously are used. The servo-brake should give proportionality of effort equivalent to a mechanical system, and should positively and accurately control the auxiliary power.

CLUTCHES.—Many types, including conical, internal expanding, band, single and multiple disc clutches have been used on automobiles; the last two almost universally are used (1935). For small and medium-size cars the single dry-disc clutch gives a simple design with few parts. Fig. 42 shows a typical design. For heavier cars, multiple disc clutches, Fig. 43, sometimes are used.

The torque-carrying capacity of a clutch is approxi-

where T = torque, in.-lb.; f = coefficient of friction; p = unit pressure between plates, lb. per sq. in.; A = area of contact, sq. in.; r_1 = outside diam. of disc, in.; r_2 = inside diam. of disc, in.

Fig. 44 shows the relation between torque carrying capacity and clutch pressure plate displacement of two different clutches.

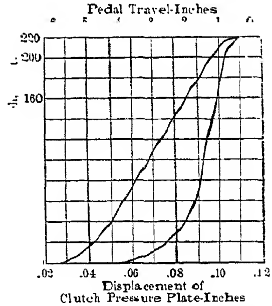


FIG. 44. Relation of Torque Carrying Capacity and Pressure Plate Displacement

PROPELLER SHAFT or drive shaft transmits engine torque from the transmission to the rear wheels. It usually is of plain carbon steel. In the torque tube drive, it is enclosed in the torque tube, and in the Hotchkiss drive it is open. Propeller shafts are made both solid and hollow. The exposed shaft in the Hotchkiss drive is usually a hollow drawn tube.

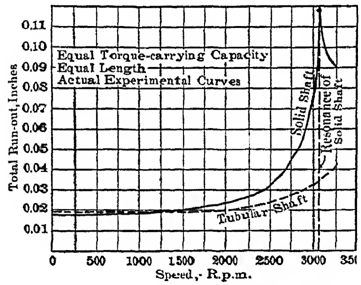
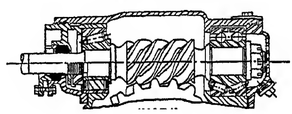


Fig. 45. Run-out of Tubular and Solid Propeller Shafts

For calculating stresses in the shaft the following formula is used: $M_t = S_r I_p / r$, where M_t = torque, in.-lb.; S_r = shear stress, lb. per sq. in.; I_p = polar moment of inertia of shaft section; r = radius of shaft, in.

One of the greatest difficulties in propeller shafts is the tendency to whip and cause noise at high engine speeds. Whip is due to shaft unbalance, lack of rigidity and natural frequency of vibration. Hence, high rigidity in the shaft and good initial balance are necessary, together with a resonant speed above that of the engine operating speeds. The natural frequency of a shaft with hinged ends, *i.e.*, where universal joints are used, is $F = (\pi/2L^2)\sqrt{EIg/A\gamma}$, where F = natural frequency, cycles per sec.; L = length of shaft, in.; E = modulus of elasticity = 30×10^6 for steel; I = moment of inertia of the shaft section = $\pi d^4/64$ for solid round section; g = acceleration of gravity = 32.2; A = area of shaft section, sq. in.; γ = density of material, lb. per cu. in.

Since a tubular shaft is more rigid and has a higher natural frequency of vibration, the amplitude of vibration will be smaller and come at a higher engine speed. Fig. 45 shows the amplitude of vibration at the center of a shaft or the "run out" for a solid and tubular shaft used on the same car.



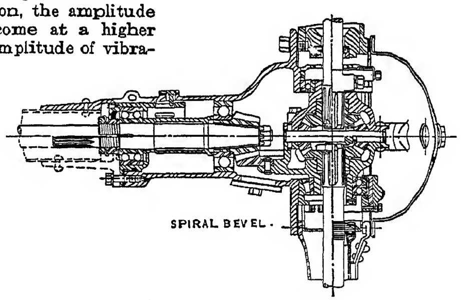


Fig. 46. Rear Axles

REAR AXLES.—The final drive is through a gear reduction which varies with the type and use of the vehicle. The most common type used (1935) is the spiral bevel gear. To lower the center of gravity, the worm and hypoid gear is used sometimes. Worm gears are used on trucks and buses, which also use double reduction units with internal gears or combined spiral and spur gears. Fig. 46 shows worm and spiral bevel types. Both hypoid and worm gears have much more sliding friction than spiral bevel gears, run hotter and require better lubrication. Fig. 46 also shows the application of roller, tapered roller, or ball bearings in the rear axle.

Table 10 gives comparative test results on three types of rear axles installed on the same chassis. A speed of 35 m.p.h. at road torque was used in all tests, with mineral oil lubricant.

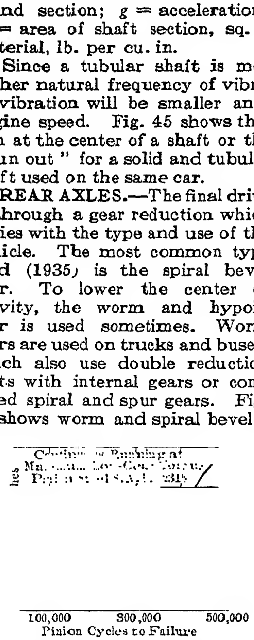


Fig. 47. Effect of Case Depth on Fatigue Resistance of Spiral Bevel Gears

Passenger cars mostly use spiral bevel gears. Pinion material is somewhat better than the gear material to withstand the greater stress. Heat treatment of gears materially affects fatigue strength. Fig. 47 shows results of tests on rear axle gears run under maximum low-gear torque. With an increase in case depth, resistance to fatigue failure increases, becoming less with the greater depths. This indicates that gear tooth durability depends

on a relatively thin surface layer. In gear design, fatigue strength, rather than static strength, should be considered. Fatigue strength of spiral bevel gears is affected by: 1. Bending stress (design elements of gear teeth, as circular pitch, tooth thickness, pressure angle and spiral angle); 2. Stiffness of carrier, bearings and gears; 3. Material and heat treatment; 4. Adjustment (position of total tooth contact area under load); 5. Machine accuracy and finish. The resistance to scoring and pitting is affected by these factors, and also by compressive stress, surface hardness, sliding velocities of teeth in contact, and physical and chemical properties of the lubricant. (See Gearing, Vol. 3 of this series.)

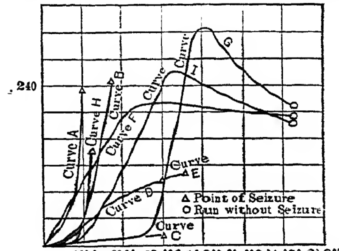


Fig. 48. Lubricant Tests. Mineral Oils: Curve *A*, viscosity, 48 sec. at 210° F.; curve *B*, 90 sec. at 210° F.; curve *C*, 200 sec. at 210° F. Mineral oils containing sulphur: Curve *F*, 40 sec. at 210° F.; curve *G*, 300 sec. at 210° F. Lard Oil: Curve D, 60 sec. at 210° F. Castor Oil: Curve *E*, 99 sec. at 210° F. Fluid Grease containing mineral oil and soap: Curve *H*, 500 sec. at 210° F., and 2500 sec. at 100° F. under 20 lb. per sq. in. Fluid Grease containing mineral oil, soap and sulphur: Curve *I*, 90 sec. at 210° F. and 1500 sec. at 100° F. under 20 lb. per sq. in.

Petroleum oils, free from excessive water, sediment, acid or other substances detrimental to proper performance, are used for rear axle lubrication. Fillers, as talc, pulp, cork, fuller's earth, graphite, mica or asbestos should not be used. The grade of oil is designated by S.A.E. viscosity numbers from 80 to 250. S.A.E. 80 and 90 are for extremely low temperatures. S.A.E. 250 is extra heavy for extremely high temperatures. S.A.E. 110 and 160 are for more normal temperatures.

In some rear axles what are known as E.P. (extreme pressure) lubricants are necessary. When loads are such as to cause a failure of fluid film lubrication, seizure and galling occur. A small percentage of some other material, as sulphur, will prevent seizure and permit carrying of loads two or three times heavier than can the mineral oil alone. Fig. 48 from Symposium on Developments in Automotive Materials, Regional Meeting A.S.T.M., Detroit, March, 1930, gives results of tests of a number of lubricants of varying viscosity and chemical composition.

TRANSMISSION.—Fig. 49 shows a typical 3-speed spur-gear transmission. Table 11 gives efficiencies of such a transmission. To make gear shifting easier for the driver, synchro-mesh transmission is used in the U. S. and England. In synchro-mesh transmission either two mating gears or dog clutches are brought up to an equal speed by a small cone clutch before they are engaged. A small differential in speed of the two mating members just before engagement insures that gear or clutch teeth will not meet end to end. In the constant-mesh type, helical gears may be used to give quiet operation. Synchronization between second and third, shifting up or down, is most used but synchronization on all speeds has been used.

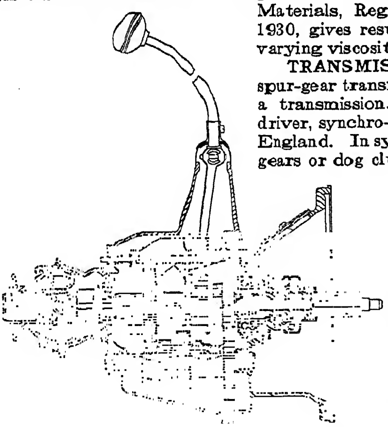


Fig. 49. Transmission and Hand Control

For ease of operation in shifting gears, a vacuum-operated clutch, Fig. 50, also is used. An interconnection between clutch and throttle engages or disengages the clutch. In the system shown, with the con-

Table 10.—Comparative Test Results of Rear Axle Gears

Type of Gear.	piral Bevel	Hypoid	Worm Gear 1	Worm Gear 2
Maximum friction Hp.	12.9	14.3	14.1	12.1
Minimum friction Hp.	10.2	11.5	12.4	11.8
Maximum temperature, deg. F.	165	181	366	334
Temperature range, deg. F.	136–165	131–181	194–366	167–334
Temperature change, deg. F.	29	50	172	167
Average rate of temp. rise, deg. per min.	0.36	0.50	2.64	1 50

Table 11.—Efficiency of a 3-speed Spur-gear Transmission

Transmission Gear	Speed, m.p.h.	Load	Efficiency, percent
Direct	10	Full	98.8
"	20	"	99.3
"	10	Part	97.2
"	20	"	98.7
Second speed	10	Full	95.7
" "	20	"	94.8
" "	10	Part	93.7
" "	20	"	93.3
First speed	10	Full	90.3
" "	10	Part	91.7

trol button pressed down and the foot removed from the throttle, the clutch is disengaged, and gears can be shifted without depressing the clutch pedal, or the car can coast with the rear wheels disconnected from the engine. A free-wheeling device or over-running clutch also is used to obtain coaster action and easy shifting. Fig. 51 shows a roller type mechanism. When the engine is throttled and the car tends to go faster than the engine, the free-wheeling device disconnects them. A lock-out device prevents free-wheeling down hills and allows use of the engine as a brake.

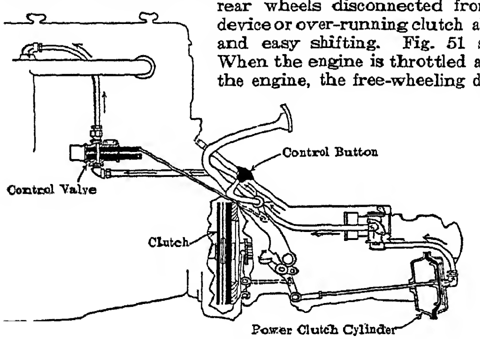


Fig. 50. Vacuum-operated Clutch

STEERING GEAR AND FRONT SUSPENSION.—Automobiles are steered by the divided axle or Ackerman system. The front wheels, suspended from the ends of the front axle, pivot around the king pins. The front axle must sustain the weight of the car and the braking torque. It usually is of I-section in the center, to carry the load, tapering to a circular section outside of the spring hangers to take the braking torque. Fig. 52 shows a typical steering system.

Independent Front Wheel Suspension improves riding quality and helps eliminate front wheel shimmy, wheel fight and tramp. Fig. 53 shows diagrams of various types.

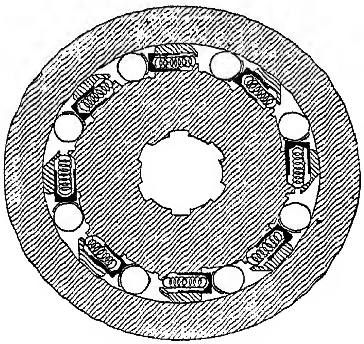


Fig. 51. Free Wheeling Unit

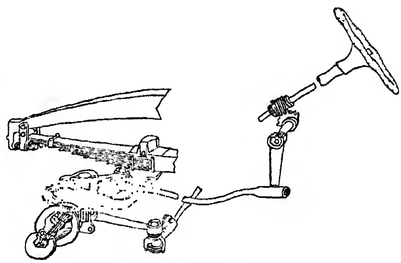


Fig. 52. Steering System

In the types used, a lower rate spring can be used in front than with the semi-elliptic type, resulting in better riding qualities. The spring parallelogram gives parallel wheel motion, but a change in tread. The spring and link type relieves springs of brake torque, but changes camber and track in proportion to the length of link. The link parallelogram type relieves springs of torque and allows proportioning length of links to obtain desired characteristics in camber and track changes. Swinging axles give large variations in

camber and track, but small lean of wheels on curves. Guided springs produce no changes in camber or track when vertical guides are used, but give hard steering.

The Geometry of the Front Wheels is chosen to give the best steering conditions on the steering system used. Wheels must be cambered, *i.e.*, the tops slightly further apart

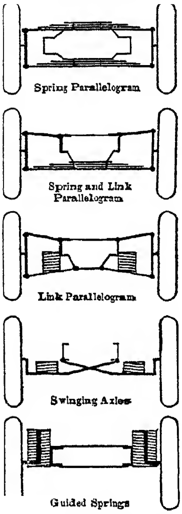


FIG. 53. Independent Front Wheel Suspensions

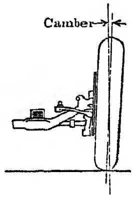


FIG. 54. Camber

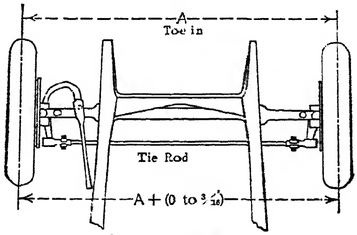


FIG. 55. Toe-in

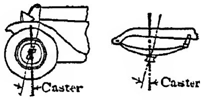


FIG. 56. Caster

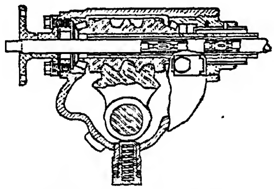


FIG. 57. Worm and Sector Steering Mechanism

than the bottoms (Fig. 54). The front of the wheels " toe-in," *i.e.*, are closer together than the rear (Fig. 55). The front axle and king pin is tilted backward (caster) so the bottom is further ahead than the top (Fig. 56). Toe-in varies from 0 to $^3/_{16}$ in. measured at the outer diameter of the wheels. The tie rod is adjustable to vary this factor. Camber is used to bring the contact points of tires and road directly under the center of the king pins to give easy steering. It is designed into the car by making the angle between king pin and the front axle end other than 90°. In practice, camber is measured as the angle between the king pin axis and the vertical axis of the wheel. Camber varies between 0.2° and 1.5°.

Tilting the front axle gives a caster effect by bringing the point of tire contact with the road at a point back of a line drawn through the center of the king pin. Caster varies from 1° to 4°, although a negative caster has been used.

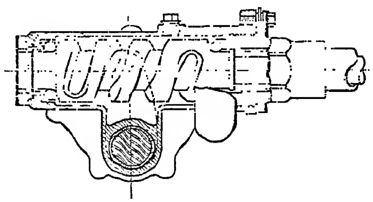


FIG. 58. Cam and Lever Steering Mechanism

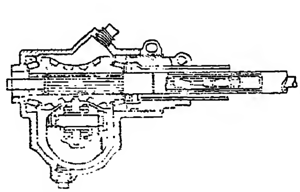


FIG. 59. Worm and Roller Steering Mechanism

Steering Mechanism.—The wheels are steered by a reduction mechanism. The most common types are: 1. Screw and nut; 2. Worm and sector (Fig. 57); 3. Cam and lever (Fig. 58); 4. Worm and roller (Fig. 59).

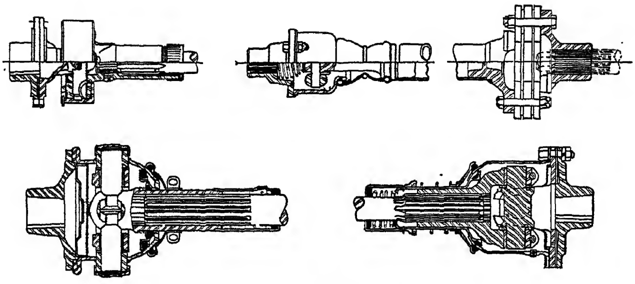


FIG. 60. Universal Joints

The ratio of the reduction unit ranges from 12 : 1 to 19 : 1; it is highest on large cars. From full left to full right, the arc of the steering wheel varies from 1000° to 1500° and the arc of the front wheels from 55° to 70°. The efficiency of the gear unit has much to do with the ease of steering. Factors affecting efficiency include type of unit, temperature, lubrication and use of anti-friction bearings. Average efficiencies for the various types of gear units are: Screw and nut, 20 to 40%; worm and sector, 40 to 50%; cam and lever, 40 to 65%; worm and roller, 60 to 70%. Under exceptionally good conditions, the efficiency of a worm and roller mechanism with anti-friction bearings may be as high as 85%.

UNIVERSAL JOINTS.—Where torque tube drive is used, only one mechanical universal joint is required. Where a Hotchkiss drive is used, a universal joint on both ends of the propeller shaft, either mechanical or fabric, is necessary Splines are used on the drive shaft end and a square section, key or spline on the transmission or rear axle end. Fig. 60 shows several types of universal joints.

8. ROAD TESTS

TYPES OF TESTS.—As yet, (1935) road tests have not been standardized by any society representing the automotive industry. Each manufacturer has developed his own procedure, and reported results of road tests are not comparative due to the wide variation in methods of conducting tests. Two distinct methods have been developed. One requires a specially built proving ground, where all conditions may be accurately controlled. The other depends on using public highways, with resulting interference from traffic. Highway tests sometimes are highly desirable, if test results demand conditions not easily duplicated at a proving ground. Two notable locations for special road work are: Uniontown Hill, in the mountains near Uniontown, Pa., often used when a long, almost even grade is required; Death Valley, Cal., 276 ft. below sea level, for road tests requiring extremely high temperatures, where for many successive days temperatures may be over 120° F., frequently reaching 130°.

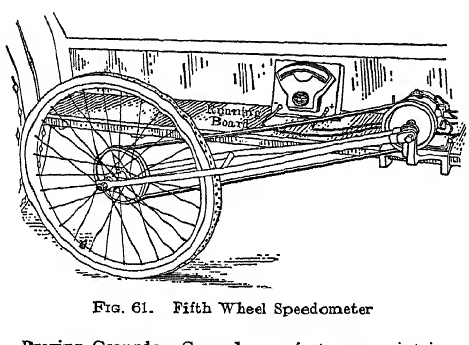


FIG. 61. Fifth Wheel Speedometer

Proving Grounds.—Several manufacturers maintain proving grounds where test conditions as well as procedure can be standardized exactly. To make results comparative, either the 2-door coach or 4-door 5-passenger sedan are used in all tests. These grounds are in a hilly district where various types of roads, including gravel, asphalt, concrete and dirt surfaces, may be built. Grades up to 30% are required. A straight stretch is necessary for acceleration, economy and speed tests. A speed loop is desirable where speeds in excess of 100 m.p.h. may be maintained. Proving grounds principally are used for comparative road tests of automobiles, and road tests of new models prior to production.

INSTRUMENTS AND EQUIPMENT FOR ROAD TESTS, in general, must be portable, not easily affected by rough usage, but equivalent in accuracy to laboratory instruments. Such conditions impose severe limitations on design and make instrumentation a subject of major importance.

The Fifth Wheel Speedometer (Fig. 61) gives accurate readings of speed in miles per hour. A bicycle wheel mounted in a frame clamped on the running board drives, through a web belt, a magneto whose open-circuit voltage is 6 at 1000 r.p.m., resistance, 100 ohms, and maximum speed 2000 r.p.m. The voltage is indicated on a voltmeter calibrated in m.p.h., one scale running from 0 to 50 m.p.h. and a second from 50 to 100 m.p.h. The connection between voltmeter and magneto is long enough to permit the meter to be held by the observer.

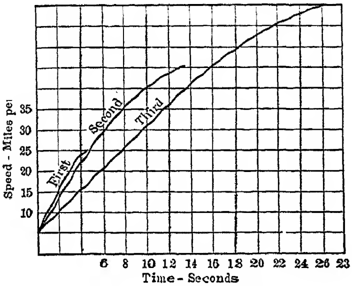


FIG. 62. Speed-Time Curve

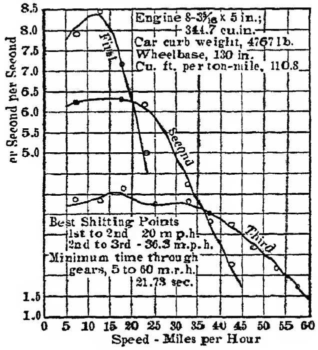


FIG. 63. Speed-Acceleration Curve

A variation of the above instrument incorporates a counter reading directly in feet traveled. A magnetic clutch operates the counter and a stop watch, to give simultaneous readings of speed in miles per hour, distance in feet and time in seconds. The maximum error in the distance covered was found to be 5 ft. in 1 mile; the average error was less than 2 ft. The data for the speed-acceleration curves, Figs. 62 and 63, in the three transmission ratios were obtained from the above instruments.

Fuel Economy Apparatus.—A good fuel economy measuring device consists of two 480 c.c. burettes, mounted side by side on a board and filled by an electric fuel pump mounted on the back of the board. A 2-way valve feeds gasoline from either burette to the carburetor directly, or to the fuel pump, depending on the fuel system used. This apparatus is used in connection with the distance recording device. Tests are run for one mile in opposite directions to average wind effect. Time (min. and sec.) and cu. cm. of fuel used are recorded. Readings start at 10 m.p.h., with 5 m.p.h. increments up to car speed. See Fig. 64.

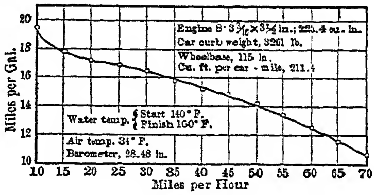


FIG. 64. Speed-Fuel-Consumption Curve

The Brake Decelerometer measures braking ability in terms of deceleration, ft. per sec. per sec., for a corresponding pedal pressure in pounds. Fig. 65 shows the essential parts; A is a 6-volt d.c. motor driving the recording paper; B is a fixed recording pen reading the zero position for pedal pressure pen C and deceleration pen D; E is the foot element for transmitting pedal pressure to spiral tube AA. Elements AA, F and E are filled with liquid. Pen D is so connected to an inverted pendulum that when the car slows down, and the pendulum moves forward, it moves across the paper in proportion to the deceleration rate. A device also is used to show the pedal travel in inches with deceleration.

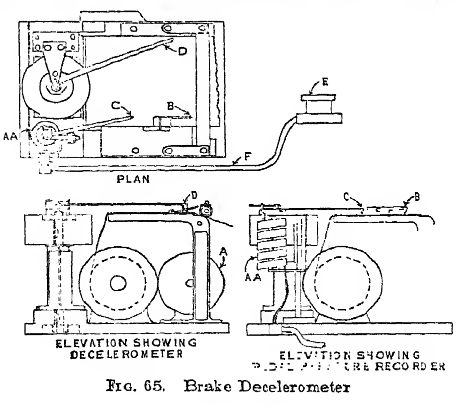


FIG. 65. Brake Decelerometer

Figs. 66 and 67 show results of tests under different conditions. Fig. 66 is the performance of a set of 4-wheel mechanical brakes when stopping from 50 m.p.h. with the brakes cold and hot. To heat the brakes prior to the test, they were applied with the throttle wide open, to reduce car speed to 20 m.p.h. Fig. 67 shows the result of stopping from 75 m.p.h. with a constant brake pressure.

Procedure for Obtaining Data on Road Tests

A driver and an observer are required for road tests. The tabulation (p. 14–83) shows the data and type of information most necessary for comparisons in road performance tests. These data usually are tabulated or plotted on curves. Instructions for each type of test follow.

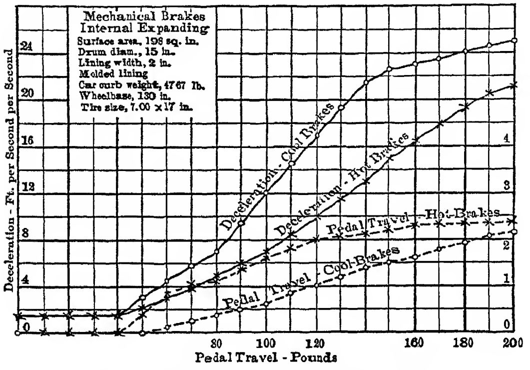


FIG. 66. Brake Curve at 50 Miles per Hour

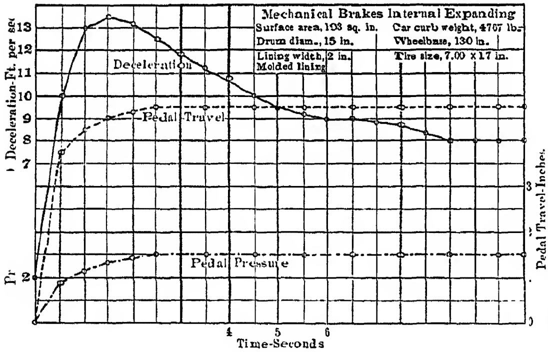


FIG. 67. Constant-pressure Brake Curve at 75 Miles per Hour

INSTRUCTIONS FOR OBTAINING GENERAL CAR DATA.—1. Obtain model number from instruction book or invoice, engine and serial number, tire size and make of car. 2. Bore and stroke will be obtained in garage. 3. Measure wheelbase from center to center of hub caps on each side. Obtain nominal wheel-base from instruction book. 4. Take curb weight with gas, oil and water to capacity; include tools and spare tire. This measurement is very important; care should be taken to get correct weight. Note weight of tools separately. 5. Oil capacity is quantity to fill to gage mark when oil has been drained through drain pet-cock. 6. Water capacity is capacity of block and radiator from level of drain-cock. 7. Obtain tire pressures from instruction book where possible, or compute from manufacturers' data. Computed pressures generally will be less than recommended pressures. 8. Turning diameter: Turn wheel hard over, and turn car around until it is traveling in a consistent circle; pour oil or water on tire tread to mark the ground, and measure diameter of resultant circle at center of tread. 9. Arc of steering wheel is angular travel from full right to full left, omitting lost motion at each end. 10. Number of feet for ten turns of the rear wheel—taken after inflating tires to correct pressure and loading car with 450 lb. weight. Take reading of rear tires on each side and record average. 11. Gear ratios: Obtain by jacking up *one* rear wheel and counting revolutions of distributor rotor while wheel is turned through 20 rev. Multiply readings by two and record. Count pinion and gear teeth and record. 12. Moment of inertia: Obtain by removing one front and one rear wheel, placing each separately on a standard disc of known moment of inertia and take at least two measurements of the time of 10 periods. Repeat for 10 periods for the standard disc alone. 13. Clearance data: Inflate tires to correct pressure. Put car on level surface and measure clearance of front and rear axle and fly-wheel housing. Measure distance from fly-wheel housing to center of front axle, from point on fly-wheel where clearance was measured. 14. Spring ride: Measure from bumper to bumper, *i.e.*, the maximum possible deflection is desired; measure length of springs from center to center of eyes. 15. Service and emergency brakes: Record whether they are 2-wheel, 4-wheel, transmission; whether hydraulic or mechanical; whether internal expanding or external contracting. Record

length and width of linings. If lining is in sections, record length of each section. Give drum diameter at point of contact. 16. Widest section of frame: Record measurements of channel section through the frame at its heaviest point.

DATA NECESSARY IN ROAD TESTS.—Information necessary for complete road tests is tabulated below.

DIMENSIONS

General Car Data
- Car manufacturer
- Body style
- Model No.
- Engine No.
- Serial No.
- Wheel-base—right and left, in.
- No. of cylinders
- Bore and stroke, in.
- Displacement, cu. in.
- Gas tank capacity, gal.
- Oil capacity, gal.
- Water capacity, gal.

Curb Weight, lb.
- Right front
- Left front
- Right rear
- Left rear
- Total

Tires
- Kind and size, in.
- Pressure, front and rear, lb. per sq. in.

- Rolling diameter, ft.
- Lb. per cu. in. displacement
- Lb. per in. of wheel-base

Gear Ratios
- First, second, third, fourth, reverse.
- Engine r.p.m. per m.p.h.
- Piston travel, ft. per car-mile
- Displacement, cu. ft. per ton-mile
- Displacement, cu. ft. per car-mile

Service Brakes
- Type
- Lining length, front and rear, in.
- Lining width, front and rear, in.
- Drum diam., front and rear, in.
- Braking surface, sq. in., front and rear

Emergency Brake
- Type
- Lining length, in.
- Lining width, in.
- Braking surface, sq. in.
- Minimum road clearance
- Front and rear axle, in.

Spring Ride
- Right front, in.
- Left front, in.
- Right rear, in.
- Left rear, in.

Front Spring
- Length and width, in.
- No. and thickness of leaves, in.

Rear Spring
- Length and width, in.
- No. and thickness of leaves, in.

Moment of Inertia of Wheels, Lb.-ft.^{2}

MAKE AND MODEL OF

Engine
Carburetor
Carburetor silencer
Air cleaner
Fuel feed system
Gas strainer
Oil purifying system
Crank-case ventilator
Radiator
Fan belt
Front end drive
Steering gear
Distributor and coil
Generator
Starting motor
Spark plugs
Horn

Speedometer
Oil gage
Ammeter
Windshield wiper
Rear vision mirror
Clock
Cigar lighter
Car lock
Gas gage
Tail light
Stop light
Bumpers
Shock absorbers
Headlights
Lenses
Lubrication system
Motor temperature control or indicator

Tires
Rims
Wheels
Brakes
Springs
Front axle
Rear axle
Frame
Universal
Transmission
Clutch
Muffler
Battery
Tire lock
Electric starter control
Free wheeling

CAR SPECIFICATIONS AND PERFORMANCE INFORMATION

Steering mechanism
- Arc of steering wheel, deg.
- Arc of left front wheel, deg.
- Arc of Pitman arm, deg.
- Steering gear ratio
- Tread, front and rear, in.
- Camber, deg.
- Caster, deg.
- Toe-in, in.
- Turning diam., ft., right and left

Volume of engine clearance space, cu. in. (each cylinder)
Cylinder displacement, cu. in.
Compression ratio (each cylinder)
Compression pressure, lb. per sq. in. (each cylinder)
Speedometer calibration
Odometer calibration
Clutch pedal pressure, lb.
Steering effort, lb., over standard curves
- 5 m.p.h., right and left
- 10 m.p.h., right and left
- 15 m.p.h., right and left

Frame, heaviest section
- Width, in.
- Depth, in.
- Thickness of stock, in.
- Section modulus

Center of Gravity
- Height above ground, in.
- Distance forward from center line of rear wheels, in.
- Lateral distance in from right rear wheel, in.

Minimum idling speed, m.p.h.
Minimum speed for flexibility, m.p.h.
Maximum speed, two directions, m.p.h.
Acceleration, sec.
- Third speed, 10–25, 10–45, 5–60, 10–60, 20–60, 30–60 m.p.h.
- Second speed, 5–45 m.p.h.
- First speed, 5–25 m.p.h.

Hill climb on 7 to 11% grade; start at 10, 20, 30 m.p.h.
- Maximum speed, m.p.h.
- Speed at top of hill, m.p.h.
- Time, sec., to go over top

Fuel Consumption
- From 5 m.p.h. to max. speed, by 5 m.p.h. intervals

Procedure for Car Performance Tests

BEFORE LEAVING GARAGE.—1. Check oil, water and gas; see that oil is at proper level, and that radiator and gas tank are full. Check tire pressures, and inflate to standard pressure for car under test. See that brakes do not drag. 2. Install thermometer in water outlet from engine. 3. Attach fifth-wheel speedometer; load car with sufficient ballast to bring weight of instruments, passengers and ballast up to 450 lb. Observer should have a good 30-sec. single-hand stop-watch and performance data sheets.

Warm Up.—1. Drive large cars 8 or 10 miles at speed of 30 to 40 m.p.h., and small cars 25 to 35 m.p.h. to warm oil and reduce friction before starting tests.

On Straightaway.—1. Make several acceleration runs at from 10 to 25 m.p.h. at different temperatures, from cool to boiling, to find temperature of best performance; record best temperature on data sheet, and hold to it on subsequent runs within $\pm 10°$ F. 2. Run at least five consistent accelerations in each direction from 10 to 25 m.p.h. Car velocity should not exceed 8 m.p.h., preferably less, when accelerator is opened. If car is braked to reduce speed, it should roll for an instant before accelerating. Record time in seconds, beginning when fifth-wheel speedometer pointer passes 10 m.p.h. and ending when it passes 25 m.p.h. 3. Repeat (2) for 10–45, 5–60, 10–60, 20–60, 30–60 m.p.h. 4. Find and record minimum speed, in both directions, at which car will idle, *i.e.*, minimum speed with accelerator as far as possible from floorboard. 5. Find and record lowest speed in each direction from which car will accelerate smoothly without bucking. This speed generally is less than that found by (4); usually car must be braked to obtain it. Driver should brake car to speed desired by observer and note whether or not the get-away is good. Trials at different speeds usually will show a certain well-defined speed below which car will not accelerate; this speed is recorded.

Hill Climb.—Make three series of tests on a grade, starting at 10, 20 and 30 m.p.h., respectively. Take readings, in seconds, at each starting speed, of time to climb grade, maximum speed, and speed at end of grade.

Maximum Speed.—Return to straightaway and record maximum speed in each direction; run sufficient distance to maintain constant speed for 0.1 to 0.2 mi.

General Notes.—Record date, hour, temperature and speedometer mileage midway of test, and approximate direction and velocity of wind. In general, tests should not be made with wind stronger than moderate, except in cases of brake tests. Observers should sign all test sheets.

Procedure for Making Economy Tests

BEFORE LEAVING GARAGE.—Follow same procedure as for car performance test. Also install fuel measuring burettes and tank in car. Disconnect carburetor from gasoline pump; connect burettes to carburetor. Provide fuel economy data sheets.

On Straightaway.—1. Make at least four trips from end to end of straightaway to warm oil and reduce friction. Make final trip at speed at which first test run is to be made. 2. To start test, fill one burette to zero mark, engine running meanwhile on other burette. Bring car up to desired speed, and have it constant, before reaching starting post. On passing starting post, throw valve on lower 3-way cock to take fuel from first burette, and start stop-watch. On passing finish post reverse valve on lower 3-way cock and stop stop-watch; record time and cu. cm. of fuel used. 3. Length of course should be one mile. Start first test run at 10 m.p.h.; speed of succeeding test runs should increase by 5 m.p.h. increments unless otherwise instructed. Make two runs in each direction at each speed, to and including 35 m.p.h., and one run in each direction at a higher speed.

General.—In addition to time and fuel readings, data sheet should show: Car number, name and special features; weight of instruments and passengers; date, barometer, temperature, direction and approximate strength of wind; odometer.

Procedure for Brake Tests

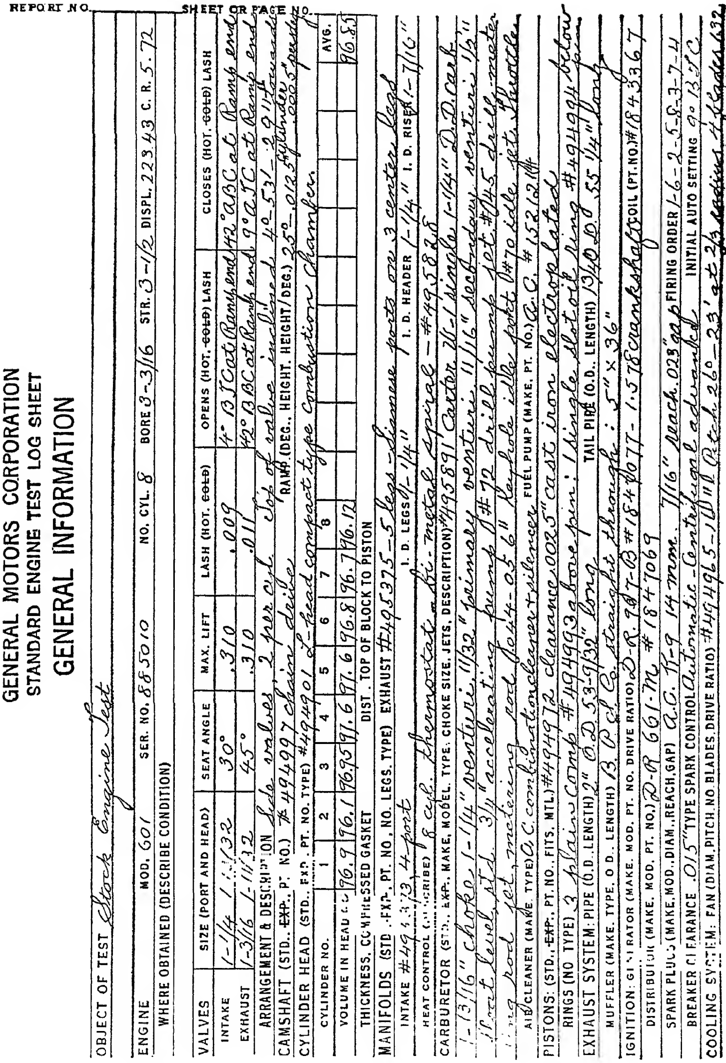
BEFORE LEAVING GARAGE.—1. Run car on brake testing machine; adjust and properly equalize brakes to slide the wheels. Check on pavement outside of garage. 2. Install decelerometer and storage battery in car; attach pedal pressure element to brake pedal. See that decelerometer motor runs freely.

On Straightaway.—1. Level decelerometer; deceleration pen should read zero with car standing still. 2. To start test, bring car to speed, start decelerometer motor, declutch and apply brakes. 3. Take readings from zero pedal pressure to maximum; increments should be as uniform as possible. Make tests in both directions to eliminate effect of wind. About 20 properly spaced readings usually are sufficient, though more are better.

In Office.—1. Read data from rolls by celluloid template in conjunction with the fixed base pen of the decelerometer; take readings where deceleration and pedal pressure are approximately constant; average slight variations by a light average pencil line, and read from this line. 2. Plot points obtained and draw an average curve.

9. ENGINE TESTS

The procedure used in engine testing varies in the different laboratories. Since so many variables materially affect results, information obtained at different places is not at all comparative unless a standard procedure is used. The General Motors Corporation has adopted an engine test code which is used in the engineering departments of its divisions. An abstract of this code follows.



REPORT NO. ____ SHEET OR PAGE NO. ____

GENERAL MOTORS CORPORATION
STANDARD ENGINE TEST LOG SHEET
GENERAL INFORMATION

OBJECT OF TEST Stock Engine Test

ENGINE MOD. 601 SER. NO. 885010 NO. CYL. 8 BORE 3-3/16 STR. 3-1/2 DISPL. 223.43 C. R. 5.72
WHERE OBTAINED (DESCRIBE CONDITION)

VALVES	SIZE (PORT AND HEAD)	SEAT ANGLE	MAX. LIFT	LASH (HOT. ~~COLD~~)	OPENS (HOT. ~~COLD~~) LASH	CLOSES (HOT. ~~COLD~~) LASH
INTAKE	1-1/4 1-11/32	30°	.310	.009	4° BTC at Ramp end	42° ABC at Ramp end
EXHAUST	1-3/16 1-11/32	45°	.310	.011	42° BBC at Ramp end	9° ATC at Ramp end

ARRANGEMENT & DESCRIPTION Side valves, 2 per cyl. Top of valve inclined 4°-53'-29" towards cylinder
CAMSHAFT (STD., ~~EXP.~~, PT. NO.) #494997 chain drive RAMP (DEG., HEIGHT, HEIGHT/DEG.) 25°-.0125"-.0005" per deg.
CYLINDER HEAD (STD., EXP., PT. NO. TYPE) #494901 L-head compact type combustion chamber.

CYLINDER NO.	1	2	3	4	5	6	7	8	AVG.
VOLUME IN HEAD c.c.	96.9	96.1	96.95	97.6	97.6	96.8	96.7	96.12	96.85

THICKNESS, COMPRESSED GASKET DIST. TOP OF BLOCK TO PISTON
MANIFOLDS (STD., EXP., PT. NO., NO. LEGS, TYPE) EXHAUST #495375-5 legs-Siamese ports on 3 center legs
INTAKE #494373, 4-port I. D. LEGS 1-1/4" I. D. HEADER 1-1/4" I. D. RISER 1-7/16"
HEAT CONTROL (DESCRIBE) 8 cyl. thermostat bi-metal spiral-#495828
CARBURETOR (STD., EXP., MAKE, MODEL, TYPE, CHOKE SIZE, JETS, DESCRIPTION) #495891 Carter W-1 single 1-1/4" D.D. carb. 1-13/16" choke 1-1/4" venturi, 11/32" primary venturi, 11/16" secondary venturi 1/2" float level, std. 3/4" accelerating pump #72 drill pump jet #45 drill metering rod jet, metering rod .044-.056" keyhole idle port #70 idle jet. Throttle
AIR CLEANER (MAKE, TYPE) A.C. combination cleaner + silencer FUEL PUMP (MAKE, PT. NO.) A.C. #1521214
PISTONS: (STD., ~~EXP.~~, PT. NO., FITS, MTL.) #494972 clearance .0025" cast iron electroplated
RINGS (NO. TYPE) 3 plain comp. #494993 above pin: 1 single slot oil ring #494994 below pin
EXHAUST SYSTEM: PIPE (O.D., LENGTH) 2" O.D. 53-9/32" long TAIL PIPE (O.D., LENGTH) 13/4 O.D. 55 1/4" long
MUFFLER (MAKE, TYPE, O.D., LENGTH) B. P. & Co. straight through: 5" x 36"
IGNITION: GENERATOR (MAKE, MOD., PT. NO., DRIVE RATIO) D-R. 907-B #1840077-1.578 crankshaft COIL (PT. NO.) #1843367
DISTRIBUTOR (MAKE, MOD., PT. NO.) D-R 661-M #1847069
SPARK PLUG (MAKE, MOD., DIAM., REACH, GAP) A.C. K-9 14 mm. 7/16" reach .025" gap FIRING ORDER 1-6-2-5-8-3-7-4
BREAKER CLEARANCE .015" TYPE SPARK CONTROL Automatic-Centrifugal advance INITIAL AUTO SETTING 9° BTC
COOLING SYSTEM: FAN (DIAM., PITCH, NO. BLADES, DRIVE RATIO) #494965-16" Pitch 26°-23' at 2/3 radius 4 blades 1.32 crankshaft

FIG. 68. Engine Test Log Sheet

GENERAL ENGINE INFORMATION.—Information on engine specifications, equipment and settings shall be obtained at the time the test is made and permanently attached to the test data. See Fig. 68.

VALVE LIFT DIAGRAMS.—Valve lift diagrams for a representative cylinder shall accompany data for any standard test. A special curve sheet (Fig. 69) is suitable for plotting full lift diagrams and exaggerated diagrams for the ramp portion of the lift curve.

REQUIRED ENGINE EQUIPMENT.—1. Transmission and clutch (omitted in dynamometer testing). 2. Intake system includes, when standard for the engine: *a*, Hot air pipe; *b*, air cleaner; *c*, carburetor; *d*, choke valve (in open position); *e*, riser (uptake or downtake); *f*, manifold heat

control system; *g*, intake manifold. Item *a* may be omitted, if necessary, for volumetric test. If air measuring equipment is used, it is noted on log sheets.

EXHAUST SYSTEM.—Tests are made with and without mufflers. For tests with muffler, the standard exhaust system includes: *a*, Exhaust manifold; *b*, automatic heat valve; *c*, exhaust pipe; *d*, tail-pipe. For tests without muffler, the manifold is connected directly to dynamometer exhaust system.

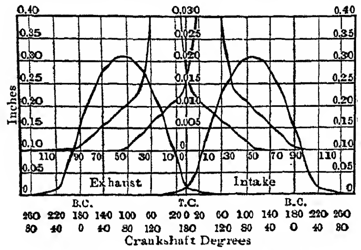


FIG. Valve Lift Diagrams

Cooling System.—The standard water pump is used, operated at normal speed; car radiator is omitted; standard cooling fan is used, driven at normal speed with normal belt tension. For the water heat-rejection test, direct cooling is required.

Ignition System.—Standard generator is used, operating at normal speed with leads grounded to eliminate charging; standard distributor and coil are used, even with manual control of spark.

Fuel System.—Standard fuel pump and carburetor are used.

Lubricating System.—Standard oil pump is used.

Engine Adjustments and Settings

Throttle.—Most tests are made at full throttle. For tests at part throttle, arbitrary standard road power data are used, simulating level-road driving requirements. See Fig. 70. The curves are based on the formula

$$\text{Hp.} = (0.017W + 0.018AV^2)(V/375)$$

W = car weight, lb. based on 15 lb. per cu. in.; A = projected frontal area, sq. ft.; V = road speed, m.p.h.; K and K_1 = constants. See Table 12 for values of K and K_1.

The curves, Fig. 70, give the power to be used in the road power tests. Torque may be calculated by the equation $Q = 5250\ \text{Hp.}/N$.

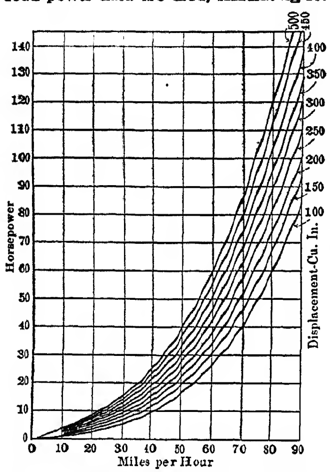


FIG. 70. Passenger Car Standard Road Power Data Based on Engine Displacement

Engine speed for a given road speed may be calculated from rear axle ratio and tire circumference, or assumed to be 50 r.p.m. per m.p.h., which is the average for passenger cars.

Carburetor.—The best compromise settings are determined and maintained constant throughout a series of tests. The following type adjustments are mentioned: *a*, Leanest for best torque, any condition; *b*, richest for best torque, any condition; *c*, combination of good idling at 200–300 r.p.m., and leanest for best torque at 1000 r.p.m.; *d*, variable fuel flow, any condition, to obtain the progressive effect of this variable over the range from best torque to best economy. Adjustment *c* is usually used.

Intake Manifold Heat Control where optional, is maintained in the *summer* position; where automatic, is so maintained.

Ignition Settings.—For all power tests, spark plug gaps and breaker clearances are adjusted standard.

Valve Lash is adjusted as specified by manufacturer. As the lash used in the car is not always suitable for dynamometer testing, it is necessary to check it under running conditions. Standard lash is used for the compression test.

Preliminary Engine Checking Procedure.—Prior to making any standard test: 1. Engine is

Table 12.—Values of K and K_1 in Horsepower Formula

Engine Displacement, cu. in.	Weight, lb.	Frontal Area, sq. ft.	K	K_1
100	1500	21.65	0.017	0.03897
150	2250	23.47	0.017	0.04225
200	3000	25.30	0.017	0.04554
250	3750	27.14	0.017	0.04885
300	4500	28.98	0.017	0.05216
350	5250	30.82	0.017	0.05548
400	6000	32.66	0.017	0.05879
450	6750	34.50	0.017	0.06210
500	7500	36.34	0.017	0.06541

thoroughly conditioned; 2. All engine equipment and settings are checked and recorded; 3. A performance check test is made.

FUELS AND LUBRICANTS. Fuels.—Special fuels are used for engines requiring them; normally, any good grade of fuel may be used. A fuel of given knock rating is not required, but knock rating must be stated.

Lubricants.—Any good grade of engine oil is used. It is required only that the viscosity be that recommended by the manufacturer for summer conditions.

ENGINE STABILITY.—Performance data are obtained under stabilized operating conditions. Sudden heat changes should be avoided. In any test, the series of runs should progress continuously. For any test run: 1. No data are taken until torque, speed and temperatures have been maintained without noticeable change for at least one minute; there should be no important change in these items during the test run; 2. No adjustments of engine settings are made during the run; 3. As an overall check, two separate torque readings are recorded, one before and one after recording all other data.

Torque, Speed and Power

TORQUE.—The electric cradle dynamometer is used. It should have absorption capacity and speed rating to permit testing under any simulated road condition.

SPEED.—Instrumentation. An accurately checked indicating tachometer is used for compression and friction tests. An accurately calibrated revolution counter operated from dynamometer or engine shaft and synchronized with a time measuring device is used for the power test.

Time.—The conventional stop watch is satisfactory. Time units operated from stabilized alternating-current circuits give a finer degree of precision.

A Speed Measurement Interval of at least 40 sec., preferably not less than 1 min., is used when measuring speed with the automatically synchronized counter-watch combinations; for hand operation, not less than 60 sec.

Limiting Speed is defined either by maximum car speed or by limitations of testing equipment.

Computations

Notations.—B.Hp., F.Hp., I.Hp. = brake, friction and indicated horsepower, respectively; L = average of two torque scale readings, lb.; N = r.p.m.; K = dynamometer constant; R, r = torque scale arm, ft. and in., respectively; D = engine displacement, cu. in.; P = M.E.P. = mean effective pressure, lb. per sq. in.; Q = torque, lb-ft.; B_c = carburetor air pressure, in. Hg, absolute; e = water vapor pressure, in. Hg, absolute; T_c = observed, and T_s = standard, carburetor air temperature, deg. F., absolute; E_m = mechanical efficiency.

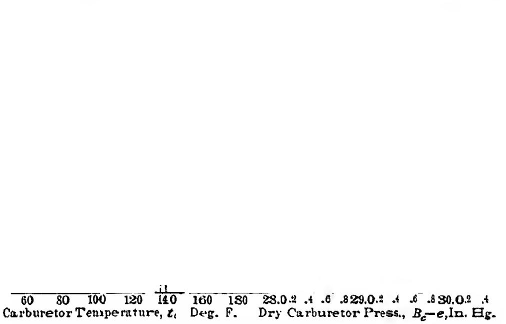


FIG. 71. Power Correction for Pressure, Humidity and Temperature

OBSERVED POWER.—B.Hp. = $N\,L/K$. $K = 5250/R = 63000/r$. K = 6000 for $r = 10.5$; 4000 for $r = 15.75$; 3000 for $r = 21$. For 4-stroke cycle engine, Hp. = $DPN/792{,}000$ = $0.0001903\,NQ$. Torque is computed directly from torque scale reading. $Q = LR$. For 4-stroke cycle engine, Q = 5250 Hp./N = $5250\,L/K$ = $DP/150.8$. Mean effective pressure is computed directly from torque scale reading. For 4-stroke cycle engine, $P = 150.8\,LR/D$ = $150.8\,Q/D$ = 792,000 Hp./DN = $792{,}000\,L/DK$. A curve is plotted of observed motoring F.Hp.* *vs.* speed. Values from this curve for the speed of the brake power test are added to brake power to obtain indicated power.

CORRECTED POWER.—Full-throttle power data, including power at borderline detonation, are corrected to standard conditions for carburetor air. Part-throttle data are not corrected. Standard pressure is 29.92 in. Hg; atmospheric water vapor pressure standard temperatures are 60° F., 103° F., 150° F. The standard for stock tests is 103° F.

Combined correction factor for indicated power = $F_{ci} = [29.42/(B_c - e)]\sqrt{T_c/T_s}$. See Table 13 and Fig. 71.

Combined correction factor for full throttle brake power = $F_{cb} = [29.42/(B_c - e)]\sqrt{(T_c/T_s)} + E_m$. See Table 14.

Mechanical Efficiency is based on corrected power.

SPARK TIMING.—The measured timing for a representative cylinder is sufficient; timing for all cylinders is often desirable. The spark control is rigged to permit maintenance of any

(*Continued on p. 14-92*)

* Obtained by driving engine by an electric dynamometer. See p. 14–94.

Tabl orrec ı Fa r Pressı mpe Hı idity

103° Da perature.

Carb. Temp., deg. F.	Dry Carburetor Pressure, in. of Mercury																	
	27.75	27.80	27.85	27.90	27.95	28.00	28.05	28.10	28.15	28.20	28.25	28.30	28.35	28.40	28.45	28.50	28.55	28.60
70	1.029	1.027	1.025	1.024	1.022	1.020	1.018	1.016	1.015	1.013	1.011	1.009	1.007	1.005	1.004	1.002	1.000	0.998
72	1.031	1.029	1.027	1.025	1.024	1.022	1.020	1.018	1.016	1.014	1.013	1.011	1.009	1.007	1.005	1.004	1.002	1.000
74	1.033	1.031	1.029	1.026	1.025	1.023	1.021	1.020	1.018	1.016	1.014	1.012	1.011	1.009	1.007	1.005	1.004	1.002
76	1.035	1.033	1.031	1.030	1.028	1.026	1.024	1.022	1.021	1.019	1.017	1.015	1.013	1.011	1.010	1.008	1.006	1.005
78	1.037	1.035	1.033	1.031	1.029	1.027	1.026	1.024	1.022	1.020	1.018	1.016	1.015	1.013	1.011	1.010	1.008	1.006
80	1.039	1.037	1.035	1.034	1.032	1.030	1.028	1.026	1.025	1.023	1.021	1.019	1.017	1.015	1.014	1.012	1.010	1.008
82	1.041	1.039	1.037	1.036	1.034	1.032	1.030	1.028	1.026	1.025	1.023	1.021	1.019	1.017	1.016	1.014	1.012	1.010
84	1.043	1.041	1.039	1.037	1.035	1.034	1.032	1.030	1.028	1.026	1.025	1.023	1.021	1.019	1.017	1.015	1.014	1.012
86	1.044	1.042	1.041	1.039	1.037	1.035	1.033	1.031	1.030	1.028	1.026	1.024	1.022	1.020	1.019	1.017	1.015	1.013
88	1.047	1.045	1.043	1.041	1.040	1.038	1.036	1.034	1.032	1.030	1.028	1.026	1.025	1.023	1.021	1.019	1.017	1.016
90	1.048	1.046	1.045	1.043	1.041	1.039	1.037	1.035	1.034	1.032	1.030	1.028	1.026	1.024	1.023	1.021	1.019	1.017
92	1.050	1.048	1.046	1.044	1.042	1.040	1.038	1.036	1.035	1.033	1.031	1.029	1.027	1.026	1.024	1.022	1.020	1.018
94	1.052	1.050	1.048	1.046	1.044	1.042	1.040	1.038	1.037	1.035	1.033	1.031	1.029	1.028	1.026	1.024	1.022	1.020
96	1.054	1.052	1.050	1.048	1.046	1.044	1.042	1.040	1.038	1.037	1.035	1.033	1.031	1.029	1.028	1.026	1.024	1.022
98	1.055	1.054	1.052	1.050	1.048	1.046	1.044	1.042	1.040	1.039	1.037	1.035	1.033	1.031	1.029	1.028	1.026	1.024
100	1.057	1.055	1.054	1.052	1.050	1.049	1.046	1.044	1.042	1.040	1.038	1.037	1.035	1.033	1.031	1.030	1.028	1.026
102	1.059	1.057	1.055	1.053	1.051	1.050	1.048	1.046	1.044	1.042	1.040	1.038	1.037	1.035	1.033	1.031	1.030	1.028
104	1.061	1.059	1.057	1.055	1.053	1.052	1.050	1.048	1.046	1.044	1.042	1.040	1.039	1.037	1.035	1.033	1.031	1.030
106	1.063	1.061	1.059	1.057	1.055	1.053	1.052	1.050	1.048	1.046	1.044	1.042	1.040	1.039	1.037	1.035	1.033	1.031
108	1.065	1.063	1.061	1.059	1.057	1.055	1.053	1.051	1.050	1.048	1.046	1.044	1.042	1.040	1.039	1.037	1.035	1.033
110	1.067	1.065	1.063	1.061	1.059	1.057	1.055	1.053	1.051	1.050	1.048	1.046	1.044	1.042	1.040	1.039	1.037	1.035
112	1.068	1.067	1.065	1.063	1.061	1.059	1.057	1.055	1.053	1.051	1.049	1.048	1.046	1.044	1.042	1.040	1.039	1.037
114	1.070	1.069	1.066	1.065	1.063	1.061	1.059	1.057	1.055	1.053	1.051	1.049	1.048	1.046	1.044	1.042	1.040	1.039
116	1.072	1.070	1.068	1.066	1.064	1.063	1.061	1.059	1.057	1.055	1.053	1.051	1.050	1.048	1.046	1.044	1.042	1.040
118	1.074	1.072	1.070	1.068	1.066	1.065	1.063	1.061	1.059	1.057	1.055	1.053	1.051	1.050	1.048	1.046	1.044	1.042
120	1.076	1.074	1.072	1.070	1.068	1.066	1.065	1.062	1.061	1.059	1.057	1.055	1.053	1.051	1.050	1.048	1.046	1.044
122	1.078	1.076	1.074	1.072	1.070	1.068	1.066	1.064	1.062	1.061	1.059	1.056	1.055	1.053	1.051	1.050	1.048	1.046
124	1.080	1.078	1.076	1.074	1.072	1.070	1.068	1.066	1.064	1.063	1.060	1.059	1.057	1.055	1.053	1.051	1.050	1.048
126	1.081	1.079	1.077	1.076	1.074	1.072	1.070	1.068	1.066	1.064	1.062	1.060	1.059	1.057	1.055	1.053	1.051	1.049
128	1.083	1.081	1.079	1.077	1.075	1.074	1.072	1.070	1.068	1.066	1.064	1.062	1.060	1.058	1.057	1.055	1.053	1.051
130	1.085	1.083	1.081	1.079	1.077	1.076	1.074	1.072	1.070	1.068	1.066	1.064	1.062	1.060	1.059	1.057	1.055	1.053
132	1.087	1.085	1.083	1.081	1.079	1.077	1.075	1.073	1.072	1.070	1.068	1.066	1.064	1.062	1.060	1.059	1.057	1.055
134	1.089	1.087	1.085	1.083	1.081	1.079	1.077	1.075	1.073	1.072	1.069	1.068	1.066	1.064	1.062	1.060	1.058	1.057
136	1.091	1.089	1.087	1.085	1.083	1.081	1.079	1.077	1.075	1.073	1.071	1.069	1.068	1.066	1.064	1.062	1.060	1.058
138	1.093	1.091	1.089	1.087	1.085	1.083	1.081	1.079	1.077	1.075	1.073	1.071	1.070	1.068	1.066	1.064	1.062	1.060
140	1.094	1.092	1.091	1.089	1.086	1.085	1.083	1.081	1.079	1.077	1.075	1.073	1.071	1.069	1.068	1.066	1.064	1.062

Table 13.—Indicated Power Correction Factors (*Continued*)

Carb. Temp., deg. F.	Dry Carburetor Pressure, in. of Mercury																	
	28.65	28.70	28.75	28.80	28.85	28.90	28.95	29.00	29.05	29.10	29.15	29.20	29.25	29.30	29.35	29.40	29.45	29.50
70	0.996	0.994	0.993	0.992	0.990	0.988	0.986	0.984	0.983	0.982	0.980	0.978	0.976	0.974	0.973	0.971	0.970	0.968
72	.998	.996	.995	.993	.991	.989	.988	.986	.984	.983	.981	.979	.978	.976	.974	.972	.971	.969
74	.999	.999	.997	.995	.994	.992	.990	.989	.987	.985	.984	.982	.980	.978	.977	.975	.973	.972
76	1.003	1.001	.999	.998	.996	.994	.993	.991	.990	.988	.986	.984	.982	.981	.979	.977	.976	.974
78	1.004	1.002	1.000	1.000	.998	.996	.995	.993	.992	.990	.988	.986	.984	.982	.981	.980	.978	.976
80	1.006	1.005	1.003	1.001	.999	.998	.996	.994	.993	.991	.989	.988	.986	.984	.983	.981	.979	.977
82	1.009	1.007	1.005	1.003	1.002	1.000	.998	.996	.995	.993	.991	.990	.988	.986	.984	.982	.981	.979
84	1.010	1.008	1.006	1.005	1.003	1.001	1.000	.998	.997	.995	.992	.991	.989	.988	.985	.983	.982	.981
86	1.012	1.010	1.008	1.006	1.004	1.003	1.001	.999	.998	.996	.994	.992	.991	.989	.987	.986	.984	.982
88	1.014	1.012	1.010	1.009	1.007	1.005	1.003	1.001	1.000	.997	.995	.994	.992	.990	.989	.987	.985	.983
90	1.015	1.013	1.012	1.010	1.008	1.007	1.005	1.003	1.001	1.000	.998	.996	.994	.993	.991	.989	.988	.986
92	1.017	1.014	1.013	1.011	1.009	1.008	1.006	1.004	1.002	1.001	.999	.997	.996	.994	.992	.991	.989	.987
94	1.019	1.016	1.015	1.013	1.012	1.010	1.008	1.006	1.005	1.003	1.001	.999	.998	.996	.994	.993	.991	.990
96	1.020	1.018	1.017	1.015	1.013	1.011	1.010	1.007	1.006	1.005	1.003	1.001	.999	.998	.996	.994	.993	.991
98	1.022	1.020	1.019	1.017	1.015	1.013	1.012	1.010	1.008	1.006	1.005	1.003	1.001	1.000	.998	.996	.995	.993
100	1.024	1.022	1.021	1.019	1.017	1.015	1.013	1.011	1.010	1.008	1.007	1.005	1.003	1.001	1.000	.998	.996	.995
102	1.026	1.024	1.022	1.021	1.019	1.017	1.015	1.013	1.012	1.010	1.008	1.007	1.005	1.003	1.001	1.000	.998	.996
104	1.028	1.026	1.024	1.022	1.021	1.019	1.017	1.015	1.013	1.012	1.010	1.008	1.007	1.005	1.003	1.001	1.000	.998
106	1.030	1.027	1.026	1.024	1.022	1.021	1.019	1.016	1.015	1.014	1.012	1.010	1.008	1.007	1.005	1.003	1.002	1.000
108	1.031	1.029	1.027	1.026	1.024	1.022	1.020	1.019	1.017	1.015	1.014	1.012	1.010	1.009	1.007	1.005	1.003	1.002
110	1.033	1.031	1.030	1.028	1.026	1.024	1.022	1.020	1.019	1.017	1.016	1.014	1.012	1.010	1.009	1.007	1.005	1.003
112	1.035	1.033	1.031	1.030	1.028	1.026	1.024	1.022	1.021	1.019	1.017	1.016	1.014	1.012	1.010	1.009	1.007	1.005
114	1.037	1.035	1.033	1.031	1.030	1.028	1.026	1.024	1.022	1.021	1.019	1.017	1.016	1.014	1.012	1.010	1.009	1.007
116	1.039	1.036	1.035	1.033	1.031	1.030	1.028	1.025	1.024	1.022	1.021	1.019	1.017	1.016	1.014	1.012	1.010	1.009
118	1.040	1.038	1.037	1.035	1.033	1.031	1.030	1.027	1.026	1.024	1.023	1.021	1.019	1.017	1.016	1.014	1.012	1.010
120	1.042	1.040	1.038	1.037	1.035	1.033	1.031	1.029	1.028	1.026	1.024	1.023	1.021	1.019	1.017	1.016	1.014	1.012
122	1.044	1.042	1.040	1.039	1.037	1.035	1.033	1.031	1.030	1.028	1.026	1.024	1.023	1.021	1.019	1.017	1.016	1.014
124	1.046	1.044	1.042	1.040	1.039	1.037	1.035	1.033	1.031	1.030	1.028	1.026	1.024	1.023	1.021	1.019	1.017	1.015
126	1.048	1.045	1.044	1.042	1.040	1.039	1.037	1.034	1.033	1.031	1.030	1.028	1.026	1.024	1.023	1.021	1.020	1.018
128	1.049	1.047	1.046	1.044	1.042	1.040	1.038	1.036	1.035	1.033	1.031	1.029	1.028	1.026	1.024	1.022	1.021	1.019
130	1.051	1.049	1.048	1.046	1.044	1.042	1.040	1.038	1.037	1.035	1.033	1.031	1.030	1.028	1.026	1.024	1.023	1.021
132	1.053	1.051	1.049	1.047	1.046	1.044	1.042	1.040	1.038	1.037	1.035	1.033	1.031	1.030	1.028	1.026	1.024	1.023
134	1.055	1.053	1.051	1.049	1.047	1.046	1.044	1.041	1.040	1.038	1.037	1.035	1.033	1.031	1.030	1.028	1.026	1.024
136	1.057	1.054	1.053	1.051	1.049	1.047	1.046	1.043	1.042	1.040	1.038	1.037	1.035	1.033	1.031	1.029	1.028	1.026
138	1.059	1.056	1.055	1.053	1.051	1.049	1.047	1.045	1.044	1.042	1.040	1.039	1.037	1.035	1.033	1.031	1.030	1.028
140	1.060	1.058	1.056	1.054	1.053	1.051	1.049	1.047	1.045	1.044	1.042	1.040	1.038	1.037	1.035	1.033	1.031	1.030

Table 14.—Full Throttle Brake Power Correction Factors for Pressure, Temperature and Humidity

(Based on $E_m = 0.85$. 103° F. Datum Temperature)

NOTE: These factors are to be applied directly to full throttle brake power or torque. They should not be used for part throttle power.

Carb. Temp., deg. F.	Dry Carburetor Pressure, in. of Mercury																	
	27.75	27.80	27.85	27.90	27.95	28.00	28.05	28.10	28.15	28.20	28.25	28.30	28.35	28.40	28.45	28.50	28.55	28.60
70	1.034	1.032	1.029	1.028	1.026	1.024	1.021	1.019	1.018	1.015	1.013	1.011	1.008	1.006	1.005	1.002	1.000	.998
72	1.036	1.034	1.032	1.029	1.028	1.026	1.024	1.021	1.019	1.016	1.015	1.013	1.011	1.008	1.006	1.005	1.002	1.000
74	1.038	1.036	1.034	1.031	1.029	1.027	1.025	1.024	1.021	1.019	1.016	1.014	1.013	1.011	1.008	1.006	1.005	1.002
76	1.041	1.039	1.036	1.035	1.033	1.031	1.028	1.026	1.025	1.021	1.020	1.018	1.015	1.013	1.012	1.009	1.007	1.006
78	1.044	1.041	1.039	1.036	1.034	1.032	1.031	1.028	1.026	1.024	1.021	1.019	1.018	1.015	1.013	1.012	1.009	1.007
80	1.046	1.044	1.041	1.040	1.038	1.035	1.033	1.031	1.029	1.027	1.025	1.022	1.020	1.018	1.016	1.014	1.012	1.009
82	1.048	1.046	1.044	1.042	1.040	1.038	1.035	1.033	1.031	1.029	1.027	1.025	1.022	1.020	1.019	1.016	1.014	1.012
84	1.051	1.048	1.046	1.044	1.041	1.040	1.038	1.035	1.033	1.031	1.029	1.027	1.025	1.022	1.020	1.018	1.016	1.014
86	1.052	1.049	1.048	1.046	1.044	1.041	1.039	1.036	1.035	1.033	1.031	1.028	1.026	1.024	1.022	1.020	1.018	1.015
88	1.055	1.053	1.051	1.048	1.047	1.045	1.042	1.040	1.038	1.035	1.033	1.031	1.029	1.027	1.025	1.022	1.020	1.019
90	1.056	1.054	1.053	1.051	1.048	1.046	1.044	1.041	1.040	1.038	1.035	1.033	1.031	1.028	1.027	1.025	1.022	1.020
92	1.059	1.056	1.054	1.052	1.049	1.047	1.045	1.042	1.041	1.039	1.036	1.034	1.032	1.031	1.028	1.026	1.024	1.021
94	1.061	1.059	1.056	1.054	1.052	1.049	1.047	1.045	1.044	1.041	1.039	1.036	1.034	1.033	1.031	1.028	1.025	1.024
96	1.064	1.061	1.059	1.056	1.054	1.052	1.049	1.047	1.045	1.044	1.041	1.039	1.036	1.034	1.033	1.031	1.028	1.026
98	1.065	1.064	1.061	1.059	1.056	1.054	1.052	1.049	1.047	1.046	1.044	1.041	1.039	1.036	1.034	1.033	1.031	1.028
100	1.067	1.065	1.064	1.061	1.059	1.058	1.054	1.052	1.049	1.047	1.045	1.044	1.041	1.039	1.036	1.035	1.033	1.031
102	1.069	1.067	1.065	1.062	1.060	1.059	1.056	1.054	1.052	1.049	1.047	1.045	1.044	1.041	1.039	1.036	1.035	1.033
104	1.072	1.069	1.067	1.065	1.062	1.061	1.059	1.056	1.054	1.052	1.049	1.047	1.046	1.044	1.041	1.039	1.036	1.035
106	1.074	1.072	1.069	1.067	1.065	1.062	1.061	1.059	1.056	1.054	1.052	1.049	1.047	1.046	1.044	1.041	1.039	1.036
108	1.076	1.074	1.072	1.069	1.067	1.065	1.062	1.060	1.059	1.056	1.054	1.052	1.049	1.047	1.046	1.044	1.041	1.039
110	1.079	1.076	1.074	1.072	1.069	1.067	1.065	1.062	1.060	1.059	1.056	1.054	1.052	1.049	1.047	1.046	1.044	1.041
112	1.080	1.079	1.076	1.074	1.071	1.069	1.067	1.065	1.062	1.060	1.058	1.056	1.054	1.052	1.049	1.047	1.046	1.044
114	1.082	1.081	1.079	1.076	1.074	1.072	1.069	1.067	1.065	1.062	1.060	1.058	1.056	1.054	1.052	1.049	1.047	1.046
116	1.085	1.082	1.080	1.078	1.075	1.074	1.072	1.069	1.067	1.065	1.062	1.060	1.059	1.056	1.054	1.052	1.049	1.047
118	1.087	1.085	1.082	1.080	1.078	1.076	1.074	1.072	1.069	1.067	1.065	1.062	1.060	1.059	1.056	1.054	1.052	1.049
120	1.089	1.087	1.085	1.082	1.080	1.078	1.076	1.073	1.072	1.069	1.067	1.065	1.062	1.060	1.059	1.056	1.054	1.052
122	1.092	1.089	1.087	1.085	1.082	1.080	1.078	1.075	1.073	1.072	1.069	1.066	1.065	1.062	1.060	1.059	1.056	1.054
124	1.094	1.092	1.089	1.087	1.085	1.082	1.080	1.078	1.075	1.074	1.071	1.069	1.067	1.065	1.062	1.060	1.059	1.056
126	1.095	1.093	1.092	1.089	1.087	1.085	1.082	1.080	1.078	1.075	1.073	1.071	1.069	1.067	1.065	1.062	1.060	1.058
128	1.098	1.095	1.093	1.091	1.088	1.087	1.085	1.082	1.080	1.078	1.075	1.073	1.071	1.068	1.067	1.065	1.062	1.060
130	1.100	1.098	1.095	1.093	1.091	1.089	1.087	1.085	1.082	1.080	1.078	1.075	1.073	1.071	1.069	1.067	1.065	1.062
132	1.102	1.100	1.098	1.095	1.093	1.091	1.088	1.086	1.085	1.082	1.080	1.078	1.075	1.073	1.071	1.069	1.067	1.065
134	1.105	1.102	1.100	1.098	1.095	1.093	1.091	1.088	1.086	1.085	1.081	1.080	1.078	1.075	1.073	1.071	1.068	1.067
136	1.107	1.105	1.102	1.100	1.098	1.095	1.093	1.091	1.088	1.086	1.084	1.081	1.080	1.078	1.075	1.073	1.071	1.068
138	1.109	1.107	1.105	1.102	1.100	1.098	1.095	1.093	1.091	1.088	1.086	1.082	1.081	1.080	1.076	1.075	1.073	1.071
140	1.111	1.108	1.107	1.105	1.101	1.100	1.098	1.095	1.093	1.091	1.088	1.085	1.084	1.081	1.080	1.078	1.075	1.073

Carb. Temp., deg. F.	Dry Carburetor Pressure, in. of Mercury																	
	28.65	28.70	28.75	28.80	28.85	28.90	28.95	29.00	29.05	29.10	29.15	29.20	29.25	29.30	29.35	29.40	29.45	29.50
70	0.995	0.993	0.992	0.991	0.988	0.986	0.984	0.981	0.980	0.979	0.976	0.974	0.972	0.969	0.968	0.966	0.965	0.962
72	.998	.995	.994	.992	.989	.987	.986	.984	.981	.980	.978	.975	.974	.972	.969	.967	.966	.964
74	.999	.998	.996	.994	.993	.991	.988	.987	.985	.982	.981	.979	.976	.974	.973	.971	.968	.967
76	1.004	1.001	.999	.998	.995	.993	.992	.989	.988	.985	.984	.981	.979	.978	.975	.973	.972	.969
78	1.005	1.002	1.001	1.000	.998	.995	.994	.992	.991	.988	.986	.984	.981	.980	.978	.976	.974	.972
80	1.007	1.006	1.004	1.001	.999	.998	.995	.993	.992	.989	.987	.986	.984	.981	.980	.978	.975	.973
82	1.011	1.008	1.006	1.004	1.002	1.000	.998	.995	.994	.992	.989	.988	.986	.984	.981	.979	.978	.975
84	1.012	1.009	1.007	1.006	1.004	1.001	1.000	.998	.996	.994	.991	.989	.987	.986	.982	.980	.979	.978
86	1.014	1.012	1.009	1.007	1.005	1.004	1.001	.999	.998	.995	.993	.991	.989	.987	.985	.984	.981	.979
88	1.016	1.014	1.012	1.011	1.008	1.006	1.004	1.001	1.000	.996	.994	.993	.991	.988	.987	.985	.982	.980
90	1.018	1.015	1.014	1.012	1.009	1.008	1.006	1.004	1.001	1.000	.998	.995	.993	.992	.989	.987	.986	.983
92	1.020	1.016	1.015	1.013	1.011	1.009	1.007	1.005	1.002	1.001	.999	.996	.995	.993	.991	.989	.987	.985
94	1.022	1.019	1.018	1.015	1.014	1.012	1.009	1.007	1.006	1.004	1.001	.999	.998	.995	.993	.992	.989	.988
96	1.024	1.021	1.020	1.018	1.015	1.013	1.012	1.008	1.007	1.006	1.004	1.001	.999	.998	.995	.993	.992	.989
98	1.026	1.024	1.022	1.020	1.018	1.015	1.014	1.012	1.008	1.007	1.006	1.004	1.001	1.000	.998	.995	.994	.992
100	1.028	1.026	1.025	1.022	1.020	1.018	1.015	1.013	1.012	1.009	1.008	1.006	1.004	1.001	1.000	.998	.995	.994
102	1.031	1.028	1.026	1.025	1.022	1.020	1.018	1.015	1.014	1.012	1.009	1.008	1.006	1.004	1.001	1.000	.998	.995
104	1.033	1.031	1.028	1.026	1.025	1.022	1.020	1.018	1.015	1.014	1.012	1.009	1.008	1.006	1.004	1.001	1.000	.998
106	1.035	1.032	1.031	1.028	1.026	1.025	1.022	1.019	1.018	1.016	1.014	1.012	1.009	1.008	1.006	1.004	1.002	1.000
108	1.036	1.034	1.032	1.031	1.028	1.026	1.024	1.022	1.020	1.018	1.016	1.014	1.012	1.011	1.008	1.006	1.004	1.002
110	1.039	1.036	1.035	1.033	1.031	1.028	1.026	1.024	1.022	1.020	1.019	1.016	1.014	1.012	1.011	1.008	1.006	1.004
112	1.041	1.039	1.036	1.035	1.033	1.031	1.028	1.026	1.025	1.022	1.020	1.019	1.016	1.014	1.012	1.011	1.008	1.006
114	1.044	1.041	1.039	1.036	1.035	1.033	1.031	1.028	1.026	1.025	1.022	1.020	1.019	1.016	1.014	1.012	1.011	1.008
116	1.046	1.042	1.041	1.039	1.036	1.035	1.033	1.029	1.028	1.026	1.025	1.022	1.020	1.019	1.016	1.014	1.012	1.011
118	1.047	1.045	1.044	1.041	1.039	1.036	1.035	1.032	1.031	1.028	1.027	1.025	1.022	1.020	1.019	1.016	1.014	1.012
120	1.049	1.047	1.045	1.044	1.041	1.039	1.036	1.034	1.033	1.031	1.028	1.027	1.025	1.022	1.020	1.019	1.016	1.014
122	1.052	1.049	1.047	1.046	1.044	1.041	1.039	1.036	1.035	1.033	1.031	1.028	1.027	1.025	1.022	1.020	1.019	1.016
124	1.054	1.052	1.049	1.047	1.046	1.044	1.041	1.039	1.036	1.035	1.033	1.031	1.028	1.027	1.025	1.022	1.020	1.019
126	1.056	1.053	1.052	1.049	1.047	1.046	1.044	1.040	1.039	1.036	1.035	1.033	1.031	1.028	1.027	1.025	1.024	1.021
128	1.058	1.057	1.054	1.052	1.049	1.047	1.045	1.042	1.041	1.039	1.036	1.034	1.033	1.031	1.028	1.026	1.025	1.022
130	1.060	1.058	1.056	1.054	1.052	1.049	1.047	1.045	1.044	1.041	1.039	1.036	1.035	1.033	1.031	1.028	1.027	1.025
132	1.062	1.060	1.058	1.055	1.054	1.052	1.049	1.047	1.045	1.044	1.041	1.039	1.036	1.035	1.033	1.031	1.028	1.027
134	1.065	1.062	1.060	1.058	1.055	1.054	1.052	1.048	1.047	1.045	1.044	1.041	1.039	1.036	1.035	1.033	1.031	1.028
136	1.067	1.064	1.062	1.060	1.058	1.055	1.054	1.051	1.049	1.047	1.045	1.044	1.041	1.039	1.036	1.034	1.033	1.031
138	1.069	1.066	1.065	1.062	1.060	1.058	1.055	1.053	1.052	1.049	1.047	1.046	1.044	1.041	1.039	1.036	1.035	1.033
140	1.071	1.068	1.066	1.065	1.062	1.060	1.058	1.055	1.053	1.052	1.049	1.047	1.045	1.044	1.041	1.039	1.036	1.035

desired manual setting of the distributor head. For automatic control, distributor head is clamped at required initial timing.

DETONATION.—Three arbitrary grades of intensity are used. 1. None (N); 2. Incipient or borderline (BL), which is that condition where detonation is barely perceptible in the most offensive cylinder; 3. Objectionable (OBJ) which is that condition where intensity is greater than incipiency in any cylinder.

Fuel

FUEL CONSUMPTION INSTRUMENTATION.—Either weight or volumetric method is used.

FUEL TEMPERATURE.—With volumetric method, fuel temperature is measured by a glass-stem thermometer at a point adjacent to the pipettes.

MINIMUM FUEL MEASUREMENT INTERVAL is the same as for speed measurement. Speed and fuel are measured during the same interval.

Computations

Notation.—w = weight of fuel, lb.; v = volume of pipette, c.c.; sp.gr. = specific gravity of fuel at pipette temperature; W = fuel rate, lb. per hr.; t = measuring interval, sec.; D_s, D_o = standard and observed moist-air carburetor density, respectively, lb. per cu. ft.; F_s = specific fuel consumption, lb. per Hp.-hr; Q = torque. lb.-ft.; L = torque scale reading, lb.; n = engine revolutions for w lb. of fuel.

Fuel rate is expressed in lb. per hr. With the volumetric method, weight of fuel in pipette = $w = v \times$ (sp. gr.)/453.6. $W = 7.94\ v \times$ (sp. gr.)$/t$. Specific gravity of commercial gasolines is about 0.74 at 60° F.; the rate of decrease is about 0.00033 per deg. F. increase in temperature. Observed hourly fuel rates may be presented with corrected power data, but should be so designated. The correction factor to standard carburetor air conditions = $\sqrt{D_s/D_O}$. D_s for 60° F. = 0.07581; for 103° F. = 0.07002; for 150° F. = 0.06462. $F_s = W/\text{Hp.} = 315{,}000\ w/Qn$ = (for 21-in. dynamometer torque arm) $180{,}000\ w/Ln$; = (for 15.75-in. torque arm) $240{,}000\ w/Ln$. Observed specific fuel may be presented with corrected power data, but should be so designated. Corrected data may be obtained by correcting the observed rate and dividing by corrected power.

Air Consumption

INSTRUMENTATION.—Standard intake system is used. Air meter is connected to an air-tight box surrounding carburetor. Satisfactory instruments are the gasometer, smooth approach orifices or thin-plate, sharp-edged orifices of the Durley type. Restoration of atmospheric pressure at carburetor may be obtained by a motor-driven blower between orifice tank and carburetor. To reduce pulsations it is customary to use a large-volume surge tank between engine and air meter.

Computations

Notation.— D = density, lb. per cu. ft.; B_C = corrected barometer, in. Hg, absolute; T = temperature, deg. F., absolute; e = water vapor pressure, in. Hg, absolute; V = volume flow, cu. ft. per min.; W = weight rate, lb. per hr.; d = orifice diam., in.; C = orifice discharge coefficient; p = orifice depression, in. of water; A_s = specific air consumption, lb. per Hp.-hr.; E_v = volumetric efficiency; S = engine displacement, cu. in.; N = engine r.p.m.; A/F = air-fuel ratio.

DENSITY.—For dry air, $D = 1.326\ B_C/T$; for moist air $D = 1.326\ (B_C - 0.378e)/T$; weight of dry air in 1 cu. ft. of moist air = $1.326(B_C - e)/T$. Neglecting moisture gives density values higher than actual. Where absolute pressure differs but slightly from barometer, vapor pressure e is assumed equal to atmospheric pressure e.

AIR RATE.—For a displacement meter, volume flow of moist air at meter intake density is obtained directly; weight flow of moist air = $W = 79.56\ V(B_C - 0.378\ e)/T$. This is used to determine volume flow at carburetor density. The weight flow of dry air = $W = 79.56\ V(B_C - e)/T$. For orifice meters, volume flow of moist air = $V = 5.98\ dC\sqrt{p/D}$; using moist-air density at meter intake, $V = 5.195\ d^2C\sqrt{PT/(B_C - 0.378\ e)}$. Weight flow of moist air = $W = 60\ VD$. Computations are simplified by use of charts. Observed data may be presented with corrected power data, if so designated. Corrected data are obtained by correcting observed air rate, using the correction factor for indicated power.

Specific air consumption should be based on dry air rate. If so designated, however, moist air rate may be used. Specific consumption = $A_s = W/\text{Hp}$.

AIR-FUEL RATIO is based on the ratio of weight rates of air and fuel. Dry-air weight rate should be used, but moist-air weight rate may be used. Observed A/F data may be presented with corrected power data if so designated.

VOLUMETRIC EFFICIENCY is based on volume flow of moist air at carburetor moist-air density. E_v (for 4-stroke cycle engine) = $3456\ V/SN$. Observed E_v may be presented with corrected power data if so designated. Corrected E_v = observed $E_v\sqrt{T_s/T}$, where T_s = standard, and T = observed, carburetor temperature.

Temperatures

GENERAL INSTRUMENTATION.—Recommended equipment for measuring important temperatures includes: Mercury or alcohol glass-stem thermometers, thermocouple-potentiometer units, and constant-voltage, electric-resistance thermometers. Vapor or liquid-filled, distance-reading type thermometers are not recommended. Temperatures are measured in deg. F. The thermometer is shielded from radiant heat.

CARBURETOR AIR.—Reading usually is taken immediately inside air cleaner in the air stream, and used to compute power correction factors and volumetric efficiency. Standard car-

buretor air temperatures are 60° F., 103° F., and 150° F. The value 60° F. is retained for comparisons with published data corrected to this datum. The values 150 ° F. and 103° F. represent, respectively, approximate temperatures for V-type engines with carburetor in the V, and other engines. The odd value, 103 ° F. facilitates conversion of corrected data to and from the intermediate datum by adding or subtracting 4% for indicated power and 4.7% for brake power. The datum 103° F. is used for the stock test.

MIXTURE is measured at a representative point in intake manifold.

WATER TEMPERATURE.—Outlet temperature is measured near the point where water leaves the engine. Standard water outlet temperature is 165° F. for power and friction tests.

SUMP OIL preferably is measured at some point between entrance to oil pump suction pipe and the first pressure-fed bearing. For all standard tests under power, an adequate oil cooler is used to limit the temperature to 195 ± 5° F. Standard temperature is 165 ± 5° F. for the motoring friction test. See p. 14–94. For the motoring compression test it is required only that the temperature be above 100° F.

Pressures

GENERAL INSTRUMENTATION.—Glass-tube manometers usually are used for small differential pressures; calibrated Bourdon gages for the higher differentials. Single-leg inclined manometers, with compensated scale, are suitable for more exact measurements. Mercury is used for differentials greater than 1 in. Hg; for smaller differentials, a lighter liquid is used. For mercury, the top of the meniscus is read; for other liquids, the bottom.

SPECIFIC PRESSURES.—Barometer measurement at a central location is sufficient. A high-grade adjustable-reservoir mercury barometer is used. True pressure is obtained by correcting observed readings to standard temperature of 28.5° F. See Fig. 72. The standard for total pressure (dry air plus water vapor) is 29.92 in. Hg.

Carburetor Air Pressure usually equals barometric pressure. Water vapor pressure at carburetor is assumed equal to that measured in the atmosphere. Dry carburetor pressure is used in computing power correction factors; total carburetor pressure (dry air + water vapor) is used in correcting compression and computing moist air density. The standard for total carburetor pressure is 29.92 in. Hg.

Intake Manifold pressure is measured at a representative point in the manifold as determined for each engine, using a mercury U-tube manometer, suitably damped.

Exhaust Manifold pressure is measured at point in exhaust manifold near exhaust pipe flange, using a mercury U-tube manometer, with suitable damping and a water-condensing reservoir.

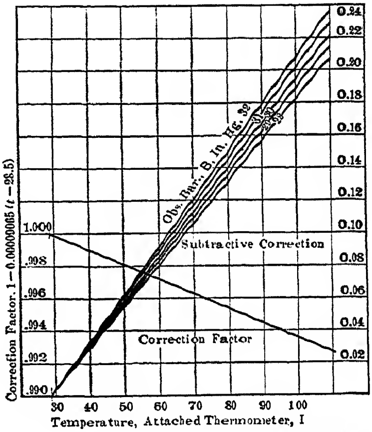


FIG. 72. Barometer Correction for Temperature

Humidity

INSTRUMENTATION.—A stationary wet-and-dry-bulb psychrometer, having a fixed-position fan giving a ventilation velocity of 10 ft. per sec., or more, past the wet bulb, is recommended.

COMPUTATIONS.—Water vapor pressure may be determined from the psychrometric chart (see p. 11–52) and equations on p. 1–08, using observed psychrometer temperatures. Vapor pressure is used in computing air density and power correction factors.

Water Heat Rejection and Circulation Rate

INSTRUMENTATION.—Direct cooling is required. The recommended set-up includes: Lagged water piping between engine and a lagged water reservoir, the latter suitably baffled to intimately mix make-up and circulation water; open-scale thermometers to measure water out, water in, make-up and overflow water temperatures; make-up water line leading to the reservoir, fitted with a fine control valve; overflow pipe for leading overflow water either to weighing-pan or drain; weighing-pan and scale. The *temperature method* (see below) of determining circulation rate is recommended.

Computations

Notation.—Heat rejection is expressed in B.t.u. per min.; specific heat rejection, in B.t.u. per Hp.-min.; w = overflow water rate, lb. per min.; t_1 and t_2 = respectively, overflow and make-up water temperatures, deg. F.; t_3 and t_4 = respectively, engine water out and in temperatures, deg. F.

Water heat rejection = $w\,(t_1 - t_2)$; specific heat rejection = $w\,(t_1 - t_2)/\text{B.Hp.}$ observed; flow rate through engine = lb. per min. = $w\,(t_1 - t_2)/(t_3 - t_4)$. Fuel heat is based on lower heating value = 18,500 B.t.u. per lb. 1 Hp. = 42.42 B.t.u. per min. When overflow water is measured by volume, density of water is considered. Pure water weighs 62.428 lb. per cu. ft. at 39.3° F.; 1 gal. weighs 8.345 lb. at 39.3° F.; 8.153 lb. at 160° F.; 7.995 lb. at 212° F. Observed data on

water heat may be presented with corrected power data when so designated. Heat removed by an external oil cooler is classified under radiation and exhaust losses.

Motoring Friction

TEST CONDITIONS. — Motoring the engine by an electric dynanometer and measuring input power determines friction power. Motoring friction tests are made with: 1. Full throttle; 2. Dry carburetor; 3. Muffler and tailpipe off; 4. Water and oil temperatures at 165 ± 5° F.

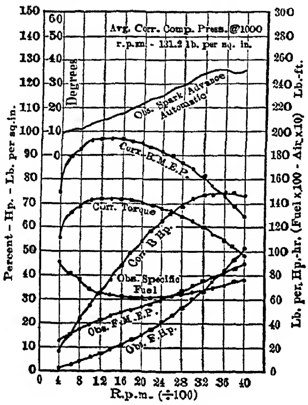


FIG. 73. Power and Economy Test Curves. Full Throttle

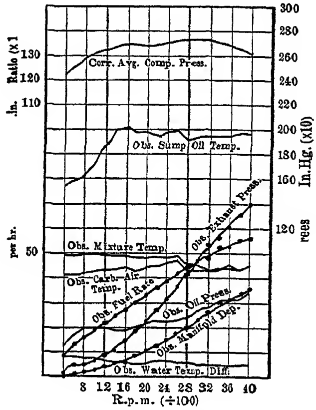


FIG. 74. Standard Engine Test, Full Throttle, Temperatures and Pressures

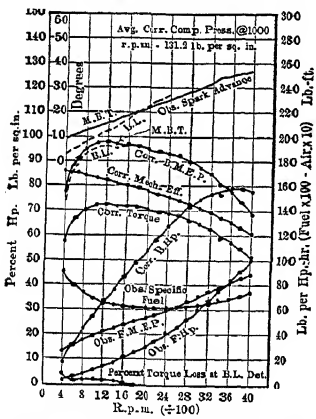


FIG. 75. Standard Engine Test (No. 2) Maximum Power and Economy

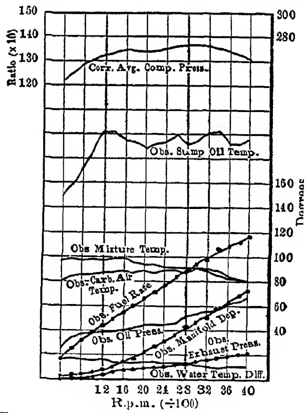


FIG. 76. Standard Engine Test (No. 2) Maximum Power and Detonation

Required temperatures are maintained by operating under power for the interval needed to restore them. Test is made in continuous progression, independent of power test, from 400 r.p.m. to limiting speed in 200-r.p.m. increments.

Motoring Compression

INSTRUMENTATION.—A suitable compression gage, reading maximum pressure, is recommended. Check valve is located, if possible, flush with inside wall of cylinder.

TEST CONDITIONS. — Motoring tests are made with: 1. Full throttle; 2. Dry carburetor; 3. Water and oil temperatures above 100° F.; 4. Muffler off; 5. Ignition off. When measurement is made at one speed, 1000 r.p.m. is standard.

Computations

Notation.—P_c = total carburetor pressure (dry air + vapor pressure); B_c = true barometric pressure, in. Hg. Standard datum pressure for compression is 29.92 in. Hg. Correction factor = $29.92/P_c$. Usually $P_c = B_c$.

TEST 1. FULL THROTTLE AS INSTALLED. Purpose.—This test gives representative full throttle performance over the speed range of engine as installed in car. It should not be used for purposes other than this. See Figs. 73, 74.

Procedure.—Stabilize on 400 r.p.m. and make test run; proceed, making test runs under stabilized conditions every 200 r.p.m. to limiting speed.

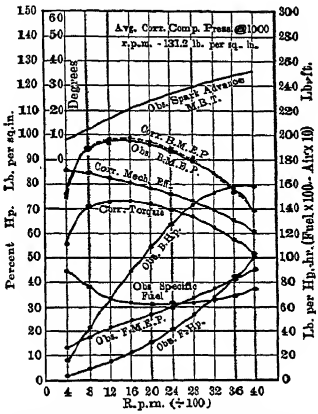


Fig. 77. Standard Engine Test (No. 3) Power and Economy

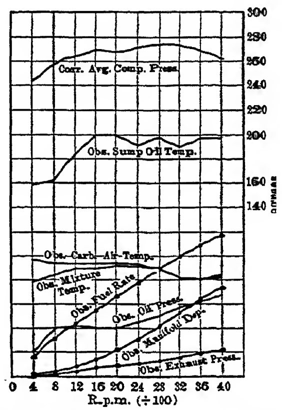


Fig. 78. Standard Engine Test (No. 3) Temperatures and Pressures

Engine Equipment, Adjustments and Settings.—Muffler and tailpipe on; throttle, full; carburetor, fixed standard or adjustment *c*; see p. 14–86; ignition, automatic, adjustment as recommended by manufacturer, using and recording standard initial setting.

TEST 2. MAXIMUM POWER AND DETONATION. Purpose.—This test is used as a check on fundamental design. It gives both maximum power performance and engine detonation charactertistics. See Figs. 75, 76.

Procedure.—Stabilize on 400 r.p.m. with spark retarded to eliminate detonation; advance spark to BL detonation (see p. 14–92) and note spark timing; advance 2° or until a definite increase in detonation intensity occurs; reset on original BL timing, and if original impressions are checked, record timing and torque. Next adjust spark timing to MBT as outlined in Test 3; stabilize and make complete test run; repeat procedure at 200-r.p.m. increments to that speed where overadvance for BL detonation causes 1% torque loss from MBT spark; at higher speeds make test at MBT spark only.

Engine Equipment, Adjustments and Settings.—Standard exhaust pipe, muffler and tailpipe off; throttle, full; carburetor, fixed standard or adjustment *c*; ignition, manual; adjustments as noted above, or adjustments *b* and *c*.

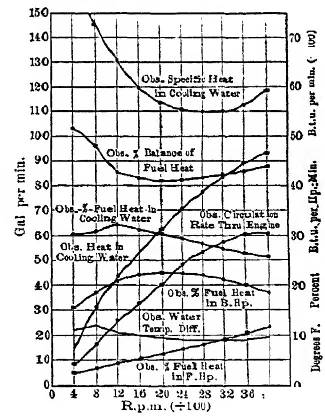


Fig. 79. Standard Engine Test (No. 3) Heat Distribution

TEST 3. HEAT DISTRIBUTION. Purpose.—These tests involve making: 1. A water heat-rejection test; 2. A motoring friction test. As it usually is of no interest to separate exhaust, radiation and oil heat, fuel heat need be divided into only: 1. Brake power; 2. Friction power; 3. Heat to cooling water; 4. Balance. The rate of water circulation through engine is determined from water heat-rejection data. In a heat-rejection test to ascertain radiator requirements in the car, the standard engine exhaust system is used, and also a standard radiator to simulate water flow restriction. See Figs. 77, 78, 79.

Procedure.—Stabilize at 400 r.p.m. using MBT spark setting; make complete test run and proceed in 400-r.p.m. increments to limiting speed.

Engine Equipment, Adjustments and Settings.—Standard exhaust pipe, muffler and tailpipe off; throttle, full; carburetor, fixed standard or adjustment *c*; ignition, manual, adjustment *b*.

TEST 4. VOLUMETRIC EFFICIENCY. Purpose.—This test, used to determine full-throttle volumetric efficiency, is made separately. Power and economy are of secondary importance, but all engine data are recorded with the air consumption data. See Figs. 80, 81.

Procedure depends on type of air measuring equipment used.

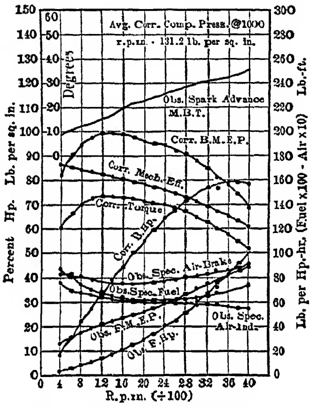


Fig. 80. Standard Engine Test (No. 4) Power and Economy. Volumetric Efficiency

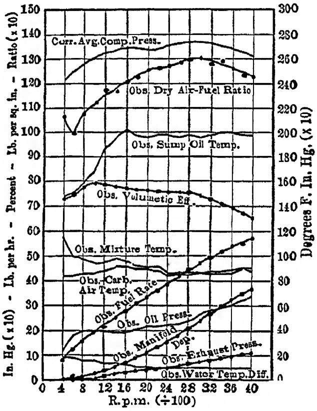


Fig. 81. Standard Engine Test (No. 4) Volumetric Efficiency. Temperatures and Pressures

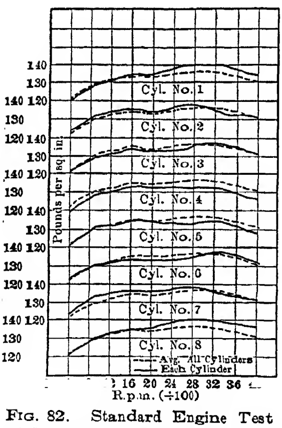


Fig. 82. Standard Engine Test (No. 6) Motoring Compression

Engine Equipment, Adjustments and Settings.—Standard exhaust pipe, muffler and tailpipe off; throttle, full; carburetor, fixed standard or adjustment *c*; ignition, manual, adjustment *b*.

TEST 5. MOTORING FRICTION. Purpose is to obtain an estimate of engine friction loss. Comparative results obtained by several methods justify use of full throttle motoring method. See Friction Hp. curves, Figs. 73, 75, 77, 80.

Procedure.—Test is made in 200-r.p.m. increments from 400 r.p.m. to limiting speed.

Engine Equipment, Adjustments and Settings. — Standard exhaust pipe, muffler and tailpipe off; throttle, full; carburetor, fuel off; ignition, off.

TEST 6. MOTORING COMPRESSION. Purpose is to show relative air charge distribution among cylinders and to indicate ram effects. See Fig. 82.

Procedure.—With compression gage in first cylinder, motor at 400 r.p.m. and read gage; proceed in 200-r.p.m. increments to limiting speed without interruption; repeat for all cylinders; make check test on first cylinder.

Engine Equipment, Adjustments and Settings.—Standard exhaust pipe, muffler and tailpipe off; throttle, full; carburetor, fuel off; ignition, off.

TEST 7. ROAD POWER AS INSTALLED. Purpose is to obtain fuel economy of engine as installed in car. Standard road power curves (see p. 14–86) are used and torque is adjusted for existing conditions. Interest in fuel consumption at high road speeds make test desirable at engine speeds approaching maximum car speed. See Figs. 83, 84.

Procedure.—Stabilize on 400 r.p.m. with corresponding standard road torque and make complete test run; proceed in 200-r.p.m. increments, with corresponding loads, to limiting speed.

Engine Equipment, Adjustments and Settings.—Standard exhaust pipe, muffler and tailpipe on; throttle, adjusted at each speed to standard road power; carburetor, fixed standard or adjustment *c*; ignition, adjustment *a*, or manual adjustment *b*, for engines with non-automatic distributors.

Power Correction Factors for Pressure, Humidity and Temperature

Table 13 gives factors for application to indicated power, for the standard datum temperature of 103° F. Table 14 gives similar factors for full throttle brake power based on an assumed average mechanical efficiency of 85%. Fig. 71 extends the range of pressures and temperatures of the tables.

Tables and charts are based on: 1. Datum pressure (total), 29.92 in. Hg; 2. Datum water vapor pressure = 0.5 in. Hg; 3. Datum temperatures of 60°, 103° and 150° F.; 103° F. is datum temperature for stock engine test. Datum air pressure thus is (29.92 – 0.5) = 29.42 in. Hg.

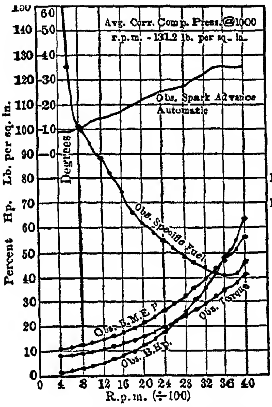


FIG. 83. Standard Engine Test (No. 7) Power and Economy. Road Power as Installed

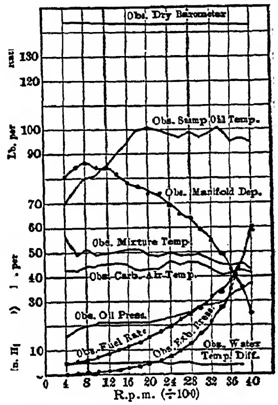


FIG. 84. Standard Engine Test (No. 7) Road Power

The combined correction factor is based on the assumption that indicated power varies; *a*, proportionally with dry air pressure at carburetor; *b*, inversely as the square root of absolute carburetor air temperature. The formula for the indicated power correction factor is:

$$F_{ci} = \frac{29.42}{\text{Dry carburetor press., in. Hg}} \times \sqrt{\frac{\text{Observed carburetor temp., deg. F., abs.}}{\text{Datum carburetor temp., deg. F., abs.}}}$$

Correction factors for full throttle brake power are based on the assumptions: *a*, no change in motoring friction when passing from observed to corrected air conditions; *b*, an average mechanical efficiency of 85%. The formula for brake power correction factor is: $F_{cb} = (F_{ci} + E_m \text{ observed} - 1)/E_m$ observed. Substituting 0.85 for E_m observed, $F_{cb} = (F_{ci} - 0.15)/0.85$. E_m = mechanical efficiency.

AERONAUTICS

By Edward P. Warner and S. Paul Johnston

DEFINITIONS AND CLASSIFICATIONS.—Machines for aerial navigation fall into two general categories.

Lighter-than-air craft are those which displace a mass of air whose weight is greater than their total weight, and which, therefore, can float in the atmosphere buoyed up by aerostatic forces.

Heavier-than-air craft do not displace enough air to achieve buoyancy, but derive their lift by the relative motion of air passing over appropriately designed surfaces. The sustaining forces are aerodynamic.

FORMS OF AIR CRAFT.—Lighter-than-air craft assume two general forms, *viz.*, balloons and airships.

Balloons consist of containers for a suitable gas whose density is less than that of the atmosphere (heated air, hydrogen, helium, etc.), are usually without means of directional control, and always without means for mechanical propulsion.

Airships differ from balloons in that they have means for mechanical propulsion and apparatus for directional control. They consist of single or multiple containers for the supporting gases and the necessary structures to house the personnel, and to support the power plants and controlling surfaces. Structurally, they fall into three classes:

1. *Non-rigid*, in which the gas container maintains its shape solely by reason of the internal gas pressure; 2. *Semi-rigid*, in which the shape of the gas container is partially maintained by means of a structural keel, or backbone; 3. *Rigid*, in which appropriate structural members completely surround and are independent of the gas containers.

Heavier-than-air Craft, so far given practical acceptance, divide into 4 general classes: 1. ***Airplanes***, fixed-wing, mechanically-driven aircraft, supported by the dynamic action of the air. 2. *Gliders*, airplanes without power plants, which utilize the forces of gravity to produce the required propulsive action. 3. *Helicopters*, machines designed to ascend vertically, due to the upward thrust of power-driven rotating airfoils or propellers. 4. *Autogiros*, in which the fixed wings of the airplane are replaced by rotating airfoils revolving under aerodynamic forces only. In spite of a superficial resemblance, an autogiro is not a helicopter, as the engine is not directly connected to the rotor when in flight.

AIRPLANES

1. CLASSIFICATION OF AIRPLANES

Airplanes consist of 5 principal elements which may be combined in a variety of ways. These are: 1. Main supporting surfaces (wings). 2. Auxiliary surfaces for stabilization and control (horizontal stabilizer, fin elevators, rudders and ailerons). 3. Housing for the personnel, power-plants and cargo (body, fuselage, nacelles, or hull). 4. Undercarriage or landing gear (wheels, tail skids, floats or hull). 5. Power-plant (engines, propellers, fuel, oil, cooling system and controls). Airplanes may be classified under four major considerations, all of which are inter-related: *a.* Structural arrangement. *b.* The nature of their terrestrial base of operations. *c.* Materials used. *d.* Power-plant arrangement.

CLASSIFICATION BY STRUCTURAL ARRANGEMENT.—Airplanes are either **monoplanes, biplanes,** or **triplanes**, depending upon whether they have one, two, or three main supporting surfaces. Biplanes in which the area of one wing is very small as compared with the other sometimes are referred to as **sesqui-planes.** Almost without exception the ailerons, or lateral control surfaces are mounted at or near the tips of the main wings, and the other auxiliary surfaces are combined into a single tail unit or *empennage*, supported some distance aft of the main wing. Tailless and "tail-first," or *canard*, designs so far have had limited practical use. With respect to landing gear, airplanes are grouped as landplanes, seaplanes, or amphibians.

Landplanes are fitted with an undercarriage, usually of three wheels, connected to the body through suitable shock-absorbing devices. Water types take two practical forms:

the float seaplane, essentially a land type in which the wheels of the undercarriage are replaced by one or more suitably designed floats, and the **flying boat**, in which the main body of the airplane is a water-tight hull, with a bottom suitably shaped to give good take-off and landing characteristics, which not only encloses the personnel but furnishes points of attachment for wings, auxiliary surfaces and power plants. An **amphibian** is a composite type designed to operate either from water or land.

Monoplanes may be of the parasol, the high-, the mid-, or the low-wing type, depending on whether the wing is carried on struts above the fuselage or is attached to the upper, the middle, or the lower part of the fuselage. Also, these wings may be of the *full cantilever* variety, in which the structure is totally enclosed within the wing contour, or they may be either *strut-* or *wire-braced* to some suitable part of the fuselage or undercarriage.

Biplanes may be either *orthogonal*, or *staggered*, depending on whether the top wing is placed directly over the bottom, or whether it is located forward or aft of the bottom wing. Either monoplane or biplane wings may be tapered in section or in plan or in both, or they may be swept back, that is, they may make an angle rearward with respect to the plane of symmetry of the craft of less than 90°.

CLASSIFICATION BY MATERIALS.—The principal structural materials used in aircraft are: wood, chiefly spruce; fabrics, chiefly cotton; and metals, chiefly steel and aluminum-alloy. For many years *composite* construction has been employed, using wood, fabrics and metals in various combinations. For military and transport aircraft, the all-metal machine is now (1935) rapidly replacing the composite.

CLASSIFICATION BY POWER PLANT ARRANGEMENT.—Modern aircraft are propelled by 2-, 3-, or 4-bladed wooden or metal propellers driven by air- or water-cooled internal combustion engines. Propellers may be arranged either as *tractors* or as *pushers*, depending on whether the propeller is ahead of or behind the engine. Airplanes may be equipped with single or multiple power plants, arranged either as pusher or tractor, or in combinations of both. Power plants may be mounted within the fuselage proper, or may be housed in nacelles supported by the fuselage and mounted either directly in or above or below or between the wing or wings. There is a tendency, as the overall dimensions of airplanes increase, to enclose engines completely within wing structures to reduce the parasite drag.

2. AERODYNAMICS

THE ATMOSPHERE.—The layer of air which surrounds the earth is a non-homogeneous fluid whose density varies inversely with the distance above sea-level. The density at 20,000 ft. is approximately one-half and at 40,000 ft. about one-quarter the sea-level density. As with any gas, a very definite relationship exists among temperature, pressure and density. The absolute values may vary considerably in any given locality, due to local meteorological conditions. To have a basis for comparison of airplane performance or calibration of instruments, a purely fictitious *standard atmosphere* has been assumed, roughly corresponding to average conditions, and defined by known altitude-temperature-pressure relations. See The Standard Atmosphere, W. S. Diehl, Nat. Advisory Comm. for Aeronautics Report No. 218. In selecting bases for the standard atmosphere, international standards have been followed. The basic data are: Standard sea-level pressure, p_0, $= 29.921$ in. Hg (2116 lb. per sq. ft.); standard temperature, $t_0 = 59.0°$ F.; standard specific weight of air, $g\rho = 0.07651$ lb. per cu. ft.; standard temperature gradient, $a = 0.003566°$ F. per ft. of altitude (1° F. per 280 ft.).

Based on the above assumption, properties of the standard atmosphere at intervals up to 25,000 ft. are given in Table 1. Up to an altitude h of about 35,000 ft. the density ratio is given correctly within $2\ ^1/_2\%$ by the formula

$$\rho/\rho_0 = 1 - 0.03\,[(h/1000) - (h/10{,}000)^2] \qquad [1]$$

FLUID RESISTANCE.—Since atmospheric air is not a perfect fluid, any solid body passing through it at any speed is opposed by a certain fluid resistance, which depends

Table 1. Standard Atmosphere—English Units

Altitude, h, ft.	t, deg. F.	$\frac{p}{p_0}$	$\frac{\rho}{\rho_0}$	p, in. Hg.	$g\rho$, lb. per cu. ft.	Altitude, h, ft.	t, deg. F.	$\frac{p}{p_0}$	$\frac{\rho}{\rho_0}$	p, in. Hg.	$g\rho$, lb. per cu. ft.
0	59.000	1.0000	1.0000	29.92	0.07651	12,000	16.206	0.6359	0.6931	19.03	0.05303
2,000	51.868	0.9298	0.9428	27.82	.07213	14,000	9.074	.5873	.6499	17.57	.04973
4,000	44.735	.8636	.8881	25.84	.06794	16,000	1.941	.5418	.6088	16.21	.04658
6,000	37.603	.8013	.8358	23.98	.06395	18,000	− 5.191	.4992	.5698	14.94	.04359
8,000	30.471	.7427	.7859	22.22	.06013	20,000	− 12.323	.4594	.5327	13.74	.04075
10,000	23.338	.6876	.7384	20.58	.05649	25,000	− 30.154	.3709	.4480	11.10	.03427

on the density of the fluid, the dimensions and form of the body, and its velocity. That is

$$R = C_R(\rho/2)SV^2 \quad . \quad . \quad . \quad . \quad . \quad . \quad [2]$$

where R = total resistance or resultant force; C_R = a coefficient depending on the form of the body and its attitude with respect to the motion of the fluid (nearly enough constant for most purposes, though to be exact C_R varies somewhat with the absolute dimensions of the body and the relative velocity between body and fluid); ρ = density of air ($\rho/2$ is used in the equation, because $(\rho/2)V^2$ has a physical significance as the pressure which results from a complete conversion of the kinetic energy of moving air into its pressure equivalent); S = projected area of body; V = velocity of motion. See also discussion of coefficients in formulas [3] and [4].

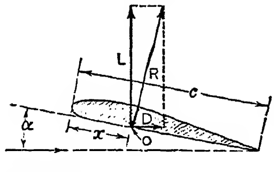


Fig. 1. Resolution of Aerodynamic Forces Acting on a Typical Airfoil

AIRFOILS are winglike surfaces, with cross-sections designed to develop a useful dynamic reaction when relative motion is set up between them and the surrounding air. Fig. 1 is a typical airfoil cross-section, and indicates the location and direction of the resultant force. For a given angle of attack α, the resultant R intersects the chord c of the airfoil at the center of pressure O at a distance x from the leading edge. For convenience, it is customary to resolve the resultant force into the lift component L, and the drag D, respectively normal and parallel to the relative wind. Since the components of any force vary in accordance with the same laws that govern the force itself, fundamental equations may be written for the lift and drag, *viz.*:

$$L = L_c SV^2 \quad . \quad . \quad . \quad . \quad . \quad [3]; \qquad D = D_c SV^2 \quad . \quad . \quad . \quad . \quad . \quad [4]$$

where L_c and D_c are lift and drag coefficients, and are functions of the angle of attack α.

The coefficients of the above formulas are of the so-called engineering type, and yield values of lift or drag in pounds when S is in square feet, and V is in miles per hour. To give dimensionally correct equations, the density factors must be introduced to put the coefficients in the so-called absolute form. The absolute coefficients of lift and drag C_L and C_D, adopted by the N.A.C.A. are designed to fit into the relations

$$L = C_L(\rho SV^2/2) \quad . \quad . \quad . \quad [5]; \qquad D = C_D(\rho SV^2/2) \quad . \quad . \quad . \quad [6]$$

The relation between the engineering and N.A.C.A. absolute coefficients, therefore, is

Engineering		N.A.C.A Absolute	
$L_c = L/SV^2$	or	$C_L = 2L/\rho SV^2$	[7]
$D_c = D/SV^2$	or	$C_D = 2D/\rho SV^2$	[8]

ρ must be in mass units. In foot-second units $\rho/2 = 0.00118$ under standard sea level conditions. To transpose coefficients from N.A.C.A. absolute units to engineering units, multiply by 0.00255; to change engineering coefficients to the absolute form, multiply by 392.

In dealing with lift and drag coefficients of wings, the area S involved is the area of the wing in plan; fundamentally, the product of span and chord, sq. ft. In applying these

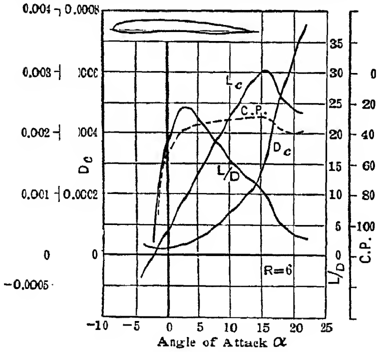


Fig. 2. Characteristics of a Relatively Thin Airfoil with Concave Lower Surface

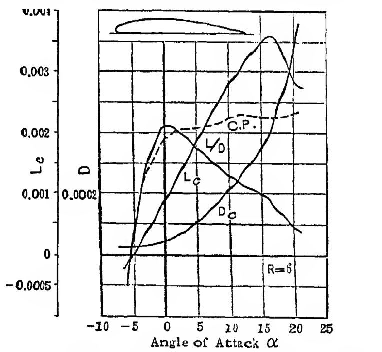


Fig. 3. Characteristics of Medium Thick Airfoil with Flat Lower Surface

equations to the calculation of parasite resistance of solid bodies, the area considered is the area of the body projected on a plane normal to the direction of relative motion.

Center of Pressure Travel and Moment Coefficient.—The location of the vector of resultant force on an airfoil can be indicated either by specifying the distance from the leading edge to the intersection of the vector with the chord (called the center of pressure), or by giving the moment of the resultant force about some convenient axis. The first method has the clearer physical significance and is valuable for purposes of illustration and instruction; the second is more convenient for calculation. The usual moment axis is a point on the chord 25% of its length back from the leading edge, as the moment about that point for a given airfoil is substantially independent of the angle of attack.

The location of the center of pressure on an airfoil is a function of its section, planform and angle of attack. For most common airfoils, as the angle of attack is increased from zero to the angle of maximum lift, the center of pressure tends to move forward from a point about 50% of the chord to a point from 30 to 35% of the chord from the leading edge. Decreasing the angle of attack to very small values causes a very rapid rearward movement of the center of pressure for most airfoils. In fact, for angles within two or three degrees of that where the lift becomes zero, the center of pressure actually may pass behind the leading edge. As the angle decreases, the center of pressure moves to make it decrease still more. Thus most airfoils are markedly unstable in pitch, and the addition of some form of auxiliary surface, usually a tail, is necessary to maintain equilibrium in flight. Airfoils whose bottom surfaces are substantially convex, and those in which the trailing edge is slightly reflexed upward, ordinarily exhibit a lower rate of center of pressure travel than non-symmetrical, positively-cambered sections. The rate of center of pressure travel depends principally on the angle which a line bisecting the angle between the upper and lower surfaces of the airfoil at the trailing edge makes with the chord. When this angle is zero, the center of pressure of that particular section will be found very nearly in a constant position for all angles of attack.

The center of pressure coefficient C_p for an airfoil is the ratio of the distance from the leading edge to center of pressure to the chord length, or

$$C_p = x/c \quad . \quad . \quad . \quad . \quad . \quad . \quad . \quad . \quad . \quad . \quad . \quad . \qquad [9]$$

Interpreting the foregoing in terms of moments, an airfoil will be found to be stable when the moment about a point 25% of the distance back on the chord is positive (of such sign as to tend to increase the angle of attack); unstable when that moment is negative. The larger the negative value of the moment, the more unstable the airfoil becomes and the more tail surface will be required to stabilize it.

Selection of Airfoils.—A vast number of airfoil sections have been tested in aerodynamic laboratories, but only a few of them are in common use. A few of the best sections in 1935 design practice are given and their properties plotted and tabulated in Figs. 2 to 5. In selecting an airfoil for a particular purpose, the following properties are, in general, and for various reasons, sought by the designers: 1. High maximum lift coefficient. 2. Low drag for the values of the lift coefficient that cover the normal working range of the aircraft, generally from 0.0003 to 0.0005 for maximum speed, from 0.0004 to 0.0007 for normal cruising, from 0.0012 to 0.0018 for best take-off and quickest climb.

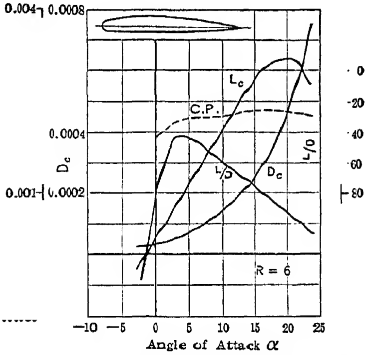


FIG. 4. Characteristics of Medium Thick Airfoil with Convex Lower Surface

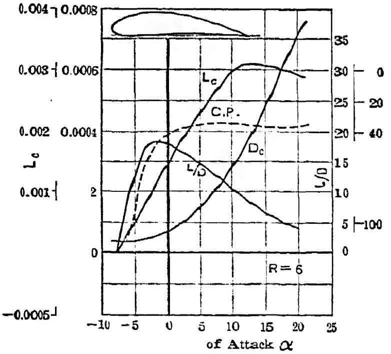


FIG. 5. Characteristics of Relatively Thick Airfoil with Slightly Concave Lower Surface

3. Minimum center of pressure travel. 4. Suitable structural characteristics, *i.e.*, thick enough to permit installation of deep spars.

Pressure Distribution Around Airfoil.—The distribution of pressure along the chord of an airfoil varies with the shape of the section, the angle of attack, and the position of the section in the span. A typical distribution for an average airfoil of medium thickness toward the center of a span of rectangular planform is shown in Fig. 6. The curves represent the condition at approximately 0° and at +15° angle of attack. Typical pressure distributions along the span of rectangular and tapered airfoils are shown in Fig. 7. Note that for angles within the ordinary flying range the span-loading is approximately elliptical. At high angles of attack, unusually high pressures tend to develop around the tips of a rectangular wing, which is undesirable from a structural standpoint. The average effects of tapering a wing in plan are indicated by the dotted lines in Fig. 7.

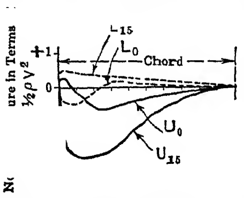


FIG. 6. Typical Pressure Distribution along Chord of Typical Airfoil at Low and High Angles of Attack

The above indicates that the shape of the wing tip influences the pressure distribution of the wing. Rectangular, or positively raked, *i.e.*, leading edge shorter than trailing edge, wings are to be avoided. Negative rake, *i.e.*, leading edge longer than trailing edge, shows a decided improvement in this respect. A distorted semi-elliptical tip, as shown in Fig. 8, is favored in 1935 designs.

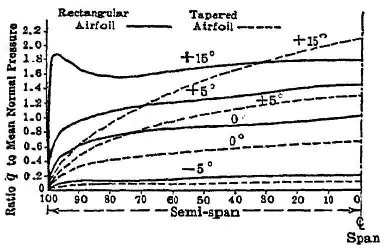


FIG. 7. Pressure Distribution along the Span of Typical Airfoils, Rectangular and Tapered

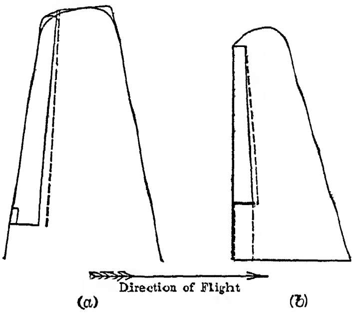


FIG. 8. Modern Wing Tip Shapes. (*a*) Martin B-10 Bomber. (*b*) Douglas DC-2 Transport

INDUCED DRAG AND ASPECT RATIO CORRECTION.—For ordinary engineering purposes, lift and drag are referred to the direction of the relative wind at some point well outside the zone of influence of the airfoil. See Fig. 1. Actually the lines of flow immediately surrounding the airfoil are deflected downward. Thus a more exact axis of reference is found by bisecting the angle at which the air flows on to the section and the angle at which it leaves, as in Fig. 9. The components of the resultant force, taken with respect to the mean relative wind, are called the *profile lift*, L_O, and the *profile drag*, D_O. In this case the ordinary lift and drag components are indicated by L_a and D_a, and may be found by taking components of the profile lift and drag parallel to the ordinary relative wind direction. Since profile drag is much smaller than profile lift, its component parallel to the lift axis may be neglected, and the profile lift may be assumed equal to the lift as ordinarily measured. The component of the profile lift along the drag axis, however, may have considerable magnitude, and is called the *induced drag*. The total drag thus is made up of two elements, profile drag and induced drag. Profile drag only depends on the shape of the section and its angle of attack. The induced drag, however, is purely a function of the *aspect ratio*, *i.e.*, ratio of span to chord, and the lift coefficient. It is independent of the individual section considered. It can be shown that the coefficient of induced drag D_{ci} may be expressed in engineering units by

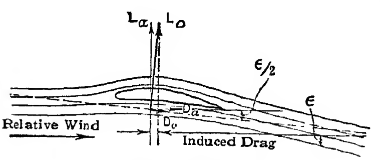


FIG. 9. Graphical Representation of Induced Drag

$$D_{ci} = 125\, L_c^2/R \quad [10]$$

corresponding relation in N.A.C.A. absolute units is

. . . . [11]

where R = aspect ratio. Thus the total drag coefficient for a given lift coefficient steadily decreases with increasing aspect ratio.

ASPECT RATIO, as usually defined, is simply span divided by chord. For tapered, or other irregular planforms, this definition cannot apply. A more general relation is

$$R = b^2/S \qquad [12]$$

where R = aspect ratio; b = extreme span, ft.; S = area, sq. ft.

From a standpoint of aerodynamic efficiency alone, high values of R are to be preferred. The average range of aspect ratio for 1935 airplanes is: Powered monoplanes, 5 to 7 1/2; powered biplanes, 6 to 9; soaring gliders, −13 to 19. The maximum L/D generally increases by about 8% of its value at R = 6 for each unit by which the aspect ratio is increased.

BIPLANES.—Due to interference effects, the efficiency of a combination of two or more airfoils working in close proximity is not as great as that of either of the individual airfoils as a monoplane. Fig. 10 shows a typical case for an unstaggered biplane of equal spans and equal chords with a gap-chord ratio of unity. The general effect of putting two wings together as a biplane is qualitatively the same as reducing the aspect ratio of a monoplane wing, and a similar theory can be applied. The basic biplane, as compared with the monoplane, shows a loss of maximum lift from 3 to 9%; a reduction in maximum L/D of from 15 to 25%; and an increase in minimum drag of from 7 to 20%. Based on the induced drag theory, it can be shown that the relation between biplane aspect ratios (where gap/chord = 1) and the equivalent monoplane aspect ratios are approximately:

Biplane Aspect Ratio	5	6	7	8	10
Equivalent Monoplane Aspect Ratio	3.4	3.9	4.5	5.0	6.1

For a gap-chord ratio of 0.6, reduce the monoplane aspect ratios approximately 8%; for a gap-chord ratio of 1.4, increase equivalent monoplane aspect ratios by approximately 6%.

Even though dimensionally and geometrically similar, the two wings of a biplane generally carry unequal portions of the total load. Except at very low angles of attack, or for combinations with negative stagger (upper wing set behind the lower), the upper wing usually carries the major portion. For unstaggered cellules over the greater part of the normal range of lift coefficients, the ratio of unit load on the upper wing to that on the lower wing is approximately 1.1. With 30° positive stagger the ratio may go as high as 1.6 at low angles of attack, scaling down to approximately 1.2 as the condition of maximum lift is approached. For accurate calculation the relative efficiency must be corrected for span and chord ratios, gap ratios, and stagger.

SLOTS AND FLAPS.—The two devices most commonly used to increase the speed range of airplanes are: 1. Trailing-edge flaps. 2. Fixed or variable slots in the leading edge. 3. Combinations of both.

The general effect of moving trailing-edge flaps downward is to increase the concavity of

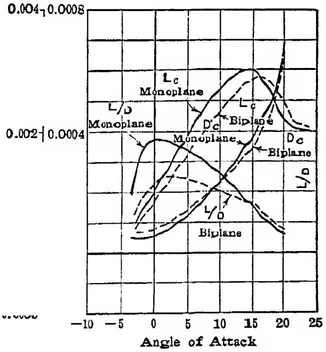


FIG. 10. Comparative Characteristics of a Typical Airfoil as a Monoplane and as a Biplane without Stagger with Gap/Chord = 1 (Clark Y — A.R. = 6)

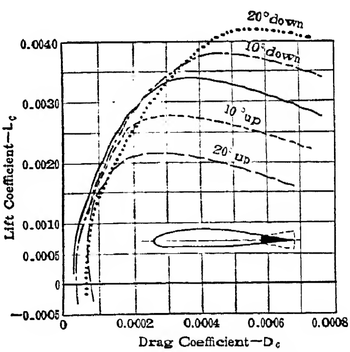
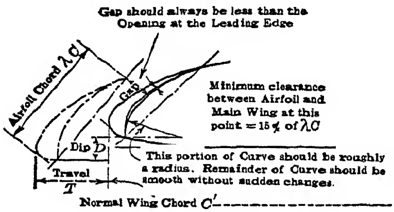


FIG. 11. Effect of Trailing Edge Flap on Lift and Drag. (N.A.C.A. M–6 Airfoil with Flap Chord 20% of Total Wing Chord)

the lower surface of the section and thus to increase the lift. Using trailing edge flaps whose chord is approximately 20 to 25% of the chord of the basic airfoil, monoplane airfoils show an increase in maximum lift to about 40% with a flap setting of 45 to 60°. In practice, however, the maximum flap travel usually is limited to about 30°, under which conditions anywhere from 50 to 70% of the maximum possible increase in lift may be realized. The general effect of a trailing edge flap on the lift and drag is shown in Fig. 11. Flaps may be either simple, the whole trailing-edge section of the wing being pulled down as a unit, or split, the lower part of the trailing edge being depressed while the upper part is left rigid and the form of the upper surface of the wing remains undisturbed. The split type is somewhat the more efficient of the two but the difference is not great.



Average divisional values—

	λC	T	D
Navy data	12.5	7.5	3.0
Handley-Page	12.5	6.5-7.5	0.8-1.8

FIG. 12. Proportions of Typical Leading Edge Slots

Fixed or variable leading-edge slots commonly are used only in that portion of the wing ahead of the ailerons, although some designs use slots over the full span. Proportions for typical installations are shown in Fig. 12, and the effect on the lift and drag of the airplane at full scale is shown in Fig. 13. The effect of the slots is to prevent "burbling" and maintain smooth air flow, with consequent increase in lift, at very large angles of attack. The maximum improvement in lift coefficient is obtained by a combined use of leading edge slots and trailing edge flaps. Some of the combinations that have been tried, all of them based on the same airfoil, are shown in Fig. 14. The maximum L_c obtained is recorded beside the sketch in each case.

SCALE EFFECTS AND REYNOLDS NUMBER.—The simple variation of resultant air force with density, area and velocity, of equation [1], is not strictly true for all cases. It can be shown that the simple coefficients R_c, L_c, and D_c are not simple numbers, but functions of Vl/ν, where V is the velocity, l is some lineal dimension of the body moving in the fluid, and ν is the kinematic viscosity, or the viscosity per unit density of the fluid. The factor Vl/ν usually is called the Reynolds Number. In most cases the variation of the coefficients with scale is a secondary factor, but it becomes an important one when predicting full-scale aerodynamic characteristics from wind tunnel tests on models.

As a general rule, the maximum lift coefficient for a comparatively thin airfoil section tends to increase slightly with increase in Reynolds Number. A test made on a small model of a Clark Y airfoil at about 30 mi. per hr. gave a maximum lift coefficient of 0.00315, and a test made at approximately full scale and flight speed yielded a coefficient of 0.00350. On the other hand, thick sections, which have a relatively high value of maximum lift coefficient, tend to show a decrease as full-scale conditions are approached. The U.S.A. 35, shows an absolute coefficient of 0.00375 under model-test conditions, and 0.00308 at full scale. Induced drag does not change with Reynolds Number, but the effect on profile drag may be marked. The general effect, regardless of section, is to show a decrease in drag as full-scale conditions are approached. This is connected intimately with the change of skin friction. See below. The reduction in drag usually is greatest for the thicker sections. Center of pressure travel virtually is unaffected by changes in Reynolds Number.

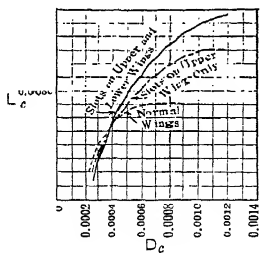
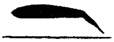
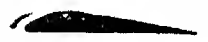
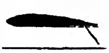
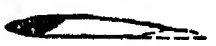
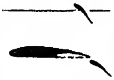
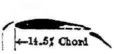
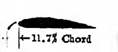


FIG. 13. Effect of Handley-Page Wing Slots at Full Scale on Bristol Fighter

3. PARASITE RESISTANCE

PARASITE DRAG is the air resistance of all parts of an aircraft except the wings. Drag arises from two sources: 1. Skin friction. 2. Turbulence. The latter, which arises from the breakdown in the smooth flow of air around the body and the subsequent formation of eddies, is usually much more important.

SKIN FRICTION may be determined by either of two formulas. The choice between the two depends on the conditions existing in the layer of air immediately adjacent to the

		Angle of lift increasing surface to basic airfoil	Flap chord in percent of basic airfoil chord	Max. lift coefficient C_L max.	Speed range ratio C_L max. / C_D min.	L/D at max. lift (2)	Angle of attack of basic airfoil at max. lift	Percent improvement in lift: Over plain airfoil (1)	Percent improvement in lift: Over simple flap	Percent improvement in speed-range ratio: Over plain airfoil (1)	Percent improvement in speed-range ratio: Over simple flap	Reference, N.A.C.A. report
Plain basic airfoil (1)				1.291	85.0	7.6	15°					T.R. 427
Simple flap		45°	30%	1.950	128.2	4.0	12°	51%		51%		T.R. 427
Slotted flap with cover plate		45°	30%	1.980	120.5	4.0	12°	53%	1.5%	42%	None	T.R. 427
Double slot and flap		45°	30%	2.442	117.5	4.0	16°	19%	25%	38%	None	T.R. 427
Fixed slot, cut in basic airfoil				1.772	73.8	5.3	24°	37%	None	None	None	T.R. 427
N.A.C.A. fixed auxiliary airfoil, ahead of basic airfoil (3)		0°	14.5%	(3) 1.705	104.5	3.5 Approx.	24°	32%	None	23%	None	T.R. 428
N.A.C.A. optimum fixed slot (3)				(3) 1.648	76.4		24°	27%	None	None	None	T.R. 400
Handley-Page type automatic slot (3)				(3) 1.632	114.2		28°	26%	None	34.5%	None	T.R. 400
Front slot and simple flap		45°	30%	2.182	91.0	3.8	19°	69%	12%	7%	None	T.R. 427
Front slot and slotted flap		45°	30%	2.261	93.2	3.8	19°	75%	16%	10%	None	T.R. 427
Triple slot and flap		45°	30%	2.600	87.3	3.8	20°	101%	33%	3%	None	T.R. 427
Split flap, rotated down, no backward movement		50°	30%	2.16	138.5	4.3	14°	70%	10.7%	63%	8%	T.N. 422
Split flap, trailing edge moved vertically downward (Zap)		60°	30%	2.35	150.8	3.7 Approx.	13°	85%	20.5%	77%	17.5%	T.N. 428
Split flap, hinge point moved back to 90% of chord		54°	40%	(3) 2.222	142.2	3.8	13°	75%	14%	67%	11%	T.N. 422
Hall wing, front slot closed		48°	34%	2.08	138.8	3.6	13°	64%	6.7%	63%	8.1%	T.N. 417
Fowler wing projected (area increased approx. 31% over basic airfoil) (3)		40°	40%	(3) 2.422	155.3	4.25	15°	90%	24.3%	83%	21.2%	T.N. 419
Fowler wing with N.A.C.A. 22 slot and round nose of basic airfoil		Slot −40° Flap +40°	Slot 14.5% Flap 40%	(3) 2.49	(3)(4) 137 (5) 199	3.76	21° to 25°	96%	28.1%	(3)(4) 61% (5) 134%	7% (5) 55%	T.N. 459
N.A.C.A. 22 slot on plain wing with rounded nose		Slot −45°	Slot 14.5%	(3) 1.78	(4) 97.7 (5) 114.2	4.8	30°	40%	None	(3)(4) 15% (5) 35%	None	T.N. 459

NOTES.—1. In comparing properties of modified sections with the plain basic section, the coefficients used in each case were obtained under similar test conditions. Drag coefficients were taken with slot closed (if movable) and with flap neutral.

2. A low value of L/D at maximum lift indicates a steep glide angle and consequently a short landing. An L/D of 8 corresponds to a gliding angle of approximately 7 deg., and a value of 3.5 means about 16 deg. (T.R. 428).

3. Based on total wing area; lift increasing device extended and projected on original chord line. Actually this area is structural area necessary and forms the basis for the comparison with the simple flap.

4. With slot and flap retracted the airfoil is not perfect, having a drag coefficient of 0.0182 compared with 0.0156 for the plain airfoil.

5. Based on contracted area.

FIG. 14.—Characteristics of High Lift Devices Applied to the Clark Y Wing (*Aviation*) (The Reynolds Number for all tests is 609,000 which corresponds to about one-third that for an ordinary small airplane at landing speed)

frictional surface. The effect of viscosity of fluid on the flow of air is confined to a *boundary layer*, within which the velocity gradient is exceedingly steep. The thickness of the boundary layer seldom exceeds 1/2 in. in practice, except on the hulls of airships, and at a distance of 2 ft. from the leading edge of a flat plate at 200 mi. per hr. it is about 3/16 in.

At low Reynolds Numbers the flow in the boundary layer is smooth and laminar, and the friction is determined by the formulas

$$D_f = D_{cf}\, S_e V^2 \quad . \; . \; . \quad [13]; \qquad D_{cf} = 0.0034/\sqrt{N} \quad . \; . \quad [14]$$

where N = Reynolds Number; S_e = total surface exposed to the air. In the case of a wing or flat plate, S_e would be twice the area as ordinarily defined. At very high Reynolds Numbers the boundary-layer flow becomes turbulent. The friction then is much higher and varies according to a different law, following the formula

$$D_{cf} = 0.00019/\sqrt[5]{N} \quad . \; . \; . \; . \; . \; . \; . \; . \; . \; . \quad [15]$$

At intermediate values of N the motion is laminar on the forward part of the surface, turbulent on the after part, and the friction has an intermediate value. The transition

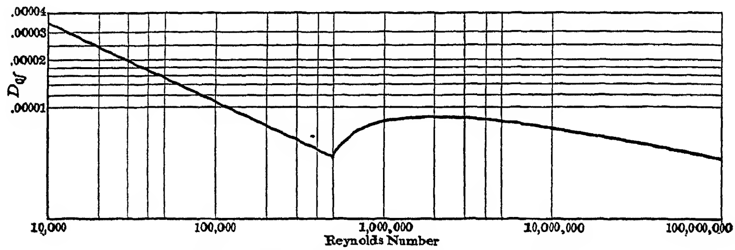


FIG. 15. Scale Effect on Skin Friction Coefficient

from laminar to turbulent motion is governed in part by the degree of roughness of the frictional surface, and in part by the general conditions existing in the stream. On the average it occurs at about N_x = 500,000. $N_x = Vx/\nu$, where x is distance from leading edge of the surface to the point under examination, and the other symbols have the same meaning as in the general definition of Reynolds Number. Fig. 15 is a typical curve of frictional coefficient against Reynolds Number showing the two distinct zones and the transition between them.

SPHERES AND HEMISPHERES.—The resistance of a sphere varies through a wide range with Reynolds Number. At Reynolds Number 500,000, corresponding to a sphere 1 ft. diam. and an air velocity of 45 mi. per hr. under the atmospheric conditions standard at sea-level, D_c = 0.0002; at 1,000,000, D_c = 0.0003; at 4,000,000, D_c = 0.00037.

Eiffel found that if he attached a solid cone, whose base diameter was that of the sphere, and whose conning angle was 20°, to the downwind face of a hemisphere with convex side toward the wind, the total drag was approximately 50% of that of a complete sphere of the same diameter.

STREAMLINE BODIES.—The resistance of a number of typical streamline bodies is shown in Fig. 16. The fineness ratio, or the maximum length divided by maximum diameter, of a streamlined body influences the total drag. In general, well-shaped streamlined bodies exhibit the best resistance values at fineness ratios of from 2 to 3. The very best streamline forms have a disc ratio, or ratio of their own resistance to that of a normal flat plate of area equal to the projected cross-section area of the body, as low as 1/40.

Good streamlined shapes are very sensitive to the effect of minor projections or slight surface discontinuities which disturb the smooth air flow. For example, the wrapping of a single turn of a thread 0.014 in. diam. around a section 1 in. from the nose of a smooth streamline body 8 in. diam. and 24 in. long has proved sufficient to increase the total drag approximately 67%. Seven sets of similar thread rings placed at 1-in. intervals from the nose practically doubled the drag of the bare body.

STRUTS.—The relative drag of struts of several cross-sections is shown in Fig. 17. Minimum drag for struts is associated with fineness ratios of 3 to 3 1/2. The drag coefficients for the best struts are about twice as high as for the best streamline bodies, due to the substitution of two-dimensional for three-dimensional flow.

Minimum strut resistance is obtained with the axis of the strut parallel to the relative wind. Yawing the average strut 7° increases the drag by 50%; 8°, approximately 100%. Round members may be used as struts without fairing if their axes lie at less than 45° to the relative wind, as the section around which the air flows has then become elliptical.

FLAT PLATES NORMAL TO WIND.—For a rectangular plate, of aspect ratio = 1, the drag coefficient D_c is 0.00328. With increase of aspect ratio, the drag coefficient must be increased by a factor, as follows:

Aspect Ratio....................	2	4	6	10	14	18
Coefficient Correction Factor......	1.05	1.08	1.10	1.15	1.24	1.30

WIRES AND CABLES.—The resistance coefficient for a single strand of round wire, cable, or streamlined wire is given in Table 2. The streamlined wire mentioned in the table actually is rolled to a roughly lenticular section, with a major axis about 2 1/2 times the minor one. When round or streamlined wires are used in pairs, the resistance of the combination usually is below that of the total for the two wires singly, the saving by interference depending on the spacing between wires. The relative drag of wires in combination is shown in Fig. 18. The drag of each exposed turnbuckle is figured equivalent to approximately 2 ft. of additional length of wire or cable. Each exposed streamlined-wire end fitting should be considered equivalent to 3 ft. of wire.

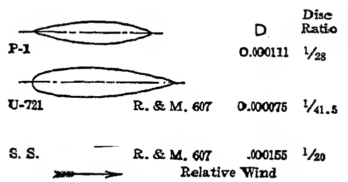


FIG. 16. Typical Streamline Bodies, their Resistance Coefficients and their Relative Resistance to a Flat Disc of Equal Diameter

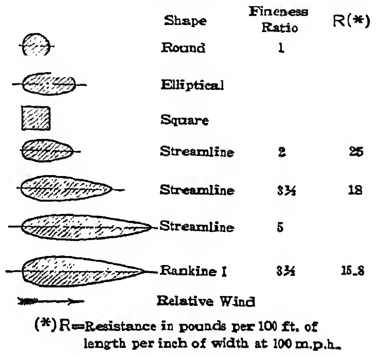


FIG. 17. Relative Drag of Struts

LANDING WHEELS.—The drag of standard wire-spoke wheels with high-pressure tires is shown in Table 3. By adding streamlined housings or "pants" to a high-pressure type wheel, the drag of the combination may be reduced to approximately 40% below that of the wheel with its ordinary disc fairings. A recent test indicated the drag of two large semi-low-pressure tires as follows:

Tire size, in..................	40 × 10	15.00 – 16
Drag, lb. at 100 mi. per hr....	28.9	33.6

Partially enclosing the tire in a streamlined housing reduced the drag 40% on the average. In both cases the axle and landing gear struts were in place, so that the results include a certain amount of interference effect from these members.

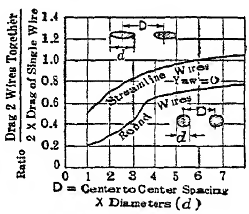


FIG. 18. Drag of Adjacent Wires and Cables

FUSELAGES.—Drag coefficients for a series of fuselage models are listed in Table 4. In all cases the fuselages were tested without landing gear, wings, or tail surfaces, but with the modifications indicated in each case. These tests were made at relatively low Reynolds Numbers. The coefficients are given in the usual form, based on cross-sectional

Table 2. Drag for Wire
(U. S. Army Data)
Pounds per foot at 100 miles per hour in an air of standard density.

Round		Cable		Streamline	
Size	Drag	Size	Drag	Size	Drag
No. 14 (B. & S.)	0.142	1/16 in.	0.180	6–40	0.0551
10 "	.245	1/8 "	.380	10–32	.0592
8 "	.318	3/16 "	.575	1/4–28	.0734
6 "	.410	1/4 "	.770	5/16–24	.0898
4 "	.525	3/8 "	1.165	3/8–24	.1121
....	....	1/2 "	1.555	7/16–20	.1388
....	....	....	....	1/2–20	.1530

area. In typical practice, the disc ratio for fuselages, either with air-cooled or liquid cooled engines, usually ranges between 1/4 and 1/8.

Cowling for Air-cooled Engines.—Mounting an air-cooled engine in the nose of a fuselage ordinarily increases the total drag, at full scale, about 200 to 400%. The N.A.C.A. in America, and Townend in England, have developed forms of annular engine cowling which offset from one-half to two-thirds of this increase. Fig. 19 shows the N.A.C.A. cowling applied to a cabin and to an open cockpit fuselage. Table 5 shows the drag in pounds for various combinations at 100 miles per hour. An application of this cowling to a standard airplane usually increases the top speed by from 10 to 15%. Roughly, the cowl reduces the additional resistance due to the engine by some 45 to 70%.

The Townend, or ring, cowling is similar to the N.A.C.A. type in principle but, as indicated in Fig. 20, it is considerably shorter and has less curvature. In both cases the cowl acts to reestablish a normal smooth flow of air along the sides of the fuselage, preventing a turbulent divergence of the air directly disturbed by passing over the engine.

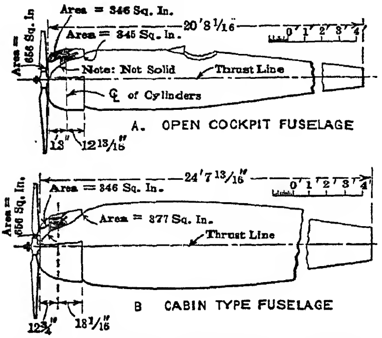


Fig. 19. Application of N.A.C.A. Cowling to Typical Fuselages

NACELLES.—The N.A.C.A. cowling has been applied to engine nacelles to reduce the drag of an air-cooled engine. Fig. 21 shows the so-called conventional, and N.A.C.A.

Table 3.—Drag of Airplane Wheels at 100 Miles per Hour
(U. S. Army Data)

Standard Tire Size, in.	Bare Wheel, Spokes Enclosed	Partial Fairing, Discs Covering Spokes Only	Full Fairing, Discs Run Tangent to Tire Sides	Standard Tire Size, in.	Bare Wheel, Spokes Enclosed	Partial Fairing, Discs Covering Spokes Only	Full Fairing, Discs Run Tangent to Tire Sides
	Drag, lb.				Drag, lb.		
26×3	10.80	7.20	3.60	36×8	33.80	22.50	11.25
28×4	14.10	9.40	4.70	44×10	51.20	34.10	17.05
30×5	18.15	12.10	6.05	54×12	75.90	50.60	25.30
32×6	22.85	15.25	7.61	......	......	......	......

Table 4.—Drag Coefficients for Several Fuselages

Fuselage	Drag Coefficient
Flat-nosed rectangular fuselage, semi-circular top, no cockpit, no projections . .	0.000680
Round-nosed elliptical fuselage, no cockpit, no projections..................	0.000159
Round-nosed elliptical fuselage, uncowled radial engine, no cockpit, no other projections..................	0.000643
Round-nosed faired rectangular fuselage, V-type engine, no cockpit, no radiators or other projections..................	0.000261
Flat-sided fuselage complete, nose radiator, two cockpits, exhaust manifold, wind-shields, headrest and tail skid..................	0.000980

Table 5.—Effect of N.A.C.A. Cowling—2 Fuselages

Fuselage	Cowling	Drag, lb. at 100 m.p.h.	Increase Over Faired Fuselage, Percent
Open cockpit Fig. 19 A.	Faired fuselage, engine removed. Windshields off, cockpit covered..................	28	−33
	Faired fuselage, engine removed, cockpit open.......	42	0
	Fuselage with engine, no enclosing cowling..........	136	224
	Fuselage with engine, N.A.C.A. cowling............	73	74
Cabin type Fig. 19 B.	Engine removed, nose rounded..................	40	0
	Fuselage with engine, no enclosing cowling..........	119	198
	Fuselage with engine, N.A.C.A. cowling............	75	88

nacelles, both for a 300-Hp., 9-cylinder engine. The relative drag of the various combinations at 100 mi. per hr. is shown in Table 6.

INTERFERENCE DRAG.—The total drag of airplane components in close proximity to one another, for example, fuselage and wings, or fuselage and landing gear, always is considerably greater than the sum of the resistances of the individual units, due to mutual interference. In general, no members should intersect at sharp angles, but generous fairings or fillets should be provided. The relation between wings and fuselages has been studied for many cases, and the importance of careful filleting has been established. In one high-wing monoplane, studied at full scale, a simple 12-in. radius fillet at the intersection of the under-surface of the wing and cabin effected a reduction in the total resistance of the combination of almost 2%. Research has indicated that the longitudinal rate of change of the radius of the fillet is important, and that the radius at the trailing edge of the wing should be much larger than at the leading edge. Fig. 22 shows the lines for the optimum fillet for a low-wing monoplane.

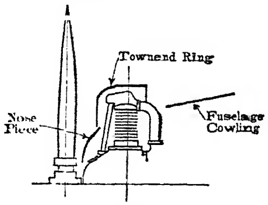


FIG. 20. Engine Fitted Cowling Ring of Townend Type

The location of an outboard engine nacelle with respect to a wing has an important bearing on interference effect and propulsive efficiency. Studies made by the N.A.C.A. indicate that all nacelles should be completely cowled. In general, nacelles placed above a wing show much greater interference effect than those placed below. Nacelles placed closer than approximately one nacelle diameter below the lower surface of a wing should be completely faired into the wing, but in no case should the engine cowling hood be faired into the wing. The best location for a completely cowled nacelle for minimum drag and for greatest propulsive efficiency is with the propeller hub in line with, and about 25% ahead of the leading edge of the wing. The location and fairing of nacelles have a great effect on top speeds, but little influence on climbing powers.

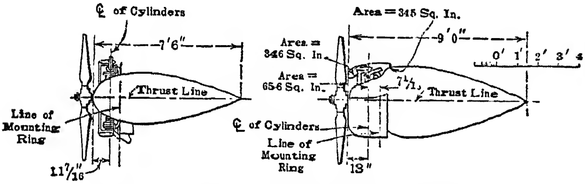


FIG. 21. Conventional and N.A.C.A. Nacelles

FLOATS AND HULLS.—Wind tunnel tests on various groups of floats and hulls yielded the range of drag coefficients, in terms of square feet of maximum cross section, given in Table 7. In most cases the minimum drag occurs at 0° pitch of the deck line, and on an average the drag is doubled at a pitch angle of 10°, and multiplied by five at about 18°. Floats and hulls at positive pitch angles yield considerable lift. In most cases the lift-drag ratio has a maximum value of 2.2 to 2.5 at a pitch angle of 12 or 14°.

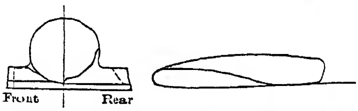


FIG. 22. Optimum Fillet for Low-wing Monoplane (Klein)

Table 6.—Drag of Engine and Nacelles

Form	Drag, lb. at 100 mi. per hr.	Percent Reduction from Bare Engine
Bare engine	178	0
Engine in conventional nacelle	155	12.9
Engine in N.A.C.A. nacelle	43	75.9

Table 7.—Drag of Floats and Hulls

Type	No. of Tests	Drag Coefficient, D_c		
		Maximum	Minimum	Average
Seaplane floats	8	0.000855	0.000651	0.000728
Flying boat hulls	15	.000366	.000267	.000307
Wing-tip floats	....	.000800	.000400	.000600

TOTAL AIRPLANE DRAG.—The distribution of the total drag for complete airplanes of several types at full scale is indicated in Table 8.

The total structural drag of any airplane may be conveniently expressed in terms of the flat-plate area having equivalent drag. The equivalent flat-plate area of an airplane of good modern design, with a retractable or a very carefully streamlined landing-gear, can be kept at low as 1 sq. ft. for every 1000 lb. of gross weight or, especially in machines of large size, even as low as 0.40 sq. ft. in exceptional cases.

Table 8.—Percentage Drag Distribution of Typical Airplanes

	Air-cooled Cabin Monoplane	Prestone-cooled Monoplane	Prestone-cooled Sesqui-plane	Air-cooled Open Biplane
Fuselage and engine	32.3	17.2	17.9	27.8
Radiator	....	7.4	7.2	....
Landing gear	13.4	12.4	12.5	10.1
Tail surfaces	4.7	10.9	9.9	8.3
Wings and struts	49.6	52.1	52.5	53.8

4. PERFORMANCE CALCULATION

All performance calculations are based on the relation between power required for flight and power available under the given condition. The power required, P, is a direct function of the total drag, *i.e.*, wing drag + parasite drag.

$$P = DV/375 \qquad [16]$$

where D = total drag, lb.; V = speed, mi. per hr.

SLIP STREAM EFFECT.—In calculating total parasite drag at a given speed of flight, allowance must be made for slip-stream effect. The slip-stream of the propeller is taken roughly as a hollow cylinder, concentric with the thrust axis, whose inside diameter is 0.2 × propeller diameter, and whose outside diameter is from 0.8 to 0.9 × propeller diameter. The parasitic drag of all objects which lie within the slip stream must be calculated on the basis of slip stream velocity, which always is greater than the speed of normal horizontal flight. Its value is

$$V_s = V\sqrt{1 + (490\,T/D^2V^2)} \qquad [17]$$

where V_s = slip stream velocity, mi. per hr.; V = speed of flight, mi. per hr.; T = propeller thrust, lb.; D = propeller diam., ft. The propeller thrust for use in formula [17] is

$$T = 375\,P\eta/V \qquad [18]$$

where η = propeller efficiency, percent; other notation as before. For propeller efficiency see p. 14–112. The power available under any set of conditions equals the horsepower developed by the engine under those conditions, multiplied by propeller efficiency.

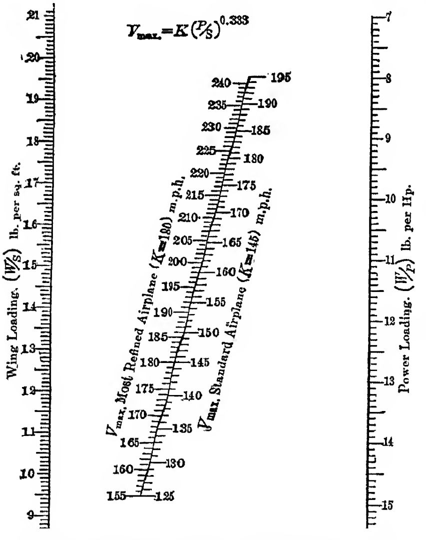


FIG. 23. Nomograph for Estimating Maximum Speed of Airplanes Based on Equation 21. (*Aviation*)

EFFECT OF ALTITUDE ON PERFORMANCE.—Increase in altitude is accompanied by a reduction in density of the air, with consequent reduction of power output for an unsupercharged engine. See Standard Atmosphere, p. 14–99. The general relation between power output, P, at constant r.p.m., and density ρ, based on average curves of a number of tests is

$$P = P_0(\rho/\rho_0)^{1.2} \qquad [19]$$

where the subscript o indicates sea-level conditions. With a controllable-pitch propeller

the power available for propulsion will follow the same formula. When the propeller is of the fixed-pitch type there is a further loss due to decrease of r.p.m. at higher altitudes, and the effective propulsive power is more nearly proportional to $\rho^{1.3}$

For all practical purposes, the total drag at any given angle of attack may be considered as independent of air density, and the power required for straight horizontal flight, P_r, is given by

$$P_r = P_{ro}(\rho_o/\rho)^{1/2}, \quad . \quad . \quad . \quad [20]$$

since the speed of flight at a given angle of attack varies inversely as the square root of the air density. The subscript o indicates sea-level conditions.

Where engines are supercharged, sea-level output may be maintained to considerable altitudes. The power output at any given altitude then must be obtained from performance curves of the engine in question. It is common practice in 1935 to supply engines supercharged to maintain as much power up to an altitude of 5000 to 10,000 ft. as it is safe to draw from them continuously at any altitude. Maximum speed will increase with altitude until the critical altitude is reached, and the airplane may be as much as 10% faster at 10,000 ft. than it is at sea-level.

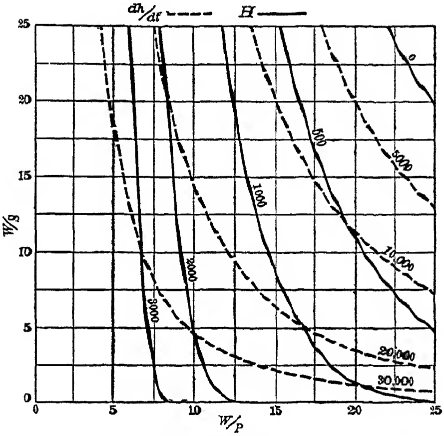


FIG. 24. Chart for Estimation Rate of Climb at Sea Level, and Ceiling

MINIMUM OR STALLING SPEED for horizontal flight is

$$V_{min} = \sqrt{W/L_{c\,max}} \quad . \quad . \quad . \quad . \quad . \quad . \quad . \quad . \quad [21]$$

where W = gross weight, lb.; S = effective wing area, sq. ft. On the average, for airplanes without flaps or slots,

$$V_{min} = 17\sqrt{W/S} \quad . \quad . \quad . \quad . \quad . \quad . \quad . \quad . \quad . \quad . \quad . \quad [22]$$

Where flaps and slots are used, the constant ranges from 12 to 15.

MAXIMUM HORIZONTAL SPEED.—An empirical formula for maximum speed, based on actual performance of a large number of airplanes is

$$V_{max} = K(P/S)^{1/3} \quad . \quad . \quad . \quad . \quad . \quad . \quad . \quad . \quad . \quad . \quad . \quad [23]$$

where P = nominal horsepower; S = wing area, sq. ft.; K = a constant. For ordinary airplanes, K = 145; for exceptionally well "cleaned-up" machines, K = 150 to 183. The best transport and bombing planes of 1935 show K = 175 to 183; pursuit planes, K = 160 to 175; racing planes, K = 155 to 165. Formula [23] gives a close approximation for monoplanes and biplanes of all types. With a little experience in estimating K, it can be used to predict the maximum speed of a new machine within a percentage error that should not exceed 4. Fig. 23 gives maximum speed in terms of horsepower per square foot of wing area for machines of average quality and for exceptionally clean ones.

RATE OF CLIMB.—The rate of climb, dh/dt, at sea-level depends on the reserve of engine power, in excess of that required for level flight, available for doing work against gravity. It is approximated by

$$(dh/dt) = \{(25{,}000\,P/W) - 225\sqrt{W/S}\} \quad . \quad . \quad . \quad . \quad . \quad . \quad . \quad . \quad [24]$$

Fig. 24 is a chart based on formulas [24] and [27], showing the relationship between rate of climb at sea-level, ceiling, wing loading, and power loading. Both ceiling and rate of climb formulas are predicated on the use of controllable-pitch propellers. This virtually is universal practice with all machines on which performance is a serious objective

CEILING CALCULATION.—An airplane reaches its absolute ceiling when the power available for flight exactly equals the power required. It can be shown that the ceiling, H, with an unsupercharged engine, can be expressed in terms of power available, P_{ao}, and power required P_{ro}, at sea-level, by

$$H = 40{,}000\log_{10}(P_{ao}/P_{ro}) \quad . \quad . \quad . \quad . \quad . \quad . \quad . \quad . \quad . \quad [25]$$

An empirical formula, based on the performance of a large number of unsupercharged airplanes, is

$$H = 40{,}000 \log_{10}\{120/(W/P)\sqrt{W/S}\} \quad [26]$$

W = gross weight, lb.; P = nominal horsepower; S = wing area, sq. ft. See Fig. 23.

Where the engine is supercharged to produce a constant power up to a certain altitude, and to fall off in accordance with the normal rule after that, the formula becomes

$$H = 0.75\, h_c + 40{,}000 \log_{10} [120/(W/P)\sqrt{W/S}] \quad [27]$$

where h_c = critical altitude.

A RANGE OF FLIGHT, x, in miles may be approximated by

$$x = 12{,}500 \log_{10} (W_1/W_2) \quad [28]$$

or more accurately by

$$x = 8000 \log_{10} (W_1/W_2) + 2400\{(W_1 - W_2)/W_1\} \quad [29]$$

where W_1 = initial gross weight, lb., and W_2 = final gross weight. $(W_1 - W_2)$ is the weight of fuel and oil consumed during the flight. For a very "clean" machine the constants in the formula may be increased as much as 25%. The range of flight, in accordance with the formula is given in Table 9.

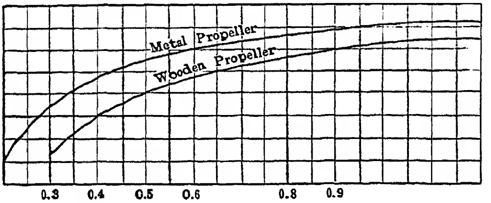


Fig. 25. Average Propeller Efficiencies. (Air Corps Data)

TAKE-OFF DISTANCE.— An empirical formula for the take-off distance x, in feet, from an ordinary field in a dead calm with a controllable-pitch propeller, but without the use of flaps or slots to reduce take-off speed, is given by

$$x = 200(W/S)/[\{110/(W/P)\} - 1] \quad [30]$$

With a propeller of fixed pitch, the coefficient of the first term in the denominator should be reduced to 85. With flaps and slots judiciously used, the run may be reduced by as much as a third of the figure given by the formula, the exact reduction depending on the particular variable-lift device selected. Taking off into the wind shortens the run. For a typical military or commercial plane, facing a 20-mile wind reduces the run about 50%.

5. POWER PLANTS

PROPELLERS.—Propeller characteristics usually are expressed in terms of the function V/ND, where V is speed of flight, N the r.p.m. and D the diameter. General propeller efficiency curves in terms of this function are given in Fig. 25. Engine power and propeller efficiency vary with airplane speeds. For most practical purposes the relationship between air speed and propeller revolutions per minute can be obtained from Fig. 26.

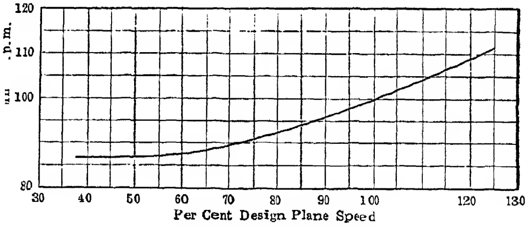


Fig. 26. Full Throttle R.P.M. at Various Speeds. (Air Corps Data)

SELECTION OF METAL PROPELLERS.—Reference should be made to N.A.C.A. Technical Report No. 350, Working Charts for the Selection of Aluminum-Alloy Propellers, by F. E. Weick, for complete charts and tables

Table 9.—Flight Range and Initial Fuel and Oil Load

(Average airplanes)

Proportion of initial gross weight in fuel and oil	0.2	0.3	0.4	0.5	0.6
Approximate range, miles	1260	1960	2740	3610	4620

for the selection of metal propellers for a wide range of fuselage shapes and conditions. Fig. 27 is a typical chart.

PROPELLER DIAMETER.—A general formula for the calculation of propeller diameter is $D =$

$K \sqrt{(\frac{\ }{100}}$

where D = propeller diam., ft.; V = maximum speed, mi. per hr.; N = rev. per min.; K = a constant = 9.6 for a 2-bladed, and 9 for a 3-bladed propeller. The weight of fixed-pitch aluminum-alloy propellers can be calculated approximately by the formula

W : . . . [32]

where W = weight, lb.; P = nominal engine horsepower. Controllable-pitch types run about 30% heavier. Average weights of 2- and 3-bladed metal fixed- and controllable-pitch (aluminum-alloy blades in steel hubs) are given in Table 10.

ENGINES. — The general characteristics of a number of well-known American engines, which are commercially available are given in Table 11.

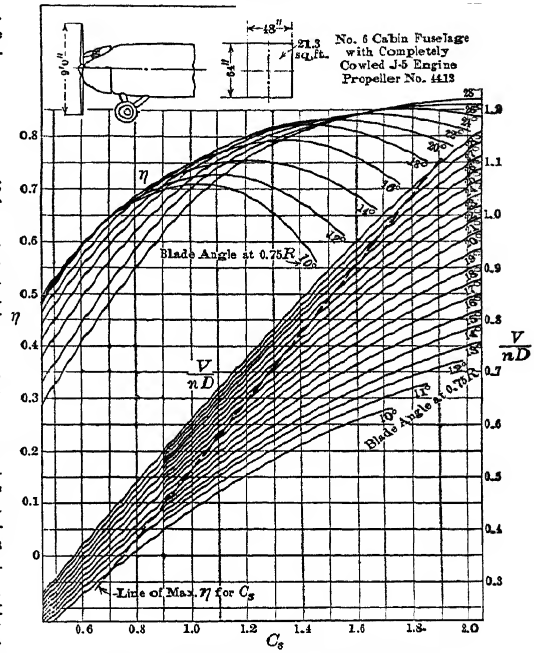


Fig. 27. Typical Chart for Propeller Selection (Weick, N.A.C.A. Report No. 350)

Table 10.—Average Weights of Metal Propellers

Fixed Pitch Type				Controllable (hydraulic) Pitch Type			
No. of Blades	Diam. Range	Hp. Range	Weight Range, lb.	No. of Blades	Diam. Range	Hp. Range	Weight Range, lb.
2	6′ 6″ to 8′ 6″	65–125	34– 46	2	8′ 0″ to 10′ 0″	225–550	105–162
	8′ 0″ to 9′ 0″	150–225	79– 88		9′ 6″ to 10′ 6″	600–750	215–218
	8′ 6″ to 10′ 9″	250–300	96–118	......	....	....	....
3	8′ 6″ to 13′ 0″	400–up	128–267	3	9′ 0″ to 10′ 0″	550–825	227–242
......	....	....	....		11′ 6″ to 13′ 0″	725–900	334–361

6. BALANCE, STABILITY AND CONTROL

Stability, balance, control and maneuvrability are closely related. The service for which any machine is intended dictates the predominant characteristic. A high degree of inherent stability is desirable in certain types intended particularly for the use of amateur pilots of limited skill, and great maneuvrability at the expense of stability is paramount for fighters.

LONGITUDINAL STABILITY AND BALANCE.—The longitudinal stability of an airplane is measured in terms of its pitching moments. These depend principally on the location of the center of pressure on the mean aerodynamic chord of the wing cellule, on the relationship between the center of pressure and the center of gravity of the airplane, and upon the size, location, and setting of the horizontal tail surfaces. For most machines the center of gravity lies in a vertical plane that intersects the mean aerody-

Table 11.—Typical American Engine Specifications

Condensed from *Aviation*, April, 1935

NOTE.—No attempt is here made to list all models made by the several manufacturers. The table is merely indicative of the range of engine sizes commercially available. Many models listed as direct drive also are available with reduction gearing.

	Model		Rating				Weights	
Aeromarine Plane & Motor...	AR-3...........	3-Rad	50	2125	S.L.		150	3.00
Aeronautical Corp. of America	Aeronca E-113B..	2-Hopp	40	2500	S.L.		121	3.02
Continental Aircraft Eng. Co.	A-40-2..........	4-Hopp	37	2550	S.L.		144	3.90
Continental Aircraft Eng. Co.	,	7-Rad	225	2000	S.L.		425	1.90
Jacobs Aircraft Engine Co....	L-4.................	7-Rad	225	2000	S.L.		415	1.85
Kinner Airplane & Motor Co.	K-5.................	5-Rad	100	1810	S.L.		280	2.80
Kinner Airplane & Motor Co.	C-5.................	5-Rad	210	1900	S.L.		420	2.00
Kinner Airplane & Motor Co.	C-7.................	7-Rad	300	1800	S.L.	73	600	2.00
Lambert Eng. & Machine Co.	R-266................	5-Rad	90	2375			214	2.40
Lycoming Mfg. Co...........	R-680-4...............	9-Rad	225	2100			505	2.24
Lycoming Mfg. Co...........	R-680-5A..............	9-Rad	260	2300			505	1.94
Menasco Mfg. Co..........	C-4.................	4-IdL	125	2175			290	2.32
Menasco Mfg. Co...........	B-6.................	6-IdL	160	1975			385	2.40
Pratt & Whitney Aircraft Co.	Wasp, Jr. TB..........	9-Rad	400	2200	S.L.		596	1.49
Pratt & Whitney Aircraft Co.	Wasp S1-H1-G.........	9-Rad	550	2200	8000		875	1.58
Pratt & Whitney Aircraft Co.	Twin Wasp, Jr. SA1-G..	14-Rad S	700	2500	8500		999	1.42
Pratt & Whitney Aircraft Co.	Twin Wasp S1-A2-G...	14-Rad S	800	2400	7000		1174	1.46
Pratt & Whitney Aircraft Co.	Hornet SD............	9-Rad	675	2050	6000		880	1.30
Pratt & Whitney Aircraft Co.	Hornet SD-G..........	9-Rad	700	2150	6500		980	1.40
Pratt & Whitney Aircraft Co.	Hornet S2E-G.........	9-Rad	750	2250	3000		1015	.35
Pratt & Whitney Aircraft Co.	Hornet T1-D1.........	9-Rad	675	2000	S.L.			.30
Pratt & Whitney Aircraft Co.	Hornet T1-D1-G......	9-Rad	675	2150	S.L.		980	
Pratt & Whitney Aircraft Co.	Hornet T1E...........	9-Rad	725	2050	S.L.		920	
Pratt & Whitney Aircraft Co.	Hornet T2E-G.........	9-Rad	750	2250	S.L.		1015	.35
Ranger Engineering Corp.....	Ranger 6-390B........	6-IdL	145	2250			350	2.33
Ranger Engineering Corp.....	Ranger V-770-SG......	12-IdV	420	2800			600	1.43
Warner Aircraft Corp........	Scarab, Jr.............	5-Rad	90	2025	S.L.		230	2.50
Warner Aircraft Corp........	Scarab................	7-Rad	125	2050			285	2.25
Warner Aircraft Corp........	Super Scarab..........	7-Rad	145	2050			300	2.05
Wright Aeronautical Corp....	Whirlwind 175........	5-Rad	175	2000	S.L.		415	2.38
Wright Aeronautical Corp....	Whirlwind 250........	7-Rad	250	2000	S.L.		495	1.98
Wright Aeronautical Corp....	Whirlwind 440........	9-Rad	440	2200	S.L.		600	1.36
Wright Aeronautical Corp....	Whirlwind STR-1510-C1	14-Rad S	715	2400	6500		985	.38
Wright Aeronautical Corp....	Cyclone GR-1820-F1...	9-Rad	700	1950	S.L.		1038	.48
Wright Aeronautical Corp....	Cyclone SR-1820-F2...	9-Rad	735	1950	4000		945	.28
Wright Aeronautical Corp....	Cyclone R-1820-F11...	9-Rad	690	1950	S.L.		937	.36
Wright Aeronautical Corp....	Cyclone SR-1820-F31..	9-Rad	670	1900	2500		937	.40
Wright Aeronautical Corp....	Cyclone SR-1820-F32..	9-Rad	675	1950	6500		945	.40
Wright Aeronautical Corp....	Cyclone SR-1820-F33...	9-Rad	650	1950	9500		945	.45

A = air cooled; W = liquid cooled; Rad. = radial; Id = inverted; L = in-line; V = vee type.

namic chord at a point about 30% of the chord from the leading edge. It seldom intersects further forward than 28% of the chord, or further aft than 33%. The longitudinal stability is steadily decreased by moving the center of gravity back along the wing chord, and most machines will become positively unstable with the center of gravity approximately at the 35% point.

HORIZONTAL TAIL AREA AND LONGITUDINAL STABILITY.—As a general rule, the total horizontal tail surface, *i.e.*, stabilizer plus elevators, for biplanes ranges from 11 to 14% of the total wing area, and for monoplanes from 13 to 16%. The distribution of horizontal area between fixed stabilizer and elevators varies, in percent of elevator area to total area, from about 38 to 55. The larger the tail surfaces in proportion to the wings, the further back the center of gravity can be placed along the wing chord, without causing instability.

VERTICAL TAIL AREA.—The total vertical tail area for biplanes ranges from 5.0 to 5.5% of the total wing area. Monoplanes run slightly higher, averaging about 6.5%.

For most landplanes the proportion of the total vertical area in the rudder varies from 60% to 70%. On seaplanes (float type), the ratio is 50 to 55%, and for flying boats, 40 to 45%. Seaplanes and flying boats require a greater percentage in fixed fin area to offset the keel plane area of the floats or hulls forward of the center of gravity.

AILERONS.—The total area of ailerons ranges from 7 to 12% of the wing area, with average values in the neighborhood of 9 to 10%. Aileron chords range from 15 to 25% of the wing chord, a good average being about 20%. Spans for ailerons range from 35 to 60% of the wing semi-span, averaging about 50%. These figures apply best to monoplanes. For biplanes with ailerons on the upper wing only, the span and chord average slightly above the figures given.

7. STRUCTURAL LOADS AND LOAD FACTORS

Most airplane structures are somewhat specialized space frameworks. They may be analyzed by applying the ordinary principles of mechanics, once the magnitude and location of the loads are known. The structural elements must be designed to withstand the worst of an infinite variety of combinations of loads which may be imposed while the machine is in flight under various conditions, or when approaching or leaving the ground.

CRITICAL LOADING CONDITIONS.—The external forces which act on an airplane are: 1. The air reactions, *i.e.*, lift, drag, etc. 2. Propeller thrust and torque. 3. Ground reaction, *i.e.*, landing and take-off. 4. Weight and inertia of the structure and its contents. A number of loading conditions have been determined to be critical, and always are investigated. These may be classified as: 1. *Main Loading Condition.* *a.* High angle of attack (as in violent maneuvers). *b.* Nose-diving. *c.* Low angle of attack (maximum flight speed). *d.* Inverted flight. 2. *Landing Conditions.* *a.* Three-point landing (wheels and tail-skid). *b.* Level landing (2-point). *c.* Drifted landing (side loading). 3. *Minor Loading Conditions.* *a.* Maximum fin and rudder load. *b.* Maximum stabilizer and elevator loads. *c.* Maximum aileron load. *d.* Nosing over on the ground. *e.* One-wheel landing. *f.* Braked-landing. For detailed rules as to method of analysis for the various conditions, see Aeronautics Bulletin No. 7A, U. S. Dept. of Commerce, Airworthiness Requirements of Air Commerce Regulations for Aircraft.

LOAD FACTORS.—To take care of accelerations in flight and in landing, structures must be analyzed not on the basis of actual weight, but on gross weight, increased by certain factors whose magnitude depends on the service for which the airplane is intended. Thus, U. S. Army specifications (1935) require a factor of 12.0 to be used in analyzing pursuit types for the high angle of attack condition. Factors for commercial aircraft, which are not called upon to do violent acrobatics, fall below these limits. The load factors required by the Department of Commerce for the wing truss at a high angle of attack (customarily the most severe condition) are determined approximately by

$$n = 1.5 + \{V/(3W/S)\}\{4/(3 + 6/R)\} \quad [33]$$

$$n = 1.5 + \{3.75 + 156{,}000/(W + 9200)\}/(W/P)^{0.435} \quad [34]$$

where n = ultimate load factor that must be sustained before failure; W = gross weight, lb.; W/S = wing loading; W/P = power loading; R = aspect ratio; V = maximum speed, mi. per hr. In any particular case, the load factor used is the higher of the two resulting from the separate application of the two formulas. The Department of Commerce rule for load factors in the landing gear and fuselage in landing is

$$n = 4.2 + 13{,}500/(W + 4000) \quad [35]$$

The loads, for which other parts of the structure must be designed, are similarly indicated by formulas especially derived for the conditions there prevailing.

Table 12 lists the approximate range of weights of a variety of airplane materials and accessories. Table 13 lists unit weights for several commonly used materials. Table 14 has been condensed from a study of weights of a number of typical existing airplanes.

LIGHTER-THAN-AIR CRAFT

Balloons and airships derive their lift from simple displacement of the air. A certain degree of dynamic lift may be developed from the forward motion of a power-driven streamlined airship envelope, but, in general, equilibrium conditions are reached at an altitude where the displaced volume of air equals the weight of the aerostat. The absolute altitude of equilibrium varies with atmospheric conditions. (See page 14–98 for definitions and classification of airship types.)

AIRSHIP CHARACTERISTICS.—The external shape of an airship envelope is dictated by structural economics and considerations of parasite drag and of control. Practically, pure streamline shapes of optimum dimensions cannot be used, as it is necessary

(Continued on p. 14–118)

Table 12.—Weights of Miscellaneous Aircraft Parts and Accessories

Low-voltage generators (15–50 amperes)	15–32 lb.
Double-voltage radio generators	21–34 lb.
Dynamotors (including control switch)	6–30 lb.
Generator control boxes	3–5 lb.
Storage batteries (6 volt–65 amperes)	65–70 lb.
Landing lights (per pair)	15–20 lb.
Running lights (set of three)	1 lb.
Transport radio-beacon receiving sets (total installed weight including dynamotor, but excluding battery)	30 lb.
Transport radio, 50-watt, 2-way transmitting and receiving equipment (average installed weight excluding storage batteries)	100–110 lb.
Transport radio-antenna and miscellaneous wiring (extra)	10–15 lb.
Two-way radio for private ships, including dynamotors	45–48 lb.
Starters, hand-inertia	20–30 lb.
" , electric-inertia	27–39 lb.
" , direct cranking electric	16–33 lb.
" , injection type	28–30 lb.
Instruments, common navigation or engine types, complete (each)	1–2 lb.
" , magnetic compasses	2–4 1/2 lb.
" , magneto or earth-inductor compasses	11–12 lb.
" , Sperry (Horizon, Directional Gyro, each)	3 1/2–4 lb.
" , Sperry Gyropilot (installed)	60–75 lb.
Parachute flares, large, with brackets	22 lb.
Shock absorber struts, static loads 1000–7500 lb., each	10–50 lb.
Adjustable airplane chairs, single	12–15 lb.
Washroom equipment, basins, complete	3–5 lb.
" " , 2-3 gal. tanks	4 1/2–5 1/2 lb.
" " , chemical closets	10 lb.
Parachutes (all types)	17–23 lb.

Seaplane floats: Weight $= 0.10 \times (\text{submerged displacement})^{0.92}$

Attachment structure for twin float installation: Weight $= 0.09 \times (\text{gross wt. of seaplane})^{0.85}$

Table 13.—Comparative and Unit Weights of Airplane Materials

Air Corps Information Circular 644

Thickness, in.	Duralumin	Steel	Plywood, 3-ply Birch
0.001	0.014581 lb. per sq. ft.	0.04079 lb. per sq. ft.	......
0.002	0.029162	0.08159	......
0.003	0.043744	0.12238	......
0.004	0.058324	0.16318	......
0.005	0.072905	0.20397	......
0.006	0.087486	0.24477	......
0.007	0.102067	0.28556	......
0.008	0.116648	0.32636	......
0.009	0.131229	0.36715	......
0.010	0.145810	0.40795	0.07437 lb. per sq. ft.
0.020	0.291620	0.81590	0.11083
0.030	0.437440	1.22385	0.14750
0.040	0.583240	1.63180	0.18437
0.050	0.729050	2.03975	0.22083
0.060	0.874860	2.44770	0.25750
0.070	1.020670	2.85560	0.29417
0.080	1.166480	3.26360	0.33083
0.090	1.312290	3.67150	0.36750
0.100	1.458100	4.07950	0.40417
0.200	2.916200	8.15900	0.77083
0.300	4.377400	12.23850	1.13750
0.400	5.832400	16.31800	1.50417
0.500	7.290500	20.39875	1.87083
0.600	8.748600	24.47700	2.23750
0.700	10.206700	28.55650	2.60416
0.800	11.664800	32.63600	2.97083
0.900	13.122900	36.71150	3.33750
1.000	14.581000	40.79500	3.70417

Ratio of Weights

	Duralumin	Steel	Spruce
Duralumin	1.0000	2.7977	0.15430
Steel	0.3574	1.0000	0.05515
Spruce	6.4805	18.1312	1.00000
Wt. of bar 1 sq. in. by 1 ft.	1.2151	3.3996	0.18750

Table 14.—Analysis of Airplane Weights

Weight break-down in percent of total empty weight

Airplane 1E = Single Engine, 3E = Tri-engine, etc.		]—Int. Br. Monoplan	E—Open mercial							-Place ervati Biplai
					Weights, Pounds					
Gross weight	2277	13,100	2690	12,500	17,859	18,560	7350	14,684	2814	3720
Total useful load	700	5,320	991	5,000	6,041	6,510	2835	5,189	816	1293
Total weight empty	1577	7,780	1709	7,500	11,818	12,050	4515	9,495	1998	2427
					Analysis of Weight Empty, Percent (Weight Empty = 100.0 Percent)					
I. Structure, total	53.1	46.4	51.2	62.0	49.7	51.9	47.2	45.5	50.2	55.1
A. Wing group, total	23.9	23.1	23.5	25.7	22.1	21.8	20.5	21.0	17.6	21.4
1. Wings, complete	16.3	22.2	18.3	17.8	15.5	21.3	20.0	20.1	13.9	17.4
2. Ailerons	1.3	0.9	1.7	0.7	1.4	0.5	0.5	0.9	0.9	1.3
3. Struts	6.3	.....	3.5	7.1	5.2	.....	.....	.....	1.5	1.5
4. Wires				0.1					1.3	1.2
B. Tail group, total	3.9	2.2	3.9	2.5	2.1	3.6	3.0	2.3	2.7	2.5
1. Stabilizer	1.4	0.8	1.8	1.2	0.9	1.9	1.6	1.2	1.4	1.0
2. Elevators	1.1	0.6	0.9	0.5	0.4	0.8	0.7	0.4	0.5	0.7
3. Fin	0.4	0.2	0.3	0.4	0.2	0.3	0.4	0.5	0.3	0.3
4. Rudder	0.4	0.3	0.6	0.4	0.4	0.6	0.3	0.2	0.4	0.4
5. Struts and wires	0.6	0.3	0.3	0.4	0.2	.....	.....	.....	0.1	0.1
C. Body group, total	25.3	21.1	23.8	33.4	25.5	26.5	23.7	22.2	29.9	31.2
1. Fuselage, or hull (incl. nacelles)	16.4	12.0	14.8	23.2	18.0	17.4	14.3	12.2	16.9	12.5
2. Landing gear (land)	8.9	9.1	9.0	8.4	7.5	9.1	9.4	10.0	13.0	.....
a. Struts and wires	5.3	2.2	3.0	4.4	2.9	4.5*	4.8	5.2*	4.0	.....
b. Axles		2.5	1.4			0.3				.....
c. Wheels and tires	2.9	4.0	4.1	3.1	3.7	3.4	3.9	4.3	3.1	.....
d. Tail skid or wheel	0.7	0.4	0.5	0.9	0.9	0.9	0.7	0.5	0.6	.....
e. Arresting gear	.....	.....	.....	.....	.....	.....	.....	.....	1.8	.....
f. Flotation gear	.....	.....	.....	.....	.....	.....	.....	.....	3.5	.....
3. Landing gear, water	.....	.....	.....	1.8	.....	.....	.....	.....	.....	18.7
II. Power plant, total	38.8	38.4	38.6	31.4	30.8	29.1	35.7	38.8	45.6	39.7
A. Engine group, total	28.1	30.4	29.4	24.7	16.4	19.9	24.5	24.5	34.5	30.6
1. Engines, dry	26.3	29.6	26.6	21.8	14.5	17.4	21.4	22.2	33.1	28.3
2. Accessories	1.5		2.2	2.3	1.4	2.0	2.6	1.9	1.1	1.9
3. Controls	0.3	0.8	0.6	0.6	0.4	0.5	0.5	0.4	0.3	0.4
B. Propeller, complete	5.4	4.0	4.2	3.3	4.6	5.7	4.7	9.6	4.5	3.7
C. Starting system, total	1.5	1.0	1.6	1.0	1.0	0.7	1.0	0.9	1.7	1.5
D. Lubricating system, total	0.7	0.7	0.7	0.8	0.7	0.5	0.9	0.8	1.4	0.6
1. Tanks	0.4	0.4	0.4	0.3	0.2	0.3	.....	0.5	1.2	0.4
2. Piping and supports	0.3	0.3	0.3	0.5	0.5	0.2	.....	0.3	0.2	0.2
E. Fuel system	3.1	2.3	2.7	1.6	2.5	2.3	4.6	3.0	3.5	3.3
1. Tanks	1.7	1.8	1.9	1.1	1.3	1.7	.....	1.8	2.7	2.3
2. Piping valves, etc	1.4	0.5	0.8	0.5	1.2	0.6	.....	1.2	0.8	1.0
III. Fixed equipment, total	8.1	15.2	10.2	6.6	19.5	19.0	17.1	15.7	4.2	5.2
A. Instruments, total	0.8	0.6	0.9	0.5	0.5	1.3	1.1	2.0	0.6	0.7
B. Surface controls, complete	3.1	1.5	2.2	1.0	2.1	2.1	2.2	3.2	1.6	2.2
C. Furnishings, total	3.7	11.8	6.7	4.8	15.2	12.0	10.0	6.1	1.7	2.0
D. Electrical equipment, total	0.5	1.3	0.4	0.3	1.7	3.6	3.8	4.4	0.3	0.3
					Unit Weights, Pounds					
I. Wing group, per sq. ft	1.55	2.11	1.36	2.63	1.73	2.81	2.55	2.94	1.45	1.63
II. Tail group, per sq. ft	1.07	.....	1.20	1.41	1.05	1.93	1.74	1.63	1.24	1.03
III. Cooling system, per Hp	.....	.....	.....	.....	0.56	.....	.....	.....	.....	.....
IV. Lubricating system per gal. cap	2.24	.....	1.09	4.50	2.08	1.70	1.82	1.27	3.35	2.08
V Fuel system, per gal. cap.	0.96	0.50	0.71	0.39	0.66	0.54	0.64	0.63	0.70	0.92
VI. Exhaust system, per Hp	.....	0.11	0.12	.....	.....	0.10	0.10	0.07	0.03	0.02
VII. Power loading, per Hp	13.40	10.30	12.00	10.00	14.90	13.10	10.00	9.79	6.90	8.30
VIII. Wing loading, per sq. ft	9.40	15.30	9.20	15.80	11.80	19.80	20.20	21.66	11.20	11.70

* Retracting landing gear, includes weight of retracting mechanism.

to modify the structure to attach control cars, power-plant housings, control surfaces, etc. For most practical purposes, the volume of the average airship envelope may be estimated from $V = \frac{1}{2}LD^2$, where V = volume, L = length, and D = maximum diameter. Under standard atmospheric conditions at sea-level, buoyancy may be taken as 64 lb. per 1000 cu. ft. for hydrogen and 58 lb. per 1000 cu. ft. for helium. In a rigid ship, with a multiplicity of gas cells, the gas volume is usually about 5% less than the volume enclosed by the outer cover.

For rigid types the fineness ratio (length/max. diam.) is tending downward. Early ships exhibited ratios of $8\frac{1}{2}$ or 9; present practice (1935), as indicated by the Akron-Macon class, reduces the ratio to 6. On non-rigid ships, structural factors favor a lower fineness ratio; it ranges between 3 and 5. Comparative statistics of three of the best known rigid airships of recent design are given in Table 1.

Table 1.—Characteristics of Typical Rigid Airships

	Los Angeles	Graf Zeppelin	Akron and Macon
Overall length, ft.	658	776	785
Maximum diameter, ft. .	91	100	133
Gas volume, cu. ft.	2,470,000	3,700,000	6,500,000
Gross lift, lb.	153,000	258,000	403,000
Total Hp.	2,000	2,750	4,480
Maximum speed, m.p.h. .	73	80	84

AIRSHIP PERFORMANCE.—The maximum speed of an airship depends simply upon its coefficient of aerodynamic resistance, complete with tail surfaces and cars. The coefficient commonly is stated in terms of the resistance per unit of volume of the airship envelope, rather than in terms of frontal area, as with airplane parts. Total resistance varies as the two-thirds power of the volume. Thus, $R = Cv^{2/3}V^2$, v being the volume of the envelope, cu. ft., and V the velocity, mi. per hr. Using the customary mi.-per-hr. and sq. ft. units, the value of C for the best rigid airship forms falls as low as 0.000035 for the bare envelope, or 0.00006 for a complete ship. Knowing the propeller efficiency, the power required for a given speed can be calculated and a maximum-speed formula obtained in the form $V_{max} = K\sqrt[3]{P/v^{2/3}} = K\sqrt[3]{P/v}(v^{1/9})$, where P = engine power and v = the volume. K attains a maximum value of approximately 165, and can be taken as 165 for large rigid airships of good design, and as 125 for typical non-rigid ships. Unlike airplanes, airships show an inherent increase in performance with increasing size. If the power and volume and lift all be doubled the maximum speed will, other things being equal, be increased by 8%.

The ceiling attainable with an airship depends almost solely on the relation between the minimum fixed, or non-disposable, weight of the ship and the total lift that would exist at sea-level if all the gas cells were filled to the limit. In attaining high altitudes the airship acts as a free balloon, and its altitude performance follows free-balloon laws. The ship will rise until the gas in the partly-filled cells has expanded to fill them completely, and then continue to rise until the density of the surrounding air has decreased to a point where the ascensional force, or difference between the weight of air displaced by the ship and the weight of the gas doing the displacing, is barely equal to the solid weight of the ship at that moment. The altitude that can be attained is given approximately, including an allowance of 1000 ft. for the effect of dynamic lift in carrying the ship above the level of static equilibrium, by $H = 29{,}000(1.75 - \sqrt{4(W_2/W_1) - 1})$, where W_1 = maximum lift of airship at sea-level with gas cells full; W_2 = total weight that has to be carried to the highest altitude.

The range of an airship in still air increases steadily with declining speed, the lowest speed always being the most economical under that condition. Other things being equal it increases, like maximum speed, with increasing size. An approximate range formula, for ships using liquid fuel exclusively and inflated with helium, is

$$X = 21{,}000(W_f/W_1) \times \{\sqrt[3]{(v \times 10^{-6})}/(V/50)^2\}$$

where X = range, miles; W = maximum lift, lb. at sea-level; W_f = weight of fuel carried, lb.; and V = cruising speed mi. per hr. For a hydrogen-filled ship the constant is increased to 24,000, but W_f/W_1 also is increased, as the extra lift of the lighter gas can all be put into extra fuel. The final result of replacing helium with hydrogen is likely to be an increase of range at a given speed approximately 40%.

Section 15

ELECTRIC POWER

By E. S. Dibble

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ELECTRIC POWER

By E. S. Dibble

PURCHASED POWER

The factors influencing choice of electric current for industrial applications are summarized below.

ALTERNATING CURRENT (A.C.) can be transmitted economically over long distances and utilized by transformation of its voltage. A.C. motors and generators are available for practically all types of work, and are less expensive and of simpler construction than D.C. machines. Purchased power usually is alternating current, and conversion or local generating equipment is necessary if direct current is needed. Alternating current is generated at 25, 50 and 60 cycles. The latter is the most common as it is satisfactory for lighting, and higher speed motors may be used than with 25- or 50-cycle current. Equipment generally is standardized on 60 cycles.

DIRECT CURRENT (D.C.) is required in large amounts for some processes, as electrolytic, certain types of welding, and a few motor applications, such as variable speed drives with fine speed adjustment. Some driven machines have torque characteristics that are met only by D.C. motors. The character of plant load and relative amount of direct current needed determine whether it shall be generated as direct current or converted from alternating current at the point of use. If the amount of direct current is relatively small, the latter course generally is advisable.

1. PLANT POWER SUPPLY

The usual sources of power supply for an industrial plant are: 1. Purchased power; 2. Private generating plant; 3. Private generating plant supplementing purchased power.

PURCHASED POWER.—Rates will vary in different localities, and with the amount of power used. In a given locality, a group of consumers usually will have lower rates than a single consumer. Factors influencing low rates are: Large energy consumption; proximity to distribution lines; constant load; high power factor.

Reliability.—Efficiency of large generating stations is higher than that of the smaller stations of average industrial plants. Duplicate power lines and extensive interconnection of high-voltage distribution systems make remote the chances of serious interruption of power.

Isolated Locations.—Rates will be high and reliability low when relatively small amounts of power are transmitted over a single long line to one or a group of industries. Duplicate lines would increase reliability but would also increase cost. A private generating station, under such conditions, may be warranted, especially for continuous-process industries where reliability is of first importance.

PRIVATE GENERATING STATIONS.—**Low-pressure Process Steam** is required by textile, sugar and paper mills, glue factories, dye works, chemical plants and other industries. A complete steam plant is necessary, which can, by minor changes in design, produce the same weight of steam at a pressure sufficient to operate turbo-generators, the exhaust being used as process steam. With boiler and exhaust pressures of 250 lb., gage, and 15 lb., gage, respectively, about 1 kw.-hr. may be obtained per 30 lb. of steam. That is, process steam requirements of 30,000 lb. per hour would make available 1000 kw. of electricity, at a fuel cost equal to the difference in fuel required to produce equal weights of steam at 250 lb. and 15 lb., gage.

Waste Heat or Waste Fuel may be available as by-products in such industries as smelting and refining. Electricity generated from such by-products, either by means of steam produced in waste heat boilers, or by using the fuel in Diesel or gas engines, has little or no fuel cost chargeable to it. The total cost of boilers, etc., however, must be included in the cost of electrical equipment.

2. SUBSTATIONS

THE MAIN SUBSTATION reduces transmission line voltage to a voltage suitable for industrial plant use. Common public utility line voltages are 13,200, 11,000, 6600, 4000, and 2300 volts. Usual industrial plant voltages for motors are 220, 440, and 550, and for lighting, 115 to 120 volts. Some large industries, with extensive distribution within the plant, may take current at 6600 volts. The substation also may include conversion equipment to provide D.C. power or A.C. power of a different frequency.

Ownership of the Substation usually rests with the public utility, which operates and maintains it.

Location of the Substation depends on the relative location of the customer and the public utility transmission lines. On high-voltage lines, it probably will be located at the point where the lines are tapped. For a short extension of an existing line operating at 22,000 volts or lower, the sub-station would be placed at the receiving end of the extension.

INDOOR AND OUTDOOR SUBSTATIONS.—Outdoor substations are used where conditions permit. They have the following advantages: Reduced fire hazard from oil-filled transformers and circuit breakers; easier and less expensive separation of high-voltage bus structures; small difference in cost of outdoor and indoor equipment above 15,000 volts. Most industrial substations are simply transformer stations with static equipment only, located outdoors on the consumer's premises. In congested areas, substation equipment must be indoors. It may be placed in a room or corner of an existing building, if Code requirements are met.

Substation Capacity must be sufficient to carry peak load without excessive voltage variation. Transformer ratings or the ratings of rotating conversion equipment will be governed by the load characteristics.

Demand Factor is the ratio (actual maximum demand/connected load). It is used in determining capacity of equipment, and in fixing rates for power supplied by public utilities. Demand factors, determined by actual measurement with maximum demand meters, have been tabulated by public utilities for different classes of industries.

Example.—The demand factor for industries working sheet metal is about 70%. The probable maximum demand of a sheet metal manufacturer, whose connected load is 1000 kw., would be 1000 × 0.70 = 700 kw. Substation equipment would be installed on this basis. The time interval over which measurement has been taken should be specified.

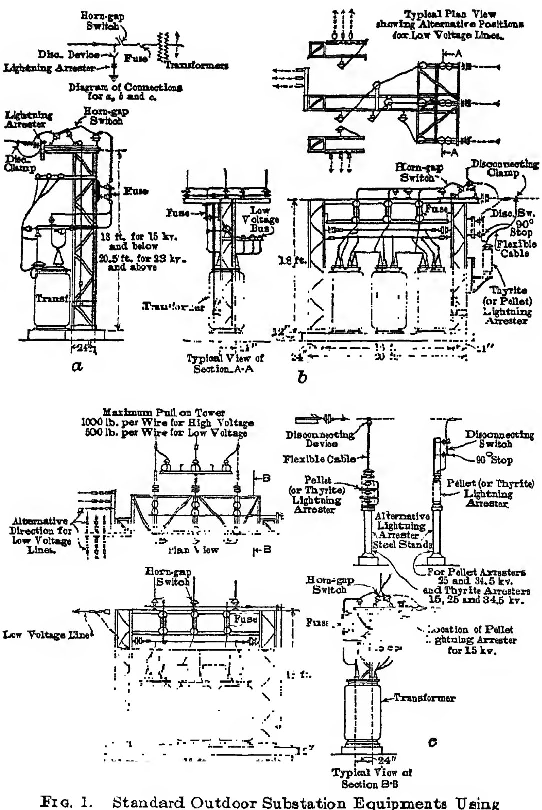


Fig. 1. Standard Outdoor Substation Equipments Using High-voltage Fuses

Load Factor is the ratio (average power ÷ peak power). It also is used in fixing rates, and in determining capacity of equipment. The interval of maximum load must be specified, and also the period over which load is averaged, as a 15-min. monthly, or a 30-min. yearly load factor. Continuous-rated equipment is installed when load factor is at or near unity. Overload capacity is relied on to carry peak loads of short duration, occurring only occasionally.

Switching Arrangements and Number of Circuits in the substation depend on the nature of the service and probability of system disturbance. Power interruptions of short duration, causing only loss in production and labor are not serious, but they are serious in industries using batch or continuous processes. In the latter case insurance of the continuity of service will warrant considerable expense.

MAIN SUBSTATION EQUIPMENT is essentially the same for outdoor substations served by an overhead line, or indoor substations served by underground cable. Fig. 1 shows typical standard substation equipment for outdoor use. The range of capacities of the arrangements shown are as follows:

Fig. No.	1*a*	1*b*, 1*c*
Transformer capacity, kva.	75–1500	150–1500
Maximum primary voltage	34,500	,
Secondary volts, up to 450 kva.	440 or 2300	440 or 2300
Secondary volts, over 450 kva.	2300	2300

A single equipment includes: One each of galvanized steel structure, galvanized steel bracket for distribution-type arresters for 23,000 to 34,500 volts; horn gap switch with manual operating mechanism; three each of fuse disconnecting switches, disconnecting clamps for isolating distribution-type arresters, or disconnecting switches for isolating station-type arresters, distribution-type or station-type lightning arresters.

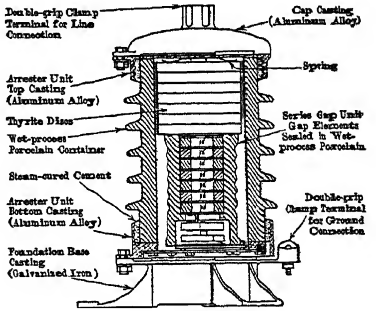


Fig. 2. Thyrite Arrester Unit

Substation Metering Equipment comprises current and potential transformers connected on the high- or low-voltage side of the transformer bank, depending on who is to be charged with transformer losses, and metering instruments. The latter comprise arresters, voltmeter, watt-hour meter and reactive-volt-ampere-hour meter, located either outdoors or on an incoming line panel.

Horn-gap Switches will interrupt the magnetizing current of the transformer bank, and, in emergencies, may interrupt load current. They are not recommended for load-break switches or for breaking heavy line charging current. For such purposes, an oil circuit breaker would be used with disconnecting switches.

3. LIGHTNING ARRESTERS

Lightning arresters are necessary only for exposed lines, and should be placed close to the equipment to be protected. Distribution-type arresters are used for transformer banks up to 1000 kva., and station-type arresters for banks over 1000 kva.

THYRITE ARRESTERS.—Station-type Arresters consist of stacks of arrester units, the number depending on the voltage of the circuit. Table 1 shows the maximum and minimum arrester voltage ratings and the corresponding rated circuit voltages at which these arresters should be used.

Table 1.—Thyrite Station-type Lightning Arrester Ratings

Rated circuit voltage	2300	4600	6900	11,500	13,800	23,000
Rated arrester voltage	1000*	3000*	6000*	9,000*	9,000*	23,000
Max. arrester voltage	3000	6000	9000	15,000	15,000	25,000

* Minimum voltage rating.

Thyrite Unit Arresters comprise discs of thyrite material, assembled as in Fig. 2. Thyrite is an insulator at one voltage and a conductor at a higher voltage, its resistance being a function of voltage only. Doubling the applied voltage will increase a given current I to about $12.5\ I$.

Thyrite Bushing Arresters are mounted inside the bushings of pole-type, 2400- and 4800-volt distribution transformers, rated up to 100 kva. The arresters are so connected that they shunt the high-voltage and low-voltage insulations. Lightning discharges are dissipated through ground on the low-voltage side, or to the transformer tank. Protection is obtained at a saving of 40 to 60 percent in cost, as compared with externally mounted arresters. Fig. 3 shows single-phase connections.

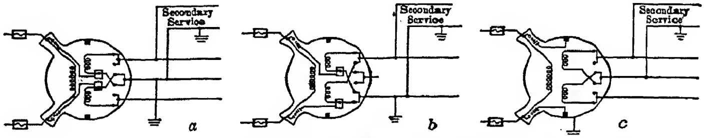


FIG. 3. Connections for Bushing Arresters for Distribution Transformers

DISTRIBUTION LIGHTNING ARRESTERS.—**Pellet-type Arresters** are used to protect distribution transformers, medium-size substations, and cable terminals. Fig. 4 shows construction for voltages up to 15,000 kva. Fig. 5 illustrates typical connections for protecting distribution circuits and transformers. An installation is shown in Fig. 7.

Diagram *a* shows 3-phase and single-phase applications to 3-phase ungrounded-neutral systems or systems with the neutral grounded through resistance or reactance. The rating of the arresters is selected on the basis of line-to-line voltage. Diagram *b* shows 3-phase and single-phase applications to 3-phase, 3-wire, solidly grounded neutral systems. Diagram *c* shows 3-phase and single-phase applications to 3-phase, 4-wire, solidly grounded neutral system with the neutral wire carried out. The rating of the arresters, on the main circuit wires in *b* and *c*, can be less than line-to-line voltage, usually of the next lower rating to that used on delta or ungrounded neutral systems. The maximum rating of the arrester on the neutral of *c* is equal to, or higher than, the potential from neutral to ground due to any phase unbalancing. The ground on the neutral at the power source does not in any way reduce the lightning potentials occurring on the neutral wire out on the circuit. It is just as important to apply arresters to the neutral as to the phase wires, where the neutral is connected to the apparatus.

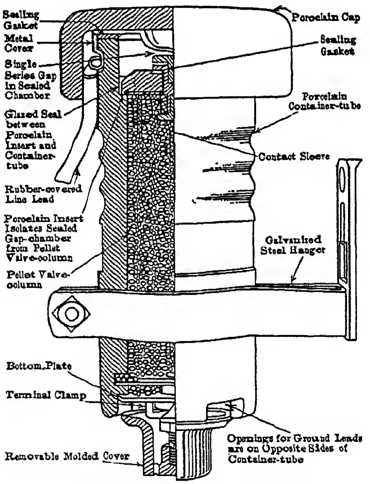


FIG. 4. Pellet-type Lightning Arrester

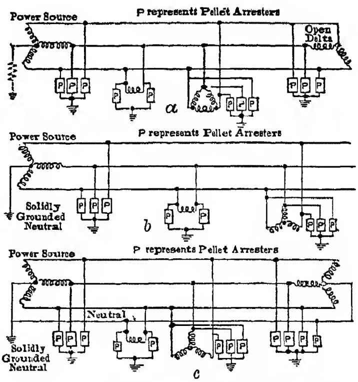


FIG. 5. Application of Lightning Arresters

Compression Chamber Arresters can be used on A.C. circuits up to 750 volts, which includes secondary lighting and power circuits.

4. DISCONNECTING EQUIPMENT

DISCONNECTING SWITCHES are used for isolating equipment, for bus sectionalizing, for transfer switching and for line sectionalizing. Standard sizes of indoor types are 200, 400, 600, 1200, 2000 and 3000 amperes, mounted on insulating bases for 5000-volt circuits, and on insulators for higher voltage. They are made single-throw, double-throw, front- or rear-connected, as required by the installation. Fig. 6 shows a typical outdoor switch with manual operating mechanism.

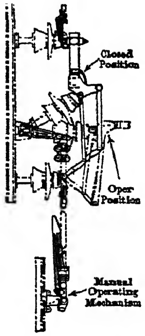


FIG. 6. Typical Outdoor Disconnecting Switch

SUBSTATION CIRCUIT BREAKERS.—For small industrial plants a substation may consist of a distribution transformer and its protective equipment. See Fig. 7. Usual voltage transformation for this type of station is 2300 to 220/440 volts. Fuse cutouts give full protection against short circuits and oil switches are unnecessary. However, automatic switching and interruptions of circuits under load require oil circuit breakers. Duplicate lines feeding the substation require oil circuit breakers to provide selective switching to clear faults and insure continuity of service.

OIL CIRCUIT BREAKERS are designed to interrupt circuits under load and provide protection against short circuit currents by making and breaking contacts under oil. Rating is based on normal continuous current in the circuit, maximum current that can be carried for a short time without mechanical failure and maximum current that can be interrupted on short circuit.

Classification.—1. Service, indoor or outdoor. 2. Operating method, manual or electric. 3. Method of mounting. *a.* On back of panel; manual operation recommended if voltage does not exceed 5000 volts between lines. *b.* Alone on panel frame to match and line up with existing installation. *c.* Remotely on frame, either manually or electrically operated. *d.* In cubicles, for increased safety and decreased fire hazard.

Manual Operation of large breakers is not feasible because of the excessive thrust required. Small and medium size breakers usually are operated by a lever on the front of the panel.

Electrical Operation is used for breakers remotely located, or too heavy for manual operation. Either a solenoid or a motor can be used. Coils of standard solenoid-operated mechanisms are wound for 48, 125 and 250 volts. Low-voltage operating coils are not recommended for large breakers.

Methods of Tripping.—1. Over-current trip coils, operated from current transformers, will meet requirements of most industrial plants. For large industrial plants, with more complicated systems of control, and those generating their own power, relays will insure more accurate tripping. 2. Circuit opening relays, operated from current transformers, giving more accurate time delay in tripping on over-current, which opens contacts and permits current to flow in trip coils. Relays which close the trip circuit will give under-voltage or over-voltage tripping.

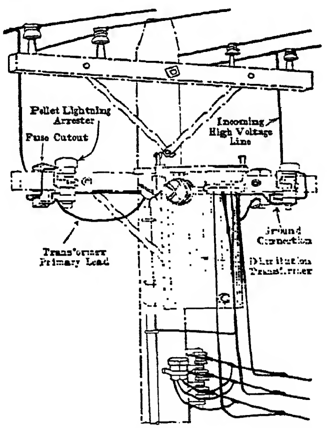


FIG. 7. Typical Distribution Substation for Small Power Users

SELECTION OF OIL CIRCUIT BREAKERS depends on: 1. Normal voltage of circuit in which breaker is to operate, defined as highest rated voltage of the secondary of transformers supplying the system. 2. Normal root-mean-square current of the circuit. 3. Magnitude of short circuits to which breaker will be subjected. 4. Root-mean-square value of short circuit currents. 5. Operating time of relaying devices and breaker-operating mechanisms, determining the interrupting duty under a given condition. 6. Altitude at which breaker is to operate, guiding proper selection of high-voltage bushings, the amount of insulation required varying with altitude. 7. Lowest temperature to which breaker is subjected, enabling proper selection of oil.

OIL CIRCUIT BREAKER RATINGS.—**Interrupting Rating** is determined by the highest root-mean-square amperes that the breaker successfully will interrupt at any specified voltage and for a specified duty cycle. See Table 2.

Short Time Rating is determined by the highest root-mean-square amperes that breaker will carry for a specified time without damage.

Table 2.—Oil Circuit Breaker Characteristics

Volts *	Rated Amperes 60 Cycles	Rated Amperes 25 Cycles	Ratings in Root-mean-square Total Amperes: Short-time Current 1 Sec.	Short-time Current 5 Sec.	Interrupting Max. Rating	Interrupting At Rated Voltage	Interrupting Rating in 3-phase, kva. at Rated Voltage	Weight, lb.	Method of Operation †	Method of Mounting †
Indoor Breakers with OCO-2 min.-OCO Duty Cycle‡										
5,000	200	200	15,000	10,000	12,000	2,400	20,000	100	*H*	*P*
5,000	400	400	15,000	10,000				110	*H*	*P*
7,500	600	700	20,000	20,000	20,000	2,000	25,000	210	*H*	*P*
2,500	800	950	20,000	20,000	20,000	6,000	25,000	260	*H*	*P*
600	3000	4000	80,000	80,000	40,000	40,000	40,000	3,400	*M*	*F*
600	4000	5000	100,000	100,000	40,000	40,000	40,000	5,200	*M*	*F*
600	5000	5000	100,000	100,000	40,000	40,000	40,000	5,400	*M*	*F*
15,000	600	700	25,000	25,000	20,000	2,000	50,000	550	*H*	*P*
7,500	1200	1400	35,000	35,000	20,000	4,000	50,000	630	*H*	*P*
5,000	2000	2250	40,000	40,000	20,000	6,000	50,000	1,280	*H*	*P* or *F*
Indoor Breakers with OCO-15 sec.-OCO Duty Cycle‡										
15,000	600	700	40,000	30,000	20,000	2,000	50,000	550	H_d or H_r	*F* or *C*
7,500	1200	1400	40,000	40,000	20,000	4,000	50,000	630	H_d or H_r	*F* or *C*
5,000	2000	2250	40,000	40,000	20,000	6,000	50,000	1,120	H_d or H_r	*F* or *C*
15,000	600	700	50,000	30,000	30,000	4,000	100,000	1,100	H_d or H_r	*F* or *C*
15,000	1200	1400	60,000	50,000	30,000	4,000	100,000	1,120	H_d or H_r	*F* or *C*
7,500	2000	2250	60,000	50,000	30,000	8,000	100,000	1,180	H_d or H_r	*F* or *C*
15,000	600	700	60,000	50,000	30,000	4,000	100,000	1,180	H_d or H_r	*F* or *C*
15,000	1200	1400	60,000	50,000				1,210	H_d or H_r	*F* or *C*
15,000	600	700	50,000	30,000	36,000	6,000	150,000	1,850	*E*	*F* or *C*
15,000	1200	1400	60,000	50,000				2,000	*E*	*F* or *C*
15,000	600	700	50,000	30,000	50,000	10,000	250,000	1,950	*E*	*F* or *C*
15,000	1200	1400	60,000	50,000	60,000			2,100	*E*	*F* or *C*
15,000	600	700	50,000	30,000	36,000	6,000	150,000	1,080	H_r	*F* or *C*
15,000	1200	1400	60,000	50,000				1,140	H_r	*F* or *C*
15,000	600	700	50,000	30,000	50,000	10,000	250,000	1,100	H_r	*F* or *C*
15,000	1200	1400	60,000	50,000	60,000			1,170	H_r	*F* or *C*
15,000	2000	2250	70,000	60,000	60,000			2,300	H_r	*F* or *C*
15,000	600	700	50,000	30,000	50,000	20,000	500,000	2,060	H_r	*F* or *C*
15,000	1200	1400	60,000	50,000	60,000			2,280	H_r	*F* or *C*
15,000	2000	2250	80,000	80,000	80,000			2,380	H_r	*F* or *C*
23,000	600	700	50,000	30,000	25,000	12,500	500,000	6,900	*S*	*F* or *C*
23,000	1200	1400	60,000	50,000				6,900	*S*	*F* or *C*
23,000	2000	2250	60,000	60,000				10,100	*S*	*F* or *C*
Outdoor Breakers with OCO-2 min.-OCO Duty Cycle‡										
7,500	200	200	5,000	5,000	3,000	240	3,000	140	*H*	*V*
5,000	100	100	5,000	5,000	3,000	360	3,000	175	*H*	*L*
7,500	200	200	7,000	7,000	5,100	400	5,000	250	*H*	*L*
15,000	200	200	7,000	7,000	6,200	240	6,000	350	*H*	*L*
5,000	400	400	20,000	15,000	12,000	3,000	25,000	1,125	*H* or *S*	*F*
5,000	600	700	25,000	20,000				1,125	*H* or *S*	*F*
Outdoor Breakers with OCO-15 sec.-OCO Duty Cycle‡										
7,500	400	400	20,000	15,000	12,000	4,000	50,000	1,150	*H* or *S*	*F*
7,500	600	700	25,000	20,000				1,150	*H* or *S*	*F*
15,000	600	700	40,000	30,000	12,000	2,000	50,000	1,780	*H*	*P*
15,000	600	700	40,000	30,000	24,000	4,000	100,000	2,060	*H*	*P*
15,000	600	700	40,000	30,000	40,000	7,000	175,000	2,150	*H*	*P*
15,000	1200	1400	40,000	30,000				2,550	*H*	*P*
23,000	600	700	50,000	30,000	40,000	12,500	500,000	6,955	*S*	*F*
23,000	1200	1400	60,000	50,000	50,000			6,955	*S*	*F*
23,000	2000	2250	60,000	60,000	60,000			10,120	*S*	*F*

* Items bracketed together represent the same type of breaker.

† *C* = cell; *E* = electrical; *F* = frame; *H* = manual; H_d = direct manual; H_r = remote manual; *L* = pole; *P* = panel; *S* = direct-current solenoid; *V* = on vertical flat surface.

‡ The duty cycle on which standard interrupting ratings of oil circuit breakers are based assumes that the breakers will interrupt a circuit under the conditions imposed by two unit operations. Each unit operation consists of closing the circuit breaker, followed immediately by its opening without purposely delayed action. The standard duty cycle of oil circuit breakers is based on two unit operations with 15 seconds interval, except for non-oil-tight breakers of 50,000 kva. rating and below, which have a standard duty cycle consisting of two unit operations with a two minute interval. For example, OCO is a unit operation. Beginning with the breaker open, it is closed and opened, and after a two minute time interval the unit operation is repeated.

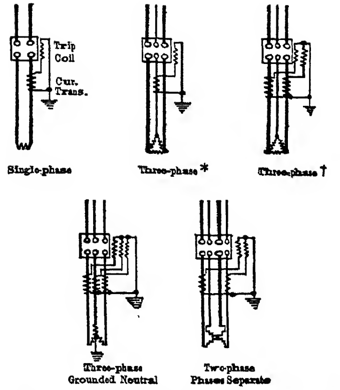


FIG. 8. Circuit Breaker Tripping Connections with Tripping Coils Operating in Secondary of Current Transformers ‡

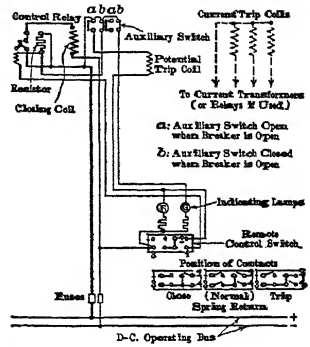


FIG. 10. Typical Control Connections for Remote-control D.C. Solenoid Circuit Breaker Operating Mechanisms

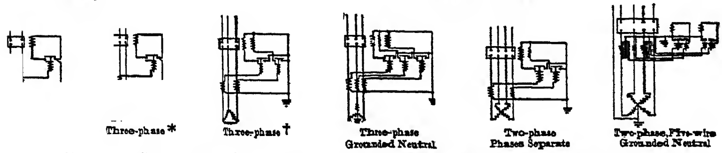


FIG. 9. Circuit Breaker Tripping Connections Using Overcurrent Relays ‡

COST OF OIL CIRCUIT BREAKERS depends on operating voltage, current rating, method of operation, and system short-circuit values. Accurate estimates of costs cannot be made until these factors have been investigated. See Table 2 for typical ratings. Manufacturers should be consulted in regard to circuit breakers to meet specific conditions.

CIRCUIT BREAKER CONNECTIONS.—Figs. 8, 9, 10 show recommended connections for trip coils operating with current transformers, relays or other tripping devices. Power storage batteries are advised for complicated relaying in connection with oil circuit

* Used for railway service only, to protect A.C. motors, synchronous converters, transformers or incoming lines.

† Always provide three over-current coils if circuit is served by transformers whose primaries are connected in Y with neutral not connected to system or grounded.

‡ For incoming service lines, where Underwriters' rules apply, an over-current unit must be provided in each ungrounded conductor.

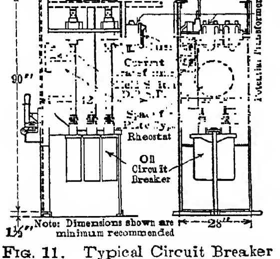


FIG. 11. Typical Circuit Breaker Installation for Circuits up to 2500 Volts

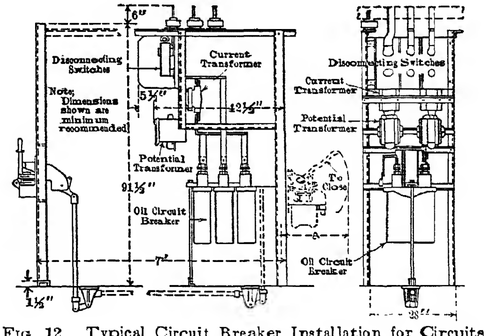


FIG. 12. Typical Circuit Breaker Installation for Circuits up to 5000 Volts

breakers using a separate D.C. source of tripping. Suggested arrangements for indoor type circuit breakers for maximum values of 2500 and 5000 volts respectively are shown in Figs. 11 and 12.

5. TRANSFORMERS

A transformer consists essentially of two insulated coils assembled on an iron core, and immersed in oil in a tank. In small sizes, coil and core may be exposed directly to the air. The coils are designated primary and secondary, or high-voltage and low-voltage. Sometimes more than two windings are involved. A.C. voltage, applied to one coil, will induce in the other coil a voltage which will be in the same ratio to the primary voltage as the ratio of the numbers of turns in the two windings. Thus, if 1000 volts be applied to the high-voltage coil of a transformer in which the ratio of high- and low-voltage turns is 10 : 1, voltage induced in the low-voltage coil will be $1000 \times (1/10) = 100$ volts. The function of a transformer is to receive power at one voltage and deliver it at another.

POWER TRANSFORMERS comprise ratings above 500 kva., 25 and 60 cycles. See Table 3. Standard voltage ratings conform to preferred ratings as covered by N.E.L.A. and N.E.M.A. publications, April, 1930. Tables 4 and 5 cover power transformers for substation and general use.

Table 3.—Standard Power Transformer Sizes

25 and 60 cycles, oil-immersed.

Kva, single-phase						Kva, 3-phase					
Self-cooled				Water-cooled		Self-cooled				Water-cooled	
667	2000	6,667	20,000	5,000	16,667	600	2000	6,000	20,000	5,000	15,000
833	2500	8,333	25,000	6,667	20,000	750	2500	7,500	25,000	6,000	20,000
1000	3333	10,000	33,333	8,333	25,000	1000	3000	10,000	30,000	7,500	25,000
1250	4000	12,500	.....	10,000	33,333	1200	3750	12,000	37,500	10,000	30,000
1667	5000	16,667	.....	12,500	.....	1500	5000	15,000	.....	12,000	37,500

Table 4.—Substation and General Purpose Single-phase Transformers for Motor Circuits

See notes, p. 15–12

Nominal System Voltage Note 1	Minimum Standard kva.	Rated High Voltage Note 2	High Voltage Taps Full Capacity Note 3		Nominal Low Voltage (Note 4)					
					220 or 440	550	2300 or 4000	6,600 or 11,000	11,000	13,200
					Maximum Standard kva.					
					240/480	1000	5000	No Limit	No Limit	No Limit
					Rated Low Voltage					
2,300 * and 4,000 *	667	2,400/4,160 Y	2,280	2,160	240/480	600	....	....	......	......
4,600	667	2,400/4,800	2,280 4,560	2,160 4,320	240/480	600	....	....	......	......
6,600 and 11,000	667	6,900/11,950 Y	6,555	6,210	230/460	575	....	....	......	......
	667	6,600/11,430 Y	6,270	5,940	....	....	2300	....	......	......
	667	11,500	10,925	10,350	230/460	575	....	....	......	......
	667	11,000	10,450	9,900	....	....	2300/4000 Y	....	......	......
13,200 *	667	13,200	12,540	11,880	240/480	600	2400/4160 Y	....	......	......
	667	7,620/13,200 Y	7,240	6,860	240/480	600	2400	....	......	......
22,000	667	22,000	20,900	19,800	240/480	600	2400/4160 Y	6,900/11,950 Y	11,500	13,800
	667	12,700/22,000 Y	12,070	11,430	240/480	600	2400	6,900	11,500	13,800
33,000 *	667	33,000	31,350	29,700	240/480	600	2400/4160 Y	6,900/11,950 Y	11,500	13,800
	667	19,050/33,000 Y	18,100	17,150	240/480	600	2400	6,900	11,500	13,800
44,000	667	44,000	41,800	39,600	....	....	2400/4160 Y	6.900/11,950 Y	11,500	13,800
	667	25,400/44,000 Y	24,140	22,860	....	....	2400	6,900	11,500	13,800
66,000 *	667	66,000	62,700	59,400	....	....	2400/4160 Y	6,900/11,950 Y	11,500	13,800
	667	38,100/66,000 Y	36,200	34,290	....	....	2400	6,900	11,500	13,800
110,000	667	110,000	104,500	99,000	....	....	2400/4160 Y	6,900/11,950 Y	11,500	13,800
	667	63,500/110,000 Y	60,330	57,150	....	....	2400	6,900	11,500	13,800
132,000 *	1,000	132,000	125,400	118,800	....	....	2400/4160 Y	6,900/11,950 Y	11,500	13,800
	1,000	76,200/132,000 Y	72,390	68,580	....	....	2400	6,900	11,500	13,800
154,000	2,500	154,000	146,300	138,600	....	....	2400/4160 Y	6,900/11,950 Y	11,500	13,800
	2,500	89,000/154,000 Y	84,550	80,100	....	....	2400	6,900	11,500	13,800
220,000 *	10,000	220,000	209,000	198,000	....	....	....	6,900/11,950 Y	11,500	13,800
	10,000	127,000/220,000 Y	120,700	114,300	....	....	....	6,900	11,500	13,800

Table 5.—Substation and General Purpose 3-phase Transformers for Motor Circuits

See notes, p. 15–12

Nominal System Voltage	Minimum Standard kva.	Rated High Voltage	High Voltage Taps Full Capacity		Nominal Low Voltage (Note 4)					
					220 or 440	550	2300 or 4000	6,600 or 11,000	11,000	13,200
					Maixmum Standard kva.					
Note 1		Note 2	Note 3		3000	3000	15,000	15,000	No Limit	No Limit
					Rated Low Voltage					
2,300* and 4,000*	600	2,400/4,160 Y	2,280	2,160	240/480	600	...	...	...	...
4,600	600	4,800 Y	4,560	4,320	240/480	600	...	...	...	...
6,600	600	6,900 Y	6,555	6,210	230/460	575	...	...	...	...
	600	6,600 Y	6,270	5,940	...	...	2300	...	...	...
	600	6,600 Delta	6,270	5,940	...	...	2300/4000 Y	...	...	...
11,000	600	11,500 Y	10,925	10,350	230/460	575	...	...	...	...
	600	11,000 Y	10,450	9,900	...	...	2300	...	...	...
	600	11,000 Delta	10,450	9,900	...	...	2300/4000 Y	...	...	...
13,200*	600	13,200 Y	12,540	11,880	240/480	600	2400	...	...	...
	600	13,200 Delta	12,540	11,800	...	...	2400/4160 Y	...	...	...
22,000	600	22,000 Y	20,900	19,800	240/480	600	2400	6,900	11,500	13,800
	600	22,000 Delta	20,900	19,800	...	...	2400/4160 Y	5,900/11,950 Y	11,500	13,800
33,000*	600	33,000 Y	31,350	29,700	240/480	600	2400	6,900	11,500	13,800
	600	33,000 Delta	31,350	29,700	...	...	2400/4160 Y	6,900/11,950 Y	11,500	13,800
44,000	600	44,000 Y	41,800	39,600	...	...	2400	6,900	11,500	13,800
	600	44,000 Delta	41,800	39,600	...	...	2400/4160 Y	6,900/11,950 Y	11,500	13,800
66,000*	600	66,000 Y	62,700	59,400	...	...	2400	6,900	11,500	13,800
	600	66,000 Delta	62,700	59,400	...	...	2400/4160 Y	6,900/11,950 Y	11,500	13,800
110,000	600	110,000 Y	104,500	99,000	...	...	2400	6,900	11,500	13,800
	600	110,000 Delta	104,500	99,000	...	...	2400/4160 Y	6,900/11,950 Y	11,500	13,800
132,000*	1,500	132,000 Y	125,400	118,800	...	...	2400	6,900	11,500	13,800
	1,500	132,000 Delta	125,400	118,800	...	...	2400/4160 Y	6,900/11,950 Y	11,500	13,800
154 000	5,000	154 000 Y	146,300	138,600	...	...	2400	6,900	11,500	13,800
	5,000	154,000 Delta	146,300	138,600	...	...	2400/4160 Y	6,900/11,950 Y	11,500	13,800
220,000*	15,000	220,000 Y	209,000	198,000	...	...	...	6,900	11,500	13,800
	15,000	220,000 Delta	209,000	198,000	...	...	...	6,900/11,950 Y	11,500	13,800

Table 6.—Standard Single-phase Distribution Transformers, Oil-immersed, Self-cooled

See notes, p. 15–12

Nominal System Voltage Note 1	Standard kva. Sizes for Each Voltage Class Note 5	Rated High Voltage Note 2	High Voltage Taps Note 3 Full Capacity		Reduced Capacity	Rated Low Voltages Note 6		
440*	1.5, 3 to 200 incl.	480	456	432	...	120/240	...	...
550	1.5, 3 to 200 incl.	600	570	540	...	120/240	...	...
2,300* and 4,000*	1.5, 3 to 50 incl.	2,400/4,160 Y	...	...	...	120/240	240/480 600	...
		2,500/4,330 Y	...	...	...	125/250	...	...
	75 to 500 incl.	2,400/4,160 Y	2,280	2,160	...	120/240	240/480 600	...
		2,500/4,330 Y	2,375	2,250	...	125/250	...	...
4,600	1.5, 3 to 100 incl.	2,400/4,800	...	...	...	120/240	240/480 600	...
		2,500/5,000	...	...	...	125/250	...	...
	150 to 500 incl.	2,400/4,800	2,280	2,160	...	120/240	240/480 600	...
			4,560	4,320	...	...	...	...
		2,500/5,000	2,375	2,250	...	125/250	...	...
			4,750	4,500	...		...	...
6,600 and 11,000	1.5, 3 to 500 incl.	6,900/11,950 Y	6,585	6,275	5,960	115/230	230/460 575	...
		7,200/12,470 Y	6,875	6,545	6,220	120/240	240/480 600	...
	5 to 500 incl.	6,600/11,430 Y	6,270	5,940	...	...	...	2300
	2.5, 5, 10 to 500 incl.	11,500	10,925	10,350	...	115/230	230/460 575	...
		12,000	11,400	10,800	...	120/240	240/480 600	...
	5, 10 to 500 incl.	11,000	10,450	9,900	...	...	...	2300/4000 Y
13,200*	1.5, 3 to 500 incl.	7,620/13,200 Y	7,240	6,860	...	120/240	240/480 600	...
		7,940/13,750 Y	7,545	7,145	...	125/250	...	...
	5 to 500 incl.	7,620/13,200 Y	7,240	6,860	...	...	...	2400
	2.5, 5, 10 to 500 incl.	13,200	12,540	11,880	...	120/240	240/480 600	...
		13,750	13,060	12,375	...	125/250	...	...
	5, 10 to 500 incl.	13,200	12,540	11,880	...	...	...	2400/4160 Y
22,000	10 to 500 incl.	22,000	20,900	19,800	...	120/240	240/480 600	2400/4160 Y
33,000*	15 to 500 incl.	33,000	31,350	29,700	...	120/240	240/480 600	2400/4160 Y
44,000	25 to 500 incl.	44,000	41,800	39,600	...	120/240	240/480 600	2400/4160 Y
66,000*	50 to 500 incl.	66,000	62,700	59,400	...	120/240	240/480 600	2400/4160 Y

Table 7.—Standard Three-phase Distribution Transformers, Oil-immersed, Self-cooled

See notes below

Nominal System Voltage Note 1	Standard kva. Sizes for Each Voltage Class Note 5	Rated High Voltages Note 2	High-voltage Taps, Full Capacity Note 3		Rated Voltages, Low Voltage Note 6		
2,300 * and 4,000 *	10 to 150 incl.	2,400/4,160 Y	.....	.....	240/480	600	....
	200 to 450 incl.	2,400/4,160 Y	2,280	2,160	240/480	600	....
4,600	10 to 450 incl.	4,800 Y	4,560	4,320	240/480	600	....
6,600	10 to 450 incl.	6,900 Y	6,555	6,210	230/460	575	....
		7,200 Y	6,840	6,480	240/480	600	....
	10 to 450 incl.	6,600 Y	6,270	5,940	....	...	2300
11,000	10 to 450 incl.	11,500 Y	10,925	10,350	230/460	575	....
	10 to 450 incl.	11,000 Y	10,450	9,900	....	...	2300
13,200 *	15 to 450 incl.	13,200 Y	12,540	11,880	240/480	600	2400
22,000	25 to 450 incl.	22,000 Y	20,900	19,800	240/480	600	2400
33,000 *	50 to 450 incl.	33,000 Y	31,350	29,700	240/480	600	2400
44,000	50 to 450 incl.	44,000 Y	41,800	39,600	240/480	600	2400
66,000 *	150 to 450 incl.	66,000 Y	62,700	59,400	240/480	600	2400

Notes for Tables 4, 5, 6 and 7

Note 1.—Voltages marked * are preferred for new undertakings, and indicate suggested trend for future practice.

Note 2.—Where two voltages are given, the first figure indicates the line to line voltage on the high voltage side when transformers are connected single-phase or three are delta-connected; the second figure indicates the line to line voltage that may obtain, without exceeding the individual high voltage rating as indicated by the first figure, of three transformers Y-connected.

Note 3.—Full rated secondary voltage will be obtained with high voltages lower than rated high voltages as shown by two columns, when transformer is connected to corresponding taps. These taps provide leeway for variation in system voltage.

Note 4.—Nominal low voltage is voltage at point of use, and is that at which motor performance, etc., is guaranteed. Rated low voltages are those that will be delivered by the transformer and provide for line drop between transformer and current-using apparatus.

Note 5.—Transformers will deliver continuously full rated kva. output at rated voltage and at 5% above rated voltage without exceeding guaranteed temperature rise. They are suitable for operation at 5% below rated voltage, but may not conform to performance standards for operation under rated conditions.

Note 6.—Transformers with low-voltage rating 115/230 and 120/240 volts, 200 kva. and smaller are suitable for series multiple or 3-wire service; sizes 250–500 kva. inclusive are suitable for 3-wire service only. Transformers with low voltage rating 230/460 and 240/480 volts are suitable for series or multiple service only.

Standard transformers have a guaranteed temperature rise of 55° C. for continuous operation, referred to room temperature for self-cooled, and inlet water temperature for water-cooled transformers. The usual temperature limits specified are: room, 40° C.; inlet water, 25° C. Temperature rise is not affected by room or water temperature, but actual temperature is the sum of reference temperature and rise. The A.I.E.E. rules limit the observable temperatures after continuous load to 95° and 80° C. respectively, for self- and water-cooled transformers with insulations commonly used.

Power Transformer Construction.—Single-phase transformers usually are built on 2-legged cores and 3-phase transformers on 3-legged cores. Fig. 13 shows a typical single-phase construction with circular winding. Circular coils are produced in three ways: Assembled disc coils stacked over a Herkolite cylinder and connected; continuous disc coils wound directly on the Herkolite cylinder; helical coils with rectangular strands of the conductor wound in parallel over a Herkolite cylinder. Disc coils and continuous disc coils are used for high and medium voltage transformer ratings, and helical coils for low voltage and high current transformer ratings. High-voltage, 2-winding transformers usually have the low voltage winding over the core leg and the high voltage winding concentrically outside.

Cost of Power Transformers.—Figs. 14 and 15 show approximate 1935 prices per kva., of self-cooled step-up or step-down power transformers.

DISTRIBUTION TRANSFORMERS include ratings of 500 kva. and less, single and 3-phase, 25 and 60 cycles, for primary voltages up to 66,000 volts. The average industrial user is concerned only with primary distribution voltages of 2600 to 6600 volts. The following are standard kva. ratings of distribution transformers in sizes up to 500 kva.:

Single-phase, kva.....1.5, 2.5, 3, 5, 7.5, 10, 15, 25, 37.5, 50, 75, 100, 150, 200, 250, 333, 500
Three-phase, kva.....................10, 15, 25, 37.5, 50, 75, 100, 150, 200, 300, 450

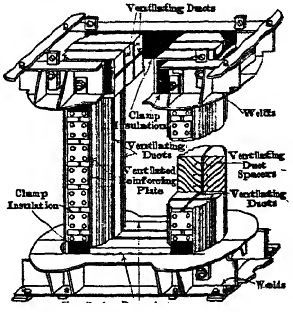
Table 6 covers the range of voltage ratings for single-phase, and Table 7 the range for 3-phase transformers.

Distribution Transformer Construction varies with differences in rating, either as to capacity or voltage. Differing heat-radiating requirements, mechanical forces, and voltage stresses are best met by changes in the core, coil, insulation or tank construction. See Table 8 for construction of single-phase transformers. Three-phase distribution transformers have core type construction.

Tank construction varies with kva. rating, a plain tank being sufficient for a small transformer. As kva. capacity increases, heat increases in proportion to volume of core and coils, but heat dissipating ability increases only as tank surface. Surface is increased by adding corrugations or tubes. Tanks are built of copper-bearing steel with parts welded together.

Self-contained lightning protection is available, in sizes of 100 kva. and smaller, for 2400/4160 and 4800/8320 grounded Y circuits. See Fig. 3 for connections as applied to single-phase units.

Cost of Distribution Transformers.—Figs. 16 and 17 show approximate 1935 prices per kva. of self-cooled 60-cycle distribution transformers. 25-cycle transformers range from 20 to 60% higher, depending on voltage and kva. rating.



Assembly of Two-legged Split Core

FIG. 13. Typical Power Transformer Core Construction

INSTRUMENT TRANSFORMERS are interposed between high-voltage circuits and meters and instruments. They are classed as potential transformers and current transformers.

Potential Transformers, built both for indoor and outdoor service, are used to obtain

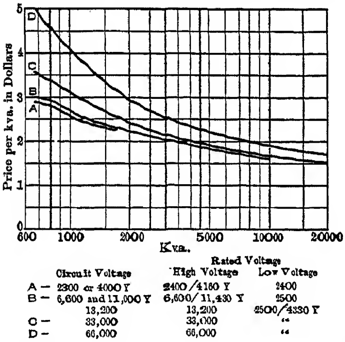


FIG. 14. Self-cooled, 60-cycle, Single-phas Power Transformers

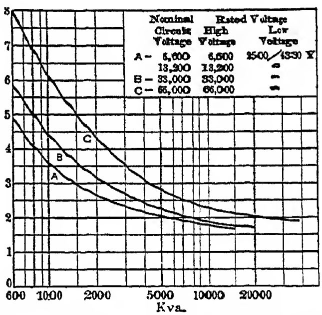


FIG. 15. Self-cooled, 60-cycle, 3-phase Power Transformers

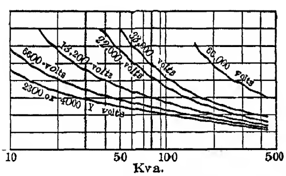


FIG. 16. Self-cooled, 3-phase, 60-cycle Distribution Transformers

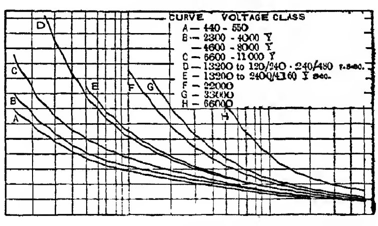


FIG. 17. Self-cooled, Single-phase, 60-cycle Distribution Transformers

Approximate 1935 Transformer Prices

voltage measurements when it is necessary to insulate switchboard appliances from line voltage on A.C. circuits. Indoor construction is built for voltages up to 13,800; outdoor

construction is used for both outdoor and indoor service with higher voltages. The accuracy of the average transformer will be within 1% of the name-plate ratio within the volt-ampere ratings. At low outputs, the phase angle will not exceed 15 min. Special transformers are designed for higher accuracy. See Table 11 for volt-ampere ratings.

Potential transformers are available with fuses mounted on the cases, or separately mounted. Current-limiting resistors are recommended on all circuits above 550 volts where primary fuses are used.

Current Transformers, built for indoor or outdoor service, are used where current measurements are taken and it is necessary to insulate against high voltage, or where circuit current is too high to be read directly. See Table 12.

Table 8.—Index to Construction of Standard 60-cycle, Single-phase Distribution Transformers

High-voltage Class	Kva. Range	Type	High-voltage Class	Kva. Range	Type
4,800/8,220 Y and below	1.5–15	shell	22,000	10–200	shell
	25–150	shell		250–500	core
	200–500	shell	33,000	15–200	shell
6,900/11,950 Y and 7,620/13,200 Y	115–150	shell		250–500	core
	200–500	core	44,000	25–200	shell
11,500 and 13,200	2.5, 5, 10–200	shell		250–500	core
	200–500	core	66,000	50–200	shell
				250–500	core

Table 9.—Electrical and Mechanical Data of Distribution Transformers

Single-phase, self-cooled, 60 cycles

Kva. Continuous 55° C. Temperature Rise	Watts Loss		Percent Efficiency				Percent Regulation Power Factor		Net Weight of Transformer, Including Oil, lb.	Gallons of Oil Required	Approximate Overall Dimensions, in.	
	No Load	Total	Full Load	3/4 Load	1/2 Load	1/4 Load	1.0	0.8			Projected Floor Space	Height
	For Nominal 2300- and 4000-Y-volt Circuits. Low Voltage 120/240 Based on 2400 volts, 60 cycles, sine wave											
1.5	20	66	95.8	96.1	95.9	94.2	3.10	3.05	120	3 1/4	16 3/4 × 16 3/4	20
3	28	96	96.9	97.1	97.0	95.9	2.30	2.70	150	4 1/2	16 3/4 × 16 3/4	21 3/4
5	36	142	97.2	97.5	97.5	96.7	2.15	2.75	220	6	18 3/4 × 18 3/4	23 1/2
7.5	48	195	97.4	97.7	97.7	97.0	2.00	2.70	290	9	18 3/4 × 18 3/4	28 1/4
10	57	245	97.6	97.8	97.9	97.3	1.90	2.70	355	11	18 3/4 × 18 3/4	31 1/2
15	77	338	97.8	98.0	98.1	97.5	1.80	2.75	500	21	22 × 21 1/2	39
25	115	503	98.0	98.2	98.3	97.8	1.60	2.65	675	27	23 1/2 × 24 1/2	39
37.5	148	660	98.2	98.4	98.5	98.1	1.40	2.65	940	32	23 1/2 × 24 1/2	45
50	186	803	98.4	98.6	98.6	98.2	1.30	2.60	1170	40	29 × 27 1/2	45
75	280	1210	98.4	98.6	98.6	98.2	1.30	3.00	1540	54	29 × 27 1/2	54
100	370	1570	98.4	98.6	98.6	98.2	1.30	3.10	1705	50	29 × 27 1/2	54
150	550	2340	98.4	98.6	98.6	98.2	1.25	3.10	2135	69	29 × 31	70
200	800	3010	98.5	98.6	98.6	98.1	1.20	2.90	3150	103	37 × 40	74
250	1115	3945	98.4	98.5	98.5	97.9	1.25	3.75	4060	145	40 × 45	90
333	1310	4835	98.5	98.7	98.7	98.2	1.20	3.70	4730	155	40 × 45	96
500	1675	6545	98.7	98.8	98.8	98.4	1.10	3.65	6670	235	47 × 53	96
	For Nominal 440- and 550-volt Circuits. Low Voltage 120/240 Based on 480 or 600 volts, 60 cycles, sine wave											
1.5	20	66	95.8	96.1	95.9	94.2	3.10	3.05	120	3 1/2	16 3/4 × 16 3/4	20
3	28	96	96.9	97.1	97.0	95.9	2.30	2.70	150	4 1/2	16 3/4 × 16 3/4	21 3/4
5	36	142	97.2	97.5	97.5	96.7	2.15	2.75	220	6	18 3/4 × 18 3/4	23 1/2
7.5	48	195	97.4	97.7	97.7	97.0	2.00	2.70	290	9	18 3/4 × 18 3/4	28 1/4
10	57	245	97.6	97.8	97.9	97.3	1.90	2.70	355	11	18 3/4 × 18 3/4	31 1/2
15	77	338	97.8	98.0	98.1	97.5	1.80	2.75	500	21	22 × 21 1/2	39
25	115	503	98.0	98.2	98.3	97.8	1.60	2.65	685	27	23 1/2 × 24 1/2	39
37.5	148	660	98.2	98.4	98.5	98.1	1.40	2.65	940	32	23 1/2 × 24 1/2	45
50	186	803	98.4	98.6	98.6	98.2	1.30	2.60	1180	40	29 × 27 1/2	45
75	280	1210	98.4	98.6	98.6	98.2	1.30	3.00	1520	54	29 × 27 1/2	54
100	370	1570	98.4	98.6	98.6	98.2	1.30	3.10	1695	50	29 × 27 1/2	54
150	550	2340	98.4	98.6	98.6	98.2	1.25	3.10	2190	70	29 × 31	70
200	800	3010	98.5	98.6	98.6	98.1	1.20	2.90	3140	102	37 × 40	74

Table 9—*Continued*

Kva. Continuous 55° C. Temperature Rise	Watts Loss No Load	Watts Loss Total	Percent Efficiency Full Load	Percent Efficiency 3/4 Load	Percent Efficiency 1/2 Load	Percent Efficiency 1/4 Load	Percent Regulation Power Factor .0	Percent Regulation Power Factor 0.8	Net Weight of Transformer, Including Oil, lb.	Gallons of Oil Required	Projected Floor Space	Height
For Nominal 6600- and 11,000-Y-volt Circuits. Low Voltage 115/230 Based on 6900 volts, 60 cycles, sine wave												
1.5	21	67	95.7	96.0	95.8	94.0	3.15	4.90	180	5	16 3/4 × 16 3/4	25 1/4
3	30	115	96.3	96.6	96.6	95.5	2.95	4.90	260	7 1/2	18 3/4 × 18 3/4	26 1/2
5	44	174	96.6	96.9	97.0	96.0	2.72	5.00	290	8	18 3/4 × 18 3/4	28 1/4
7.5	57	232	97.0	97.3	97.3	96.5	2.46	4.90	330	9 1/2	18 3/4 × 18 3/4	31 1/2
10	71	288	97.2	97.5	97.5	96.7	2.30	4.60	385	8	18 3/4 × 18 3/4	31 1/2
15	92	372	97.5	97.8	97.8	97.1	2.00	4.30	615	16	22 × 21 1/2	39
25	140	550	97.8	98.0	98.1	97.4	1.80	4.70	890	31	23 1/2 × 24 1/2	45
37.5	200	740	98.0	98.2	98.2	97.5	1.60	4.70	1065	30	25 × 24 1/2	45
50	270	940	98.1	98.3	98.2	97.5	1.50	4.30	1295	37	29 × 27 1/2	45
75	385	1285	98.3	98.4	98.4	97.6	1.35	3.90	1835	50	29 × 27 1/2	70
100	460	1690	98.3	98.4	98.4	97.8	1.35	4.10	1970	48	29 × 27 1/2	70
150	725	2445	98.3	98.5	98.4	97.8	1.30	4.00	2500	70	29 × 31	86
200	920	3260	98.3	98.5	98.5	97.9	1.30	4.05	4020	150	40 × 45	92
250	1115	3945	98.4	98.5	98.5	97.9	1.30	4.05	4360	198	43 × 48	92
333	1310	4825	98.5	98.7	98.6	98.1	1.20	4.00	4730	194	43 × 48	98
500	1675	6545	98.7	98.8	98.8	98.4	1.15	4.00	6490	289	47 × 53	110
For Nominal 13,200-volt Circuits. Low Voltage 240/480												
2.5	42	116	95.5	95.7	95.3	93.0	3.05	4.40	255	7 1/2	18 3/4 × 18 3/4	26 1/2
5	57	182	96.4	96.7	96.5	95.0	2.60	4.20	285	8	18 3/4 × 18 3/4	28 1/4
10	90	290	97.1	97.3	97.2	96.0	2.10	4.00	375	8 1/4	18 3/4 × 18 3/4	31 1/2
15	118	403	97.4	97.5	97.5	96.5	2.00	3.70	600	16	22 × 21 1/2	39
25	168	553	97.8	98.0	97.9	97.0	1.65	4.10	885	31	23 1/2 × 24 1/2	45
37.5	225	755	98.0	98.1	98.1	97.3	1.55	4.00	1030	30	25 × 24 1/2	45
50	295	965	98.1	98.2	98.1	97.3	1.50	4.35	1305	37	29 × 27 1/2	45
75	415	1375	98.2	98.3	98.2	97.5	1.40	4.20	1785	50	29 × 27 1/2	70
100	528	1758	98.2	98.4	98.3	97.6	1.35	3.80	1960	48	29 × 27 1/2	70
150	750	2520	98.3	98.4	98.4	97.7	1.30	4.15	2445	70	29 × 31	86
200	950	3240	98.4	98.5	98.5	97.8	1.30	4.15	4070	150	40 × 45	92
250	1115	3815	98.4	98.6	98.5	97.9	1.25	4.00	4280	198	43 × 48	92
333	1310	4665	98.6	98.7	98.7	98.1	1.15	3.90	4640	194	43 × 48	98
500	1675	6325	98.7	98.8	98.8	98.4	1.10	3.85	6440	291	47 × 53	110
For Nominal 33,000-volt Circuits. Low Voltage 2400/4160 Y												
15	200	550	96.4	96.5	96.3	94.4	2.50	5.10	1165	50	23 1/2 × 26 1/2	82
25	258	768	97.0	97.1	97.0	95.5	2.20	4.95	1410	62	29 × 29 1/2	82
37.5	336	1024	97.3	97.4	97.3	96.1	2.00	4.85	1490	59	29 × 29 1/2	82
50	420	1250	97.5	97.6	97.5	96.3	1.80	4.75	2075	96	29 × 36	85
75	550	1680	97.8	97.9	97.8	96.7	1.70	4.65	2460	106	33 × 38	91
100	670	2050	97.9	98.1	98.0	97.0	1.55	4.55	2610	102	33 × 38	91
150	890	2745	98.2	98.3	98.2	97.3	1.40	4.50	3520	145	40 × 45	99
200	1095	3435	98.3	98.4	98.3	97.5	1.35	4.45	3920	143	40 × 45	99
250	1260	4010	98.4	98.5	98.4	97.7	1.25	4.40	4620	231	47 × 53	99
333	1553	4904	98.5	98.6	98.5	97.9	1.20	4.30	5330	251	47 × 53	105
500	2050	6600	98.7	98.7	98.7	98.1	1.10	4.25	7480	380	51 × 57	117
For Nominal 66,000-volt Circuits. Low Voltage 2400/4160 Y												
50	640	1468	97.1	97.1	96.7	94.7	2.00	5.90	3900	219	38 × 46	124
75	820	1895	97.5	97.5	97.1	95.4	1.75	5.80	4070	213	38 × 46	124
100	970	2318	97.7	97.7	97.4	95.9	1.65	5.75	4380	212	39 × 46	124
150	1220	3070	98.0	98.0	97.8	96.5	1.55	5.70	4710	207	39 × 46	124
200	1440	3780	98.1	98.2	98.0	96.9	1.50	5.60	6090	308	47 × 53	136
250	1630	4475	98.2	98.3	98.1	97.1	1.45	5.60	8270	517	56 × 63	137
333	1875	5455	98.3	98.4	98.3	97.5	1.40	5.55	8770	513	56 × 63	137
500	2400	7160	98.5	98.6	98.5	97.8	1.25	5.45	10,280	583	56 × 63	149

Table 10.—Electrical and Mechanical Data of Distribution Transformers

Three-phase, self-cooled, 60 cycles

Kva. Continuous 55° C. Temp. Rise	Watts Loss: No Load	Watts Loss: Total	Percent Efficiency: Full Load	3/4 Load	1/2 Load	1/4 Load	Percent Regulation, Power Factor 1.0	0.8	Net Weight of Transformer, Including Oil, lb.	Gallons of Oil Required	Approximate Overall Dimensions, in. Projected Floor Space	
For Nominal 2300- and 4000-Y-volt Circuits. Low Voltage 240/480												
10	90	345	96.6	96.9	97.0	95.9	2.60	4.15	535	13	31 × 18 1/2	29
15	113	443	97.1	97.4	97.4	96.5	2.30	4.00	770	24	34 1/2 × 20	35
25	162	650	97.4	97.7	97.7	97.0	2.05	3.90	1030	36	39 × 21 1/2	38
37.5	214	890	97.6	97.9	97.9	97.3	1.90	3.80	1335	42	39 × 21 1/2	44
50	275	1117	97.8	98.0	98.0	97.4	1.80	3.75	1580	54	40 × 24	44
75	370	1525	98.0	98.2	98.2	97.7	1.65	3.70	1835	50	44 × 24	47
100	455	1860	98.1	98.3	98.4	97.8	1.50	3.65	2365	60	50 × 24 1/2	55
150	590	2445	98.4	98.5	98.6	98.1	1.35	3.50	3020	63	50 × 24 1/2	61
200	785	3093	98.4	98.6	98.6	98.1	1.25	3.45	3700	101	56 × 29	67
300	1100	4515	98.5	98.6	98.7	....	1.25	3.40	4960	169	60 × 33	81
450	1550	6350	98.6	98.7	98.7	....	1.15	3.30	6940	260	64 × 35	93
For Nominal 6600-volt Circuits. Low Voltage 230/460												
10	100	385	96.2	96.6	96.6	95.5	2.95	4.40	745	25	34 1/2 × 20	35
15	132	512	96.7	97.0	97.0	96.0	2.65	4.40	935	39	39 × 21 1/2	38
25	190	735	97.1	97.4	97.4	96.5	2.30	4.20	1155	47 1/2	39 × 21 1/2	44
37.5	260	1000	97.4	97.6	97.6	96.8	2.10	4.20	1500	57	40 × 24	44
50	315	1205	97.6	97.8	97.9	97.1	1.90	4.10	1640	56	44 × 24	44
75	440	1650	97.8	98.0	98.0	97.3	1.75	4.20	2130	66	50 × 24 1/2	68
100	560	2050	98.0	98.1	98.1	97.4	1.60	3.90	2565	79	50 × 24 1/2	74
150	730	2760	98.1	98.3	98.3	97.7	1.45	3.70	3455	107	56 × 29	80
200	850	3370	98.3	98.5	98.5	98.0	1.35	3.60	3950	121	56 × 29	86
300	1150	4640	98.4	98.6	98.6	....	1.30	4.20	5060	172	60 × 33	81
450	1600	6535	98.5	98.7	98.7	....	1.25	4.00	6910	256	64 × 35	93
For Nominal 13,200-volt Circuits. Low Voltage 2400												
15	200	600	96.1	96.3	96.1	94.3	2.75	4.30	950	40	39 × 21 1/2	38
25	245	845	96.7	96.9	96.9	95.6	2.50	4.20	1145	48	39 × 21 1/2	44
37.5	325	1100	97.1	97.3	97.3	96.1	2.15	4.00	1480	58	40 × 24	44
50	400	1340	97.4	97.5	97.5	96.4	2.00	4.00	1625	57	44 × 24	44
75	520	1840	97.6	97.8	97.7	96.8	1.85	4.00	2235	69	50 × 24 1/2	68
100	665	2225	97.8	97.9	97.9	97.0	1.65	3.75	2590	82	50 × 24 1/2	74
150	810	2940	98.0	98.2	98.2	97.5	1.55	3.65	3395	110	56 × 29	80
200	935	3435	98.3	98.4	98.4	97.8	1.35	3.60	3890	122	56 × 29	86
300	1225	4625	98.4	98.6	98.6	....	1.30	4.10	4870	144	60 × 33	77
450	1650	6450	98.5	98.7	98.7	....	1.20	3.80	6630	228	62 × 34	95
For Nominal 33,200-volt Circuits. Low Voltage 600												
50	450	1605	96.8	97.1	97.1	95.9	2.45	4.90	3630	206	58 × 30	91
75	570	2040	97.3	97.5	97.5	96.6	2.10	4.70	3890	200	58 × 30	91
100	690	2455	97.6	97.8	97.8	96.9	1.90	4.40	4260	221	58 × 30	97
150	925	3285	97.8	98.0	98.0	97.2	1.75	4.40	4530	223	62 × 34	97
200	1155	4050	98.0	98.1	98.1	97.4	1.60	4.40	5180	223	62 × 34	97
300	1550	5570	98.1	98.3	98.3	....	1.50	4.40	6400	268	64 × 35	103
450	2105	7535	98.3	98.5	98.4	....	1.40	4.30	8280	360	67 × 37	115
For Nominal 66,000-volt Circuits. Low Voltage 2400												
150	1450	3850	97.5	97.5	97.3	95.9	1.90	5.80	8,570	522	75 × 38	128
200	1700	4650	97.7	97.8	97.6	96.3	1.75	5.70	9,500	575	79 × 42	134
300	2200	6140	98.0	98.0	97.9	....	1.60	5.60	10,300	562	79 × 42	134
450	2850	8150	98.2	98.3	98.1	....	1.45	5.50	11,600	605	79 × 42	140

TRANSFORMER CONNECTIONS FOR VOLTAGE TRANSFORMATION, 3-PHASE are the delta-delta, Fig. 18; star-star, Fig. 19; star-delta, Fig. 20; delta-star, Fig. 21; open-delta, Fig. 22. The relative advantages and disadvantages are as follows:

Delta-delta.—*Advantages.* Most economical connection for large output at low voltage; a 3-phase bank can operate in open-delta for 58% output if one unit or phase fails; third harmonic

Table 11.—Potential Transformer Ratings—Indoor Type

Volts	25 Cycles On Accuracy Basis	25 Cycles On Thermal Basis	60 Cycles On Accuracy Basis	60 Cycles On Thermal Basis	Volts	25 Cycles On Accuracy Basis	25 Cycles On Thermal Basis	60 Cycle On Accuracy Basis	60 Cycle On Thermal
	Capacity, kva.					Capacity, kva.			
230	0.05	0.30	0.05	0.20	2000–3000	0.10	0.30	0.10	0.30
460	.05	.30	.05	.20	3100–3800	.20	1.10	.20	.38
575	.05	.30	.05	.20	4140–5060	.20	1.10	.20	.38
2300	.05	.30	.05	.20	5175–6325	.20	1.25	.20	.38
2500	.05	.30	.05	.20	6900	.....	......	.20	.38
207–253	.10	.30	.10	.30	6210–7590	.20	0.80	.20	.80
414–506	.10	.30	.10	.30	11,500	.20	.80	.20	.80
518–632	.10	.30	.10	.30	13,800	.20	.80	.20	.80

Table 12.—Common Current Transformer Ratios

Ampere Capacity, Primary	Ratio	Approximate Shipping Weight, lb.	Ampere Capacity, Primary	Ratio	Approximate Shipping Weight, lb.	Ampere Capacity, Primary	Ratio	Approximate Shipping Weight, lb.
5	1 : 1	32	40	8 : 1	32	150	30 : 1	32
10	2 : 1	32	50	10 : 1	32	200	40 : 1	32
15	3 : 1	32	60	12 : 1	32	300	60 : 1	36
20	4 : 1	32	75	15 : 1	32	400	80 : 1	36
25	5 : 1	32	80	16 : 1	32	500	100 : 1	43
30	6 : 1	32	100	20 : 1	32	600	120 : 1	43
						800	160 : 1	43

voltages are eliminated; easy to phase in for parallel operation; heavily unbalanced 3-wire loads can be supplied without serious voltage unbalance. *Disadvantages.* Copper cross-section of both primary and secondary windings is a minimum while number of turns and insulation per phase is a maximum, resulting in a low copper factor and least sturdy winding; neutral point of windings not available; insulation stresses higher than in a star connection.

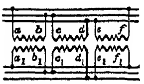


FIG. 18
Delta-delta

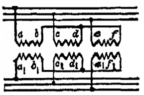


FIG. 19
Star-star

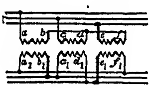


FIG. 20
Star-delta

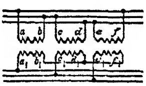


FIG. 21
Delta-star

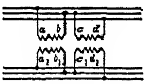


FIG. 22
Open Delta

Transformer Connections

Star-star, used mostly for 3-phase, core-type transformers for small power loads. *Advantages.* Copper cross-section is maximum, number of turns in insulation per phase is a minimum resulting in high copper factor and mechanically strong winding; able to withstand heavy external short circuits; most economical for small output at high voltages; both neutrals available for grounding or for balanced 4-wire supply; easy to phase in for parallel operations; two-phases can be operated for single-phase at 58% output. *Disadvantages.* Neutral points are inherently unstable unless solidly grounded, and unbalanced 4-wire load cannot be supplied unless primary and secondary neutral points are tied together for shell-type or banks of single-phase transformers; a fault on one phase renders a 3-phase bank or unit inoperative for 3-phase output; third harmonic voltages exist but can be eliminated if transformer has a tertiary winding, connected delta, which supplies a short-circuited path for third harmonic currents. This delta winding also can supply loads, and if connected to a synchronous motor or condenser improves power factor.

Star-delta connection is considered best for high voltage transmission. Its chief application is a balanced 3-phase load, such as motors. *Advantages.* Third harmonic voltages eliminated by delta-connected secondary; primary can be grounded; most desirable for step-down transformers; secondary delta connection stabilizes the primary neutral. *Disadvantages.* No secondary neutral available for grounding or for 3-phase, 4-wire supply; a fault on one phase renders a 3-phase unit or bank inoperative.

Delta-star connection is used chiefly for step-down transformers, 4-wire distribution, supplying motor and lighting loads, balanced or unbalanced, and also for stepping-up voltage for power transmission. *Advantages.* No third harmonic; secondary neutral available for grounding or for 3-phase, 4-wire supply; suitable for unbalanced 4-wire load, resulting unbalanced voltage being relatively small; balanced and unbalanced loads may be applied simultaneously. *Disadvantages.* No primary neutral available for grounding; a fault in one phase makes 3-phase unit or bank inoperative.

Open-Delta.—*Advantages.* A 3-phase, shell-type transformer can operate in open-delta with one damaged phase; a transformer bank consisting of three single-phase units can be operated in an open-delta, since a damaged single-phase unit can be removed entirely. *Disadvantages.* A 3-phase, core-type unit usually cannot be operated open-delta because all three phases are magnetically interlinked; with delta-connected 3-phase, shell-type transformers, a damaged phase must be disconnected and short circuited on itself to prevent voltage being induced by the good phases; to

operate a 3-phase, core-type transformer in open-delta, the damaged phase must remain open circuited and yet be capable of withstanding normal voltage induced in it from the other phase windings; when connected open-delta, current in each transformer is 30° out of phase with voltage and transformer operates at 86.6% power factor if load is non-inductive. Capacity of a 3-phase transformer, or of a 3-phase bank, connected open-delta, with the damaged phase cut out is $0.86 \times 0.666 = 0.58$, or 58% of the bank rating.

CONNECTIONS FOR PHASE TRANSFORMATION.—Two- or Three-phase to Single-phase.—It is practically impossible to transform from polyphase to single phase with transformers and obtain balanced conditions. The best method is to simply connect the transformer across one phase of a polyphase system. If resulting unbalance is serious, a polyphase motor, driving a single-phase generator, should be used.

Two-phase to Six-phase.—Fig. 23 shows the double-T connection usually used to operate a 6-phase synchronous converter from a 2-phase supply. The cost of a standard 6-phase converter so connected may be less than 2-phase transformers and special 2-phase converters.

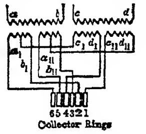


FIG. 23. 2-phase to 6-phase

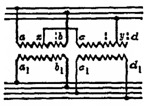


FIG. 24. 3-phase to 2-phase. (Scott Connection)

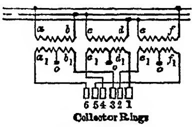


FIG. 25. 3-phase to 6-phase for Synchronous Converters

Transformer Connections

The T-connection requires two special transformers of equal impedance, each with two low-voltage windings so connected as to be displaced 180°, giving the 6-phase relation. Each transformer must have a capacity 15% greater than half the input required by the converter. Starting taps and a neutral tap complicate connections considerably.

Three-phase to Two-phase.—The most common connection is the Scott-connection, Fig. 24, using two transformers, T-connected on the 3-phase side, consisting of main and teaser windings. The teaser has 86.6% of the number of turns in the main winding. On the 2-phase side, both windings are identical and independent for supplying the 2-phase, 4-wire system.

Usually main and teaser transformers are identical. The 3-phase winding of each has a 50% and 86.6% tap. When used as a teaser, one unit will have 13.4% of its winding idle. The Scott-connection thus requires 6.7% more copper than single-phase transformers delivering the same power.

Three-phase to Six-phase transformation is required for standard 6-phase synchronous converters. Four different transformer connections can be used. The most common is the **diametrical,** which requires only one low-voltage winding on each transformer. See Fig. 25. The two leads of each secondary winding are connected to diametrically opposite points on the converter armature. See Fig. 26. With this connection it is possible to operate the 6-phase converter at reduced output with one transformer out of service.

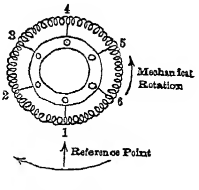


FIG. 26. Phase Rotation in 6-phase Synchronous Converter

PARALLEL OPERATION OF TRANSFORMERS.—Two transformers having the same ratio, and proper impedance, can be connected in parallel if polarity, phase rotation and angular displacement are the same. Delta-delta and star-star transformers have correct angular displacement, when the polarity and the phase rotation are correct. With delta-star or star-delta transformers, correct adjustment can be made by proper sequence of leads. If voltage diagrams are available for the transformers to be paralleled, it is only necessary that these diagrams coincide and that corresponding terminals be connected together. Leads which are to be connected will be at the same potential, which is the basic requirement for parallel operation. When voltage diagrams coincide, polarity and phase rotation must agree. When transformer leads are marked in accordance with the N.E.L.A. and A.I.E.E. standard markings, it is only necessary to connect similarly lettered leads together.

Three-phase transformers can be grouped according to their angular displacements. See Fig. 27. To operate in parallel transformers must belong to the same group. One group cannot be changed to another by interchange of external leads. For instance, two delta-delta transformers, one of group 1 and the other of group 2, cannot be operated in parallel. Table 13 shows operative and inoperative connection.

Effect of Ratio on Parallel Operation.—Circulating currents will flow in the winding of parallel-connected transformers of unequal ratios of high and low voltage windings.

Table 13.—Operative and Inoperative Parallel Connections for Transformers

	Operative Parallel Connections				Inoperative Parallel Connections			
	Low-voltage Side		High-voltage Side		Low-voltage Side		High-voltage Side	
	Bank A	Bank B	Bank A	Bank B	Bank A	Bank B	Bank A	Bank B
1	Delta	Delta	Delta	Delta	Delta	Delta	Delta	Star
2	Star	Star	Star	Star	Delta	Delta	Star	Delta
3	Delta	Star	Delta	Star	Star	Star	Delta	Star
4	Star	Delta	Star	Delta	Star	Star	Star	Delta
5	Delta	Delta	Star	Star	.....	.....	.....	.....
6	Delta	Star	Star	Delta	.....	.....	.....	.....
7	Star	Star	Delta	Delta	.....	.....	.....	.....
8	Star	Delta	Delta	Star	.....	.....	.....	.....

This current is equal to the difference of the two secondary voltages, divided by the sum of the impedances, Z_1 and Z_2, expressed in ohms or percent, of the two transformers. That is, $I_c = (e_1 - e_2)/(Z_1 + Z_2)$

EXAMPLE.—Assume a voltage difference of 2% and an impedance of 4% in each transformer. Percent of circulating current, I_c, then will be

$$I_c = 2 \times 100/(4 + 4) = 25\%.$$

The circulating current therefore is 25% of normal in both windings of the transformers. It adds to the load current in the transformer having the higher voltage and subtracts from the load current in the other.

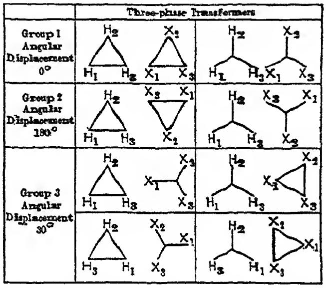


FIG. 27. Three-phase Transformer Connections According to Angular Displacement

The A.I.E.E. standards give the voltage ratio of a transformer as the ratio of root-mean-square primary terminal voltage to the root-mean-square secondary terminal voltage under specified conditions of load.

Effect of Impedance on Parallel Operation.—Impedance of a transformer is the voltage drop at normal load in percent of normal voltage. It is the resultant of the two components, resistance drop in phase with current, and the reactance drop 90° out of phase with current. For successful parallel operation, the ohmic impedance of transformers must be in inverse proportion to the load which they are to carry, so that voltage drop from no load to full load is the same in all units, both in magnitude and phase. Generally, total resistance is small as compared with reactance.

POLARITY OF TRANSFORMERS refers to voltage vector relations of transformer leads as brought outside the tank, both high and low voltage leads being in the same order. See Fig. 28. Polarity is the relative direction of induced voltage from H_1 to . as compared with that from . to X_2, both being in the same order with respect to the tank. Polarity can be additive or subtractive, as indicated, and depends on direction of winding.

A.I.E.E. standards provide that transformer lead designations also shall indicate polarity. When leads are so marked, polarity of transformers is subtractive when H_1 and X_1 are adjacent, and additive when H_1 is diagonally opposite X_1.

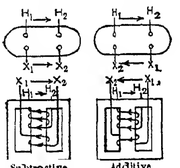


FIG. 28. Transformer Polarity

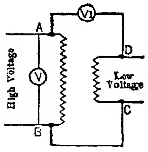


FIG. 29. Connections for Testing Transformer Polarity

Test for Polarity.—See Fig. 29. Connect one high voltage lead to the opposite low voltage lead, B to C. Apply voltage at AB and measure voltage between A and B and A and D. If V_1 is greater than V, polarity is additive; if less it is subtractive.

AUTO-TRANSFORMERS have parts of a winding in common in the primary and secondary circuits. For a given service, the advantages of auto-transformers are lower cost, greater efficiency, better regulation, smaller size, smaller exciting current.

An auto-transformer transforms only a part of the total kva., the rest flowing directly to the load circuit. Percent transformation is the same as percent voltage transformation, based on high voltage. For example, if voltage is raised 10% only 10% of the kva. supplied to the load is transformed. Therefore, physical capacity of the auto-transformer need be only 10% of the total load kva. Capacity/output = $(1 - E_2)/E_1$, where E_1 and

E_2 = respectively, high and low line voltage. See Fig. 30 for comparison of voltage and current relations of an auto-transformer and a transformer.

Disadvantages of Auto-transformers are: *a*. High stresses due to external short circuits because of the low percentage impedance. Thus, if the ratio of equivalent capacity to output is 10%, the short circuit current will be ten times as large and the short circuit stress 100 times as large as for a transformer whose line voltage is constant. *b*. Because of metallic connection between high and low voltage circuits, a disturbance in either circuit affects the other; thus a ground on the high voltage side may subject the low voltage circuit and its connected apparatus to high-voltage line voltage.

AIR-COOLED AUTO-TRANSFORMERS can be used profitably with voltages of 600 or less for both power and lighting as follows:

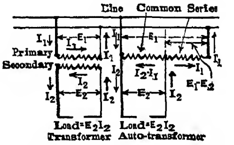


Fig. 30. Voltage and Current Relations in Transformer and Auto-transformers

Lighting.—1. To insulate lighting circuits from power circuits. 2. To boost low line voltage. 3. To operate low-voltage portable lamps in damp locations, where 32 volts may be advisable. 4. Where permitted, to step-down a power circuit voltage for light, thus eliminating separate lighting circuits. 5. To operate low-voltage lamps where concentrated illumination is desired, and where lights are subject to vibration.

Power.—1. To step-down power distribution voltages from 460 to 575 volts, to supply lighting or other 115 volt circuits, with resultant lower line losses, reduced copper cost and improved voltage regulation. 2. To operate low-voltage portable tools from power circuits instead of lighting circuits, giving the advantage of low power rates. 3. To operate 32-volt portable tools in packing plants, mines, tank cars, etc. 4. To balance the voltage on a single-phase, 3-wire system, and prevent under-voltages which would result in unsatisfactory operation of equipment. 5. Boosting or bucking the voltage of single- or 3-phase circuits, in order to operate equipment requiring more or less than the distribution voltage. 6. When phase change is made on a power system, to transform from 3- to 2-phase, in order to permit use of equipment that otherwise would be discarded.

Ratings.—Standard air-cooled transformers are available for frequencies of 25, 50, and 60 cycles in sizes of from 0.05 to 50 kva., single-phase; of from 1 to 250 kva., 3-phase, 60-cycle, for boosting voltage; and from 1 to 50 kva., 60-cycle, for changing 3-phase to 2-phase, or 2-phase to 3-phase.

Typical connections are shown in Figs. 31, 32, and 33.

Price.—Approximate 1935 prices per kva. of single-phase, 60-cycle units are shown in Fig. 34. 50-cycle transformers will cost from 5% to 10% more, and 25-cycle units approximately 40% more than 60-cycle units.

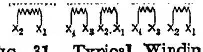


Fig. 31. Typical Winding Diagram for Air-cooled Transformer

Fig. 32. Winding Diagram of Three Single-phase, Air-cooled Auto-transformer Units Connected as a 3-phase Bank

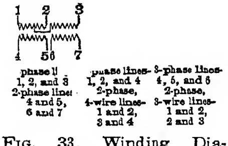


Fig. 33. Winding Diagrams for Phase-changing Transformers and Auto-transformers

Transformer Connections

Limitations.—Air-cooled auto-transformers should not be used where isolation must be maintained between phases in phase transformation. They can be used only on those two-phase systems in which phases are otherwise isolated, or in which the connection between phases, of any of its connected apparatus, corresponds precisely to that of the auto-transformer.

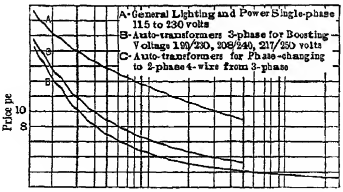


Fig. 34. Approximate 1935 Prices of Standard Air-cooled 60-cycle Transformers and Auto-transformers

TRANSFORMER COOLING INSULATING LIQUID.—Mineral oils have reached a high standard of quality, but are inflammable. The codes require oil-cooled transformers to be placed either out of doors or in fireproof vaults. This limits installations as regards location.

Pyranol now available as a cooling and insulating fluid for transformers is non-inflammable. The 1935 revision of the National Electrical Code permits installation of pyranol-filled transformers indoors, with many restrictions applying to oil-insulated transformer installations eliminated. The advantages of pyranol transformers are: 1. Can be installed indoors without expense of fireproof vaults. 2. All gases derived from the insulating fluid are non-inflammable and non-explosive. 3. Can be installed directly at load centers, providing improved voltage regulation and efficiency.

GENERATED POWER

1. CAPACITY REQUIRED

In comparing cost of generated power with that of purchased power the analysis must consider: 1. Fixed charges on initial investment, including cost of space, building and equipment; these charges cover interest, taxes, insurance, depreciation and obsolescence. 2. Cost of operation including labor, maintenance, fuel and supplies. 3. Present trend toward lower rates for purchased power. 4. Future power requirements.

REQUIRED GENERATING PLANT CAPACITY depends on the amount and character of load and the proportions of total connected load that will come on the generators at any one time. It is determined by multiplying the connected load in each department by its demand factor and dividing the result by the diversity factor.

Demand Factor is the ratio (actual maximum demand ÷ connected load). See p. 15–04.

Demand of an Installation or System is (A.I.E.E. standards) the load which is drawn from the source of supply at the receiving terminals, averaged over a suitable and specified interval of time. It is expressed in kw., kva., amperes or other suitable units. Maximum demand is the greatest of all demands which have occurred during a given period, as determined by measurement with a maximum demand meter. The most common intervals of time used are 15 and 30 min.

Diversity Factor takes account of the number of maximum demands of departmental loads that are likely to occur at the same time. Diversity factor is the ratio (sum of maximum demand of departmental loads ÷ maximum demand of total load). Table 1, from E. W. Lloyd, shows demand and load factors of typical industries in the Chicago district.

Table 1.—Demand and Load Factors of Chicago Consumers, Combined Power and Light.

Kind of Business	Load Factor, percent of 8760-hr. Year	Demand Factor, Ratio of Actual Maximum to Connected Load, percent	Kind of Business	Load Factor, percent of 8760-hr. Year	Demand Factor, Ratio of Actual Maximum to Connected Load, percent
Biscuit manufacturers	35	55	Machine shops	26	55
Boiler shops	18	45	Newspapers	20	75
Boots and shoes	25	65	Packing houses	30	75
Brass and iron beds	20	60	Paint, lead and ink manu-		
Brass manufacturing	28	50	facturers	23	45
Breweries	45	60	Paper-box manufacturers	25	50
Butter and creamers	20	60	Plumbing and pipe fitting	26	55
Can manufacturers	30	70	Pneumatic tube	50	90
Candy manufacturers	18	45	Post offices	50	30
Clothing manufacturers	15	55	Power buildings	27	40
Clubs (large)	40	85	Railroad depots	50	50
Department stores (large)	30	55	Refrigeration	50	90
Electrical manufacturing	25	55	Restaurants (large)	50	60
Electroplating	25	75	Restaurants (small)	30	70
Engraving and printing	19	60	Saw manufacturers	30	55
Express companies	40	60	Screw manufacturers	30	75
Fertilizer manufacturing	75	40	Seed cleaners	25	55
Forge shops	30	49	Sheet-metal manufacturers	18	70
Foundries	15	75	Soap manufacturers	25	60
Furniture manufacturing	28	65	Spice mills	20	55
Glove manufacturing	25	55	Stone cutters	17	55
Grain elevators	10	75	Structural steel	22	40
Grocers (wholesale)	20	55	Textile mills	20	65
Hotels (large)	50	40	Theaters	16	60
Hotels (small)	35	50	Twine mills	30	60
Ice cream manufacturing	45	75	Wood-working	28	65
Jewelry manufacturing	18	50	Woolen mills	27	80
Laundries	25	70			

Example.—Determine generator capacity for two load groups fed by separate radial feeders from the generator bus. Connected load on feeder No. 1, 1000 kva., demand factor, 0.7. Connected load on feeder No. 2, 500 kva., demand factor, 0.5. Maximum demand on feeder No. 1, $0.7 \times 1000 = 700$ kva.; on feeder No. 2, $0.5 \times 500 = 250$ kva. Assume the diversity factor to be 1.4. Maximum demand of total load then is $(250 + 700)/1.4 = 680$ kva., which is the required generator capacity.

Reserve Capacity equal to the largest generating unit in the power plant should be provided in order to carry maximum loads with one unit shut down. The degree of reliability required largely determines reserve capacity. With interconnected plants, reserve capacity can be a much smaller percentage of total generating capacity than with independent plants. From 10 to 25% often is sufficient when several sources of power are available.

Size of Generating Units.—The character of the load as shown by the load factor, and average daily load curves which would apply to the plant under consideration, determine the most economical sizes of individual generating units.

Load Factor is the ratio (average load ÷ peak load) for a specified period of time. Peak load on the generator will be determined over a period of 30 min. or 1 hr. Average load would be taken as for one day, month, or year. Power plant operating costs per unit of energy used decrease with increase of load factor. Operating expense will vary with the size of generating unit. The efficiency of large units operating below rated load is low. Operating charges such as fuel and labor do not decrease in proportion to the load.

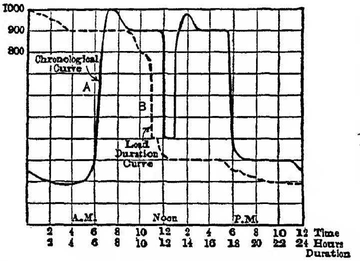


FIG. 1. Typical Industrial Plant Load Curves for Average Day of Year

LOAD CURVES are plotted with kw. load for any time interval as ordinates and the time intervals (usually 1 hr.) as abscissas. The curve is made daily. Such a curve is useful for plant analysis and for predicting magnitude and character of load on the power plant. It will include characteristics of all departmental loads, which will be well established in existing manufacturing plants. In determining power requirements for a new enterprise, predictions can be made on the basis of machinery to be installed. A study of existing plants using the same type of machinery is a good guide in applying correct diversity and demand factors to individual machines and load groups.

Fig. 1 shows a typical daily load curve of an industrial plant load. An approximate daily load curve can be obtained by averaging daily curves for two typical months. The area under the curve is the total output required in kw.-hr. for a typical day.

In Fig. 1, *A* is a chronological curve showing amount of load and time at which it occurs. *B* is the *load-duration curve* derived from *A*. In *B*, the magnitude of the load is plotted against the number of hours it exists. For any load-ordinate of the duration curve, the abscissa shows the total number of hours during the day that the load has been on the generator. To prepare the load-duration curve from the daily load curve, tabulate the load for each hour according to magnitude. Then plot these loads as ordinates, beginning with the maximum at the left and the time accumulating along the abscissas. When plant load curve is established, size and number of generating units can be fixed.

LARGE vs. SMALL GENERATING UNITS.—Generally, for small power plants, with low load factor, the most economical installation is several small generating units. For large plants, with high load factor, fewer units of larger size probably will be best.

In Fig. 1, the load-duration curve shows that for 24 hr. per day the load is 190 kw. or more; for 22 hr., 200 kw. or more; for 18 1/2 hr., 250 kw. or more; for 17 hr., 300 kw. or more; for 11 hr., 400 kw. or more; for 10 hr., 800 kw. or more; and for 9 hr. the load is between 900 and 1000 kw. Five 250-kw. units would meet the needs, and provide a spare 250-kw. unit for the initial load. Four 300-kw. units would fit the curve well, with one unit in use at night at not less than approximately 65% load. This would provide 200 kw. spare capacity, and if one unit is shut down, the other three units can carry the peak load. Future requirements, however, may dictate three 500-kw. units.

Future Growth must be considered in power plant design, in order to provide initially sufficient space for additional generating equipment. Size of individual units and reserve capacity must be determined on the basis of probable future loads. Thus, if the plant load of Fig. 1 should double in a few years, nine 250-kw. units would be poor economy, and five 500-kw. units would be better.

2. ALTERNATING-CURRENT GENERATORS

Electric generators generally are classified as alternating current (A.C.) and direct current (D.C.).

ALTERNATING CURRENT GENERATORS generate a voltage, which in an external circuit causes current to flow alternately in opposite directions. This current builds up to maximum, returns to zero, and builds up to maximum in the reverse direction according

to $i = I_m \sin \theta$, where i = current at any instant; I_m = maximum current; θ = angular position of field pole with respect to armature conductor, expressed in electrical degrees.

Frequency is the number of current reversals per second. If speed of generator is given in r.p.m., and the frequency f in cycles per second, $f = pn/2$, where p = number of poles, and n = r.p.m./60.

Construction.—Generators larger than 25 kva. have revolving fields with salient poles on the rotor for slow speed and medium speed design. High-speed turbine-driven generators have field coils wound in slots on the rotor. Armature windings are stationary. D.C. excitation is applied to the field coils through collector rings on the rotor shaft. Below 25 kva., both revolving armatures and revolving fields are used.

Excitation of alternating current generators is at either 125 or 250 volts D.C., usually obtained from direct-connected exciters.

Low-speed Direct-connected Synchronous A.C. Generators for direct connection to steam and internal combustion engines range in speed up to 450 r.p.m., 60 cycles, and in capacity from 25 to 8750 kva. The standard generator in this classification is supplied with rheostat, foundation caps and brush holder support, but without base, shaft, bearings, foundation bolts, or shaft keys.

Power Factor.—Standard generators are designed for an 80% power factor (lagging) load, but can be built for unity or any other power factor required.

Standard Ratings of low-speed engine type generators, in kva. available at generator terminals at 0.8 power factor, are: 31, 44, 63, 94, 125, 156, 187, 219, 250, 312, 375, 438, 500, 625, 750, 875, 1000, 1125, 1250, 1563, 1875, 2188, 2500, 2812, 3125, 3750, 4380, 5000, 5625, 6250, 7500, 8750, 10,000. The ratings are based on the load that the generators are capable of carrying continuously without exceeding temperature guarantee. The ratings are on a continuous basis for Diesel or gas engine drive and with a 2-hr. overload capacity of 25% for steam engine drive.

Voltage Ratings are standardized at 240, 480, 600 and 2400 volts.

Frequencies of 25, 50 and 60 cycles are standard, with the trend away from 25 and 50 cycles.

Speed Ratings in r.p.m. are:

60 cycle: 450, 400, 360, 327, 300, 277, 257, 240, 225, 200, 180, 164, 150, 138, 128, 120, 109, 100, 90, 80.
50 cycle: 429, 375, 333, 300, 273, 250, 231, 214, 200, 188, 167, 150, 136, 125, 115, 107, 100, 91, 83.
25 cycle: 375, 300, 250, 214, 188, 168, 150, 136, 125, 115, 107, 100, 94, 83.

Rated Load Current in amperes is: Single-phase = kva. × 1000/E; two-phase = kva. × 1000/2E; three-phase = kva. × 1000/$\sqrt{3}E$, where E = rated voltage.

Overload.—Standard generators can carry 150% of rated-load current for one minute, the rheostat being set for rated-load excitation.

Overspeed.—Operation at 25% overspeed is possible without causing mechanical injury. Greater overspeed requires special design.

Temperature Rise for normal maximum-rated generators, *i.e.*, as applied for internal combustion engine drive, operated at rated load until temperature is constant is: Armature core and coils, by thermometer, 50° C.; armature coils, by temperature detector, 60° C.; field coils, by resistance, 60° C.

For 25%, 2-hr. overload-rated generators, the temperature rise allowed is:

	Rated Load Until Constant	125% Load, 2 hr.
Armature core and coils, by thermometer..	40° C.	55° C.
Armature coils, by temperature detector...	50° C.	65° C.
Field coils, by resistance................	50° C.	65° C.

Regulation.—Voltage regulation is the percent rise in voltage from rated load to no load on the generator, with the excitation at rated-load value. Regulation for standard generators is:

Power factor..............................	0.8	0.9	1.0
Full-load regulation, 50° C. generator..........	40%	35%	25%
Full-load regulation, 40° C. generator..........	34%	30%	20%

Direction of Rotation (standard) is clockwise when viewed from the exciter end or end opposite the driven end.

Generator Field Rheostats are recommended for all A.C. generators, but may be omitted when: *a*, the generator is excited from its own individual exciter, which is used for no other purpose; *b*, exciter never will be paralleled with other exciters.

PARALLEL OPERATION OF ENGINE-TYPE GENERATORS is the responsibility of the engine builder, as many factors affecting parallel operation are contained in the engine and in the characteristics of the equipment connected to the system with which the generator must parallel. (See N.E.M.A. Motor and Generator Standards, Publication No. 34–22). Two generators driven by reciprocating engines, and connected in parallel, supplying power to a steady load, may transfer power back and forth due to pulsations in engine torque. The magnitude of the interchange depends on engine and generator characteristics and the total WR^2 of engine flywheels and generator rotors. It may be sufficient to increase mechanical stresses, produce excessive heating in the generators, or even cause them to pull out of synchronism. A power pulsation of 66%

is the maximum that safely may be allowed if injurious heating or other undesirable conditions are to be avoided.

In order to determine paralleling conditions, the following information on each engine and generator is necessary: *For the engine.*—1. Horsepower rating. 2. Number of cylinders. 3. Number of stroke cycles. 4. Single or double acting. 5. Amount of continuous unbalance of any one cylinder. 6. Maximum periodic displacement of the rotor in either direction from the position of uniform rotation. 7. WR^2 of flywheel, lb.-ft.^{2} *For the generator.*—1. Kva. rating. 2. Power factor. 3. Frequency. 4. Speed, r.p.m. 5. WR^2 of rotor, lb.-ft.^{2} 6. P_R, synchronizing power, or rate of change of power with respect to displacement angle, expressed in kw. per radian. With this information available, the commonly accepted method for preliminary determination for successful parallel operation is as follows:

Total flywheel effect of each of the units operating in parallel shall be such that: *a*, maximum periodic displacement of rotor in either direction from its position of uniform rotation will not exceed $3\,1/_2$ electrical deg.; *b*, natural frequency of oscillation will differ at least 20% from frequency of any forced impulse of any of the engines operating in parallel; *c*, engine governors shall not create or sustain an oscillation; *d*, the curves (percent speed *vs.* percent load) of the several engines in parallel shall be alike; *e*, engines shall be so adjusted that each cylinder will contribute its share of turning effort.

Critical Frequencies to be avoided are: *a*, *For a 4-cycle engine*, particularly one half of the revolutions of the engine crank, but also the revolutions of the crank; *b*, *For a 2-cycle engine*, particularly the revolutions of the engine crank, but also twice the revolutions of the crank.

To avoid dangerous values of flywheel effect, in any given unit, the natural frequency at which the rotor tends to oscillate can be changed by changing the flywheel effect as indicated in the equation $F = 35{,}200/n\sqrt{(P_R \times f)/WR^2}$, where F = natural frequency in periods or beats per min.; n = rev. per min.; P_R = kw. input to generator at synchronous speed, corresponding to torque exerted on rotor per radian displacement; f = frequency of generator, cycles per sec.; WR^2 = total flywheel effect, lb.-ft.^{2}. WR^2 and P_R are given in manufacturers' specifications for engine type generators.

Example.—For a 60-cycle, 150-r.p.m. direct-connected generator with P_R = 1420, and driven by a 4-cycle engine, the natural frequency must not be between 60 and 90 nor 120 and 180 periods per min. That is, total WR^2 (flywheel and generator) must not lie between 1,305,000 and 480,000 lb.-ft^2, nor between 526,000 and 145,000 lb.-ft^2.

The combined natural frequency of two units operating in parallel can be determined from

where F_{1+2} = natural frequency of two units in parallel; $(P_R)_1$, $(P_R)_2$ = synchronizing power of units operating in parallel as defined above; P_1, P_2 = number of poles on each unit; $WR_1{}^2$, $WR_2{}^2$ = total flywheel effect of each unit. The above rules are approximate, but will cover most applications.

For discussion of parallel operation of generators driven by reciprocating engines, see R. E. Doherty and R. F. Franklin, Design of Flywheels for Reciprocating Machinery, *Trans.* A.S.M.E., xlii, p. 523, 1920, and H. V. Putnam, Oscillation and Resonance in Systems of Parallel Connected Synchronous Machines, *Jour.* Franklin Inst., May, June 1924.

TORSIONAL VIBRATION.—N.E.M.A. Standards Publication No. 34–22, Item MG 15–84, states that since factors which affect torsional vibration are contained principally in the design of the engine . . ., the responsibility for avoiding torsional vibration shall rest with the engine builder. Since design of the generator rotor is an important factor which must be considered, generator builder will give complete information in so far as generator design affects torsional vibration. He will furnish the engine builder with flywheel effect, weight of generator rotor, and any other information such as stiffness of spider, as engine builder may need to successfully design the combined unit. Shaft diameter will be specified by the engine builder. . . . Before generator spider and such part of the shaft as may be supplied by generator builder are manufactured, final drawings of the same are to be submitted to the engine builder for approval in so far as their design affects torsional vibration.

LIGHT FLICKER in lamps connected to the circuit may be produced by internal combustion engine-generator units. Flicker is due to pulsations in driving torque, which originate in the engine and produce periodic variation in speed, with resultant variation in generator terminal voltage. The frequency of the variation will be the same as the frequency of torque pulsations in the engine. Whether or not flicker is objectionable depends on the frequency and magnitude of voltage fluctuation.

Fig. 2 shows relation between flicker and voltage variation. Accurate measurement of the degree of light flicker is difficult. Voltage variations producing flicker are too small to be indicated on the ordinary voltmeter, and the only device that appears to be satisfactory for measurement of power pulsations is the oscillograph. Frequency and

magnitude of power pulsations should be considered in engine design. The power pulsation produced by any engine-generator unit can be reduced by using the proper amount of inertia or flywheel effect.

NORMAL EFFICIENCIES of standard low-speed synchronous generators, engine-driven, have been standardized by N.E.M.A. See Publication No. 34-24. Efficiencies are determined by including I^2R losses of stator and rotor coils at 75° C., core losses and stray load losses, but excluding losses of field rheostat, exciter, windage and bearing friction. Efficiencies are for 80% power factor generators; approximate full load values range from 82% for a 31 kva. generator to 97% for a 10,000 kva. generator. To include exciter losses, deductions are made from standard values, ranging from 0.2% for generator efficiencies of 96.2 and above, to 1.6% for generator efficiencies of 82.2. Similar deductions can be made for rheostat losses, but only if generator is excited from a D.C. bus where voltage cannot be varied. A deduction from standard full-load efficiency of 0.5% is made for windage and bearing friction losses for machines with one, two or three bearings. Fig. 3 shows typical approximate efficiencies of engine type generators without base, shaft or bearings.

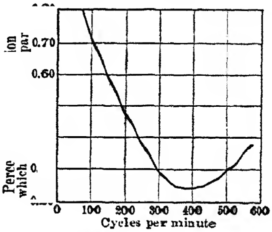


Fig. 2. Voltage Flicker at Various Frequenci... Average for 115-volt, 60-watt M...

Approximate Prices of standard 60-cycle engine-type A.C generators including direct-connected exciters are given in Fig. 4. These prices are based on 1935 figures.

Weights and Dimensions of engine-type A.C. generators are given in Table 2 for representative standard ratings, also WR^2 and required D.C. excitation.

Construction.—Frames are fabricated. Sides are cut from heavy steel plate, and a wrapper plate is formed around outside periphery and welded to the end plates. Stacking studs hold armature or stator punchings between the side plates. Both cast-iron and fabricated rotor spiders are used.

Amortisseur windings consist of hard drawn copper bars built into the field pole faces and silver-soldered into copper end segments at each end of every field pole, to form a short circuited path for induced currents. These windings are recommended for generators direct connected to internal combustion engines to help dampen oscillations set up by the pulsating torque of the engine.

HIGH-SPEED SYNCHRONOUS A.C. GENERATORS are generators of 500 r.p.m. and over, and built either for belt drive or direct connection, with or without direct-connected exciters.

Standard Ratings in kva. available at generator terminals at 0.8 power factor are: 1.25, 2.5, 3.75, 6.3, 9.4, 12.5, 18.7, 25, 31, 44, 63, 94, 125, 156, 187, 219, 250, 312, 375, 438, 500, 625, 750, 875, 1000, 1125, 1250, 1563, 1875, 2188, 2500, 2812, 3125, 3750, 4380, 5000.

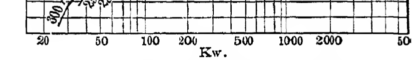


Fig. 3. Approximate Normal Full-load Efficiencies of Low-speed, Engine-driven Synchronous Generators

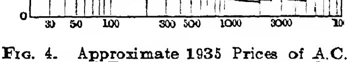


Fig. 4. Approximate 1935 Prices of A.C. Low-speed, Engine-type Synchronous Generators, Including Direct-connected Exciters

Speed Ratings are: 60-cycle, 1800, 1200, 900, 720, 600, 514; 50-cycle, 1500, 1000, 750, 600, 500; 25-cycle, 750, 500.

Standard Voltages are: Up to and including 25 kva., 1800 r.p.m., 120 and 240 volts only; up to 25 kva., 1200 r.p.m. and less, 240, 480 and 600 volts; 25 kva. and above, 240, 480, 600 and 2400 volts.

Temperature Rise, Overload, Overspeed, Direction of Rotation, and Regulation for standard high-speed generators are the same as for low-speed, engine type generators. See p. 15-23.

Construction.—Revolving armatures are usual for 1800 r.p.m. generators up to and including 25 kva., 3-phase, 60-cycle, with power taken from slip rings. Field poles are stationary. All other ratings are of the conventional revolving field, salient pole type.

Bearings of standard generators are not designed to support any weight except that of the rotor. Endshield type bearings are standard for generators up to approximately 1000 kva. at speeds of 514 to 1800 r.p.m. inclusive. Pedestal type bearings are usual on generators larger than 1000 kva. at speeds of 514 to 1800 r.p.m.

EFFECT OF ALTITUDE ON GENERATOR RATINGS.—Rating and temperature rise are based on cooling air of not over 40° C., and altitude not over 1000 meters or 3300 ft. When altitude exceeds 3300 ft. an allowance is made of 1% temperature rise for each 330 ft.

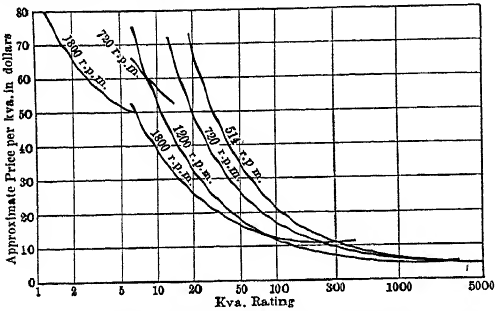


Fig. 5. Approximate 1935 Prices of A.C. High-speed Generators, including Direct-connected Exciters

Example.—Required the operating temperature at 3300 ft. of a generator to operate at 40° C. rise at 7425 ft. *Solution.* $7425 - 3300 = 4125$; $4125/330 = 12.5$; $12.5 \times 0.01 = 0.125$; $1.00 - 0.125 = 0.875$. Temperature rise at 3300 ft. $= 40 \times 0.875 = 35°$ C.

Prices of High-speed A.C. Generators.—Fig. 5 shows approximate 1935 prices of standard generators.

Approximate Weights and Dimensions of standard high-speed A.C. generators are given in Table 3.

Efficiencies.—Fig. 6 shows full load efficiencies of representative high-speed A.C. generators, including I^2R losses of stator and rotor coils at 75° C., but excluding losses of rheostat, exciter, core, stray load, windage and bearing friction. Efficiencies are determined and all tests made in accordance with 1927 A.I.E.E. standards.

Table 2.—Weights and Dimensions of Standard Engine-type A.C. Generators

Kva.	Rev. per Min.	WR^2	Synchronizing Power, kw. P_R	Maximum Excitation, kw.	Net Weight, lb.*	Dimensions, in. High	Dimensions, in. Wide	Dimensions, in. Long
31	450	440	....	2.1	1,660	37	40	28
	300	846	89	2.3	2,120	44	46	29
63	450	600	....	2.8	2,130	37	40	31
	300	1,750	191	3.3	2,860	57	62	32
125	450	1,880	391	3.6	2,920	57	62	32
	300	2,600	....	4.8	3,720	57	62	36
	200	6,900	403	9.2	4,940	67	75	36
187	450	2,750	552	4.5	3,850	57	62	36
	300	6,770	563	5.1	5,060	67	75	36
	200	14,660	548	6.3	6,640	74	89	36
250	450	4,500	....	5.4	4,430	63	68	36
	300	7,150	783	6.5	5,230	67	75	36
	200	15,380	760	9.2	6,850	74	89	36
500	450	7,000	....	7.8	7,970	64	69	50
	300	20,000	1500	9.7	10,100	75	91	44
	200	33,400	1300	11.7	11,050	79	99	43
	120	136,200	1485	16.5	19,380	107	145	45
750	450	11,600	....	9.9	9,820	68	77	54
	300	23,400	2270	13.8	11,850	75	91	50
	200	59,000	2190	13.2	13,690	89	113	43
	120	144,700	2200	23.0	19,890	107	145	45
1000	450	14,500	....	11.7	12,400	68	77	61
	300	39,000	2870	12.0	14,500	80	101	50
	200	98,000	2755	17.9	17,220	97	133	48
	120	196,400	3500	24.0	25,260	107	145	50
2500	300	120,000	....	24.2	28,800	93	123	68
	200	225,000	....	28.2	29,750	108	147	59
	120	640,000	....	34.6	47,000	132	205	65
5000	120	1,650,000	....	....	89,200	156	262	76

* Exclusive of weight of direct-connected exciter.

Belt Drive.—The limits of bearing speed and shaft deflection of two-bearing generators for belt drive are shown in Fig. 7.

3. TURBINE-DRIVEN GENERATORS

Standard turbine-driven A.C. generators usually are considered in connection with a steam turbine as a combined generating unit. They are so handled as regards price, weight, dimensions and efficiency.

TURBO-GENERATOR DESIGN AND CONSTRUCTION.—Standard generators are designed with enclosed revolving fields, 2-pole, 3-phase, 60-cycles between 500 and 7500 kw. They operate at 3600 r.p.m., direct-connected to the steam turbine, with direct-connected exciters of 125 or 250 volts. The frame of the generator is supported by feet at the sides. Within the frame, sheet steel laminations are built in sections and separated by spacers to form ventilating ducts. Laminations are slotted to receive armature coils. The generator is ventilated by fans at both ends of the rotor. Rotors are round, with field coils of copper ribbon wound edgewise and placed in slots in the periphery, and fastened by metal wedges. Collector rings are of steel. 25-cycle generators are driven through reduction gears.

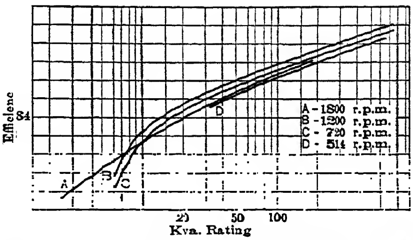


Fig. 6. Approximate Efficiencies of Standard High-speed A.C. Generators, 3-phase, 60-cycle

Standard Ratings in kw. at 0.8 power factor, and the corresponding weights are:

Rating, kw.:	500	625	750	1000	1250	1500	2000
Weight lb.:	7000	8020	9320	10,750	12,500	14,550	17,400
Rating, kw.:	2500	3000	3500	4000	5000	6000	7500
Weight, lb.:	22,100	24,900	28,600	31,550	37,700	45,570	53,900

Usual practice includes dimensions of direct-connected turbine-driven generators in the complete dimensions of the combined unit, because base and foundation construction is laid out for the combined unit.

Temperature Rise of generators for ambient temperatures of 40° C. and altitude of 3300 ft. or less is: armature, 60° C., measured by embedded detectors; field, 85° C., measured by resistance. Temperature rise of exciter is: core and windings, 40° C., measured by thermometer; commutator 55° C., measured by thermometer.

REVOLVING-ARMATURE TYPE A.C. TURBINE GENERATORS.—Units of from 12.5 to 75 kva. are available for operation at 3600 r.p.m., 60-cycle, 3-phase. The armature is wound on the rotor, and power is taken off at collecter rings on the shaft. Field poles are stationary. Voltages are limited to 240 and 480 volts.

Standard Ratings at 0.8 power factor are: 10, 15, 20, 25, 35, 50, and 60 kw. Direct-current units also are available in these ratings to operate at 3600 r.p.m., direct-connected, 250 volts.

GEARED TURBINE GENERATOR SETS from 75 to 400 kw. inclusive are available in standard units, consisting of high-speed turbines and speed reducing gears direct-connected to either an A.C. or D.C. generator. These sets are used to obtain minimum steam consumption by operating the turbine at its most efficient speed, which is different from that of the standard high-speed generator at either 50 or 60 cycles. D.C. geared units operate at 1800 r.p.m. for ratings of 75 and 100 kw. All ratings from 125 to 400 kw. inclusive operate at 1200 r.p.m. See standard high-speed D.C. generators, p. 15-31.

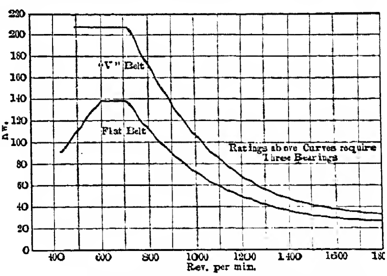


Fig. 7. Maximum Rating of 2-bearing A.C. Generators for Belt Drive

Standard Ratings in kw. of A.C. (0.8 power factor) and D.C. generators are: 75, 100, 125, 150, 200, 250, 300 and 400 kw.

Standard Voltage Ratings are: A.C., 240, 480, 600, 2400 volts. D.C., 125 and 250 volts, for 2-wire, or 125/250 volts for 3-wire.

Approximate Prices of A.C. and D.C. turbine-driven generator units, including turbines, in standard industrial sizes, are shown in Fig. 8. All D.C. generators are 250 volts and all A.C. generators are 0.8 power factor, 3-phase, 60 cycles. Prices for 50-cycle turbine-generator sets are approximately 12% higher; 25-cycle sets are approximately 20 to 25% higher.

Weights and Dimensions of turbine generator sets corresponding to curve A, Fig. 8, are given in Table 4. Table 5 applies to curves B and C, Table 6 to curve D and Table 7 to curve E.

EXCITERS for A.C. generators are supplied direct-connected or for belt drive. The standard exciter is shunt-wound, rated at 125 volts, 40° C. temperature rise, with 1.15 service factor.

Table 3.—Weights and Dimensions of Standard High-speed A.C. Generators

60 cycles, 3-phase

Kva.	Revolutions per Minute	WR^2	Maximum Excitation, kw.	Net Weight, lb.*	Dimensions, in.* High	Wide	Long
6.3	1800	4.9	0.225	435	18	18	32
	1200	15.	...	765	22	21	40
	720	85.	.65	1,540	29	28	48
12.5	1800	6.4	.25	485	18	18	34
	1200	15.	.6	765	22	21	40
	720	85.	.8	1,540	29	28	48
25	1800	19.5	.4	955	24	23	42
	1200	39.	.8	1,100	25	25	43
	720	85.	1.2	1,540	29	28	48
63	1800	75.	...	1,950	29	28	51
	1200	102.	1.25	1,775	29	28	52
	720	225.	1.8	2,375	32	31	57
	514	490.	2.7	4,135	45	44	61
125	1800	175.	1.2	2,950	32	31	61
	1200	225.	1.8	2,375	32	31	57
	720	490.	...	4,140	45	44	61
	514	1740.	3.2	4,950	55	54	61
187	1800	470.	1.6	5,040	39	39	68
	1200	450.	2.25	3,635	39	39	66
	720	1140.	3.8	5,790	50	49	67
	514	2400.	4.0	6,575	55	54	65
250	1200	535.	2.4	4,085	55	54	65
	720	1140.	4.4	5,790	50	49	67
	514	3400.	4.5	7,760	55	54	79
375	1200	800.	3.3	4,955	39	39	73
	720	1900.	5.0	7,435	50	49	76
	514	4000.	5.3	9,200	55	54	82
625	1200	1170.	...	7,600	45	44	84
	720	3000.	6.6	10,400	50	49	84
	514	6400.	8.0	12,500	55	54	90
750	1200	1400.	5.0	9,200	45	44	88
	720	4800.	6.8	11,500	55	54	86

* Weights and dimensions of exciters are not included.

Table 4.—Dimensions and Weights of Condensing Turbine-generator Sets

500 to 7500 kw., 60 cycles, 3600 r.p.m.

	Kw. Rating at 0.8 Power Factor 500	625	750	1000	1250	1500	2000	2500	3000	3500	4000	5000	6000	7500
Dimensions, in.														
Length* (A)	188 3/4	201	213 1/2	212 3/4	228 1/2	248 1/2	266	298 1/2	310 1/2	317 1/2				364
Length* (B)	161 1/2	173 3/4	188	185 1/4	199	219	224 1/2	257		276				316
Width	46 1/2	57 1/2	57 1/2	70 1/2	70 1/2	90	103		124	124				132
Height	62	62	68	69 1/2	85	85	85		85	85				
Approximate Weight, 1000 lb.														
With Exciter	20.6	25.2	35.5	39.0	41.5	55.5	69.0	78.0	94.0	98.0	108.0	118.0	129.1	142.4
Without Exciter	19.1	23.7	34.0	37.5	40.0	54.0	67.0	76.0	92.0	96.0	106.0	115.5	126.5	139.8
Heaviest Part† C	6.7	10.8	19.4	20.5	22.0	36.0	44.0	45.0	59.0	59.0	60.5	62.0	64.0	64.0
Heaviest Part† D	3.0	5.0	8.5	9.5	10.4	19.0	25.0	25.5	28.0	28.0	30.0	30.0	36.0	36.0
Turbine	8.0	12.1	21.0	22.5	24.0	38.0	46.0	47.0	62.0	62.0	63.5	65.0	68.0	68.0
Generator Stator	5.1	5.8	6.8	7.8	9.2	10.8	12.6	15.9	18.0	21.1	22.8	28.0	33.3	39.6
Generator Rotor	1.9	2.2	2.5	2.9	3.3	3.8	4.8	6.2	6.9	7.5	8.8	9.7	12.5	14.3

* A = with exciter; B = without exciter. † C = before erection; D = after erection.

Standard Exciter Ratings are: 1, 1 1/2, 2, 3, 5, 7 1/2, 10, 15, 20, 25, 30, 40, 50, 60 and 75 kw. Overhung direct-connected exciters are available for all generators of bracket-type bearing construction, as high-speed A.C. generators. With pedestal-type bearing construction, for example, low-speed engine-driven A.C. generators, exciter armatures are overhung from the generator shaft extension and the stator or field structure is supported by a sub-base built up from the main generator base.

Exciter Prices are similar to those of small standard high-speed D.C. generators. See Fig. 11.

MOTOR-GENERATOR EXCITER SETS often can be used to advantage where a separate source of power is available, especially to excite generators of very low speed. A direct-connected exciter here would be comparatively large.

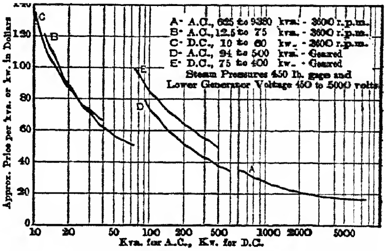


Fig. 8. Approximate 1935 Prices of Standard Turbine-driven A.C. and D.C. Generator Units in Industrial Service, Including Condensing Turbines

Table 5.—Dimensions and Weights of Direct-connected Turbine-generator Sets
10 to 60 kw., 3600 r.p.m.

Kilowatts	Exhaust Diameter, in.	A.C., 60-cycle, 3-phase, 0.8 power factor						Direct Current					
		Dimensions, in.					Weight	Dimensions, in.					Weight
		Overall			Bedplate			Overall			Bedplate		
		Long	Wide	High	Long	Wide		Long	Wide	High	Long	Wide	
10	4	82	25	26	60	22	1160	66	25	26	37	19	1100
15	4	87	25	26	65	22	1475	71	25	26	39	22	1180
20	4	87	25	26	65	22	1475	77	25	26	43	22	1400
25	6	88	40	33	63	29	2250	77	38	33	43	29	1750
30	6	....	....	....	....	....	.....	79	38	33	45	29	1850
35	6	88	40	33	63	29	2250	....	....	....	....	....	.....
40	6	....	....	....	....	....	.....	96	46	42	64	25	2750
50	6	116	46	42	83	25	3250	96	46	42	64	25	2750
60	6	116	46	42	83	25	3800	96	46	42	64	25	3090

Table 6.—Dimensions and Weights of A.C. Condensing, Geared Turbine-generator Sets
75 to 400 kw., 60 cycles, 3-phase, 1200 r.p.m.

Kilowatts	Inlet Diameter, in.	Exhaust Diameter, in.	Dimensions, in.						Approximate Net Weight, lb.	
			Overall				Bedplate			
			Length		Width	Height	Length	Width	With Exciter	Without Exciter
			With Exciter	Without Exciter						
75	4	14	144	129	63	60	99	36	10,000	9,700
100	4	14	149	134	64	62	102	38	10,800	10,500
125	4	14	153	138	66	64	106	40	11,600	11,300
150	4	14	158	143	67	66	109	41	12,400	12,000
200	4	14	167	152	70	75	116	44	13,900	13,500
250	4	14	177	162	73	79	123	47	15,400	14,900
300	4	14	186	171	76	80	130	50	17,000	16,500
350	4	20	195	180	79	84	137	54	18,500	18,000
400	4	20	204	189	81	87	144	57	20,000	19,500

Table 7.—Dimensions and Weights of D.C. Condensing, Geared Turbine-generator Sets
75 to 100 kw., 1800 r.p.m.; 125 to 400 kw., 1200 r.p.m.

Kilowatts	Inlet Diameter, in.	Exhaust Diameter, in.	Dimensions, in.									Approximate Net Weight, lb.	
			Overall					Bedplate					
			Length			Width	Height	Length			Width	125 Volt	250 and 125/250 Volt
			125 Volt	250 Volt	125/250 Volt			125 Volt	250 Volt	125/250 Volt			
75	4	14	144	144	150	63	60	96	96	102	36	10,000	10,000
100	4	14	157	149	155	64	62	106	100	106	38	11,300	11,100
125	4	14	....	153	159	66	64	....	103	111	40	12,600	12,300
125	4	14	161	....	....	66	64	111	....	....	40	12,600	12,300
150	4	14	....	158	164	67	66	....	110	116	41	13,900	13,500
150	4	14	166	....	....	67	66	116	....	....	41	13,900	13,500
200	4	14	175	167	173	70	75	119	119	125	44	16,500	15,800
250	4	14	185	177	183	73	79	128	128	134	47	19,200	18,000
300	4	20	194	186	192	77	84	138	138	144	50	21,800	20,400
400	4	20	212	204	210	81	87	156	156	162	57	27,000	25,000

4. DIRECT-CURRENT GENERATORS

A direct-current generator produces voltage, which when applied to an external circuit causes a current, constant in direction, to flow. The magnetic flux produced in the generator develops alternating current in the armature conductor, which, by means of a commutator, is caused to flow in but one direction.

ENGINE-DRIVEN D.C. GENERATORS usually are supplied without shaft, base or bearing, and are mounted by the engine builder. They are available for either 2- or 3-wire service.

Voltage Rating.—Standard voltages are 125 and 250 for 1000 kw. and less, and 250 volts above 1000 kw. For 115- and 230-volt motor circuits, generator voltages of 125 and 250 are recommended. See N.E.M.A. standards for D.C. generators. All generators will deliver rated current output at all voltages between 80% and 100% of rated voltage.

Compound-wound Generators are supplied flat compounded, *i.e.*, with windings which will give the same voltage at no load and full load, when operated at constant speed at a temperature equivalent to that which would be attained after a continuous run at rated load, and with field rheostats set to obtain rated voltage at rated load, and left unchanged. Brushes are set approximately on neutral. The curve of voltage *vs.* load current then will not vary more than 3% from a straight line between no load and full load.

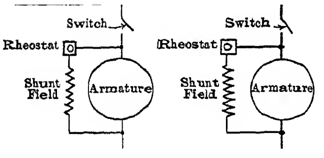


Fig. 9. Connections for Parallel Operation of Shunt-wound D.C. Generators

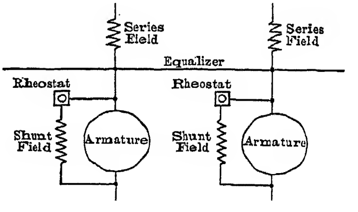


Fig. 10. Connections for Parallel Operation of Compound-wound D.C. Generators

To compensate for line drop, D.C. generators can be over-compounded to give higher voltage at full load than at no load, *i.e.* 120 volts at no load to 125 volts at full load, or 240 volts at no load to 250 volts at full load. The compounding curve may vary from 3 to 4% from a straight line connecting these two voltages.

Commutation.—D.C. generators are designed to commute successfully all loads up to 200% of rated current without burning or injuring brushes or commutator. Full load can be thrown on and off without appreciable sparking and without shifting brushes.

Speed Range.—Rated speed corresponds to A.C. 60-cycle speed. Permissible variation is a range of ±3% from rated speed without change in capacity, temperature guarantees or price. Permissible overspeed is 25%.

Rotation of standard generators is clockwise facing commutator end.

Overloads.—Generators will commutate successfully, for 1 min., loads of 200% of continuous rated amperes, with rheostats set for rated load excitation. No temperature guarantee is made. Temperature rise, by thermometer, of standard D.C. engine-driven generators, operating continuously under rated conditions, should not exceed:

	Time Rating	Core and Windings	Commutator	Bare Copper Winding
100% of rated current	Continuous	40° C.	55° C.	50° C.
125% of rated current	2 hr.	55° C.	65° C.	65° C.

Field Rheostats.—Generators rated at 250 kw., 125 volts, and 500 kw., 250 volts and smaller, are supplied with hand-operated back-of-board or chain-operated field rheostats. Generators rated at 300 kw., 125 volts, 600 kw., 250 volts and 1750 kw., 600 volts and larger, are supplied with motor-operated rheostats.

Service Factor of engine-driven generators is 1.15, *i.e.* when operated at rated speed and voltage they will carry continuously 1.15 × rated load.

Efficiencies as determined include the following losses: I^2R of armatures and fields at 75° C., brush contact, open circuit core, brush friction, stray load, field rheostat, and bearing and windage. Stray load losses are included as 1% of the output, and D.C. brush contact losses are based on a brush drop of 2 volts.

Parallel Operation.—Compound-wound D.C. generators will operate in parallel with other compound-wound generators where: *a*, load voltage characteristics are similar; *b*, equalizing connections are provided and are of the same polarity; *c*, at their respective rated loads, voltage drop from equalizers through the series fields to the bus bars is the same for all generators; it may be necessary to install resistance to get this condition; *d*, speed regulation of prime movers from no load to full load is the same. Figs. 9 and 10 show parallel connections for shunt- and compound-wound generators.

Standard Ratings of engine-driven generators are: 25, 30, 35, 40, 50, 60, 75, 100, 125, 150, 200, 250, 300, 350, 400, 500, 600, 700, 800, 900, 1000, 1250 and 1500 kw. Larger standard sizes are, for low-speed generators, 1750, 2000, 2250, 2500, 3000, 3500, 4000, 4500 and 5000 kw.

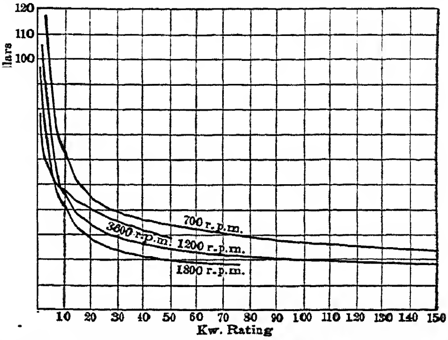


FIG. 11. Approximate 1935 Prices of Standard High-speed General-purpose D.C. Generators

STANDARD HIGH-SPEED D.C. GENERATORS for general purposes, belt driven or direct connected, are supplied with shaft and two endshield-type bearings. When supplied for belt drive, pulley and sliding base are included. The belt drive limitations of Fig. 7 apply for D.C. generators.

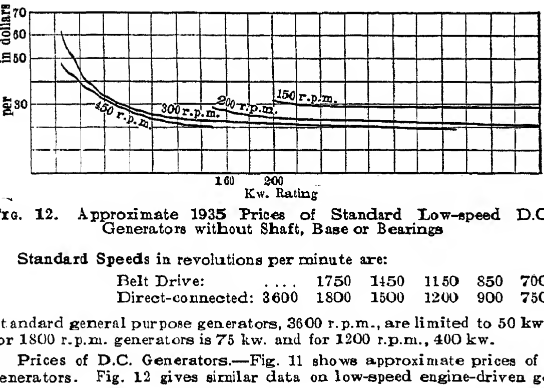


FIG. 12. Approximate 1935 Prices of Standard Low-speed D.C. Generators without Shaft, Base or Bearings

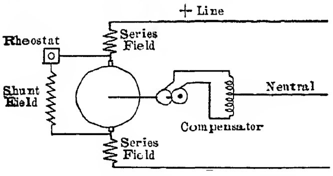
Standard Ratings are: 1, $1\frac{1}{2}$, 2, 3, 5, $7\frac{1}{2}$, 10, 15, 20, 25, 30, 40, 50, 60, 75, 100, 125, 150, 200, 250, 300, 400, 500, 600, 750, 1000, and 1250 kw.

Standard Speeds in revolutions per minute are:

Belt Drive:	....	1750	1450	1150	850	700	575	500
Direct-connected:	3600	1800	1500	1200	900	750	600	514

Standard general purpose generators, 3600 r.p.m., are limited to 50 kw. and less. The limit of size for 1800 r.p.m. generators is 75 kw. and for 1200 r.p.m., 400 kw.

Prices of D.C. Generators.—Fig. 11 shows approximate prices of high-speed general purpose generators. Fig. 12 gives similar data on low-speed engine-driven generators.



G. 13. Connections of 3-Wire D.C. Generators

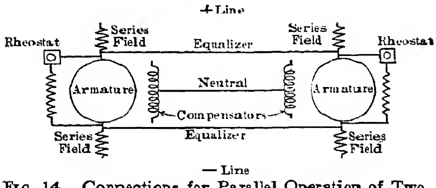


FIG. 14. Connections for Parallel Operation of Two Compound-wound 3-wire D.C. Generators

THREE-WIRE GENERATORS designed for operation on D.C. 3-wire systems are standard above $7\frac{1}{2}$ kw. The armature winding is tapped at diametrically opposite points and leads are brought out to two collector rings on the generator shaft. An auto-transformer is connected across these two rings, its mid-tap forming the neutral. Standard equipment provides for 10% unbalanced current in the neutral. Fig. 13 shows 3-wire connections, and Fig. 14 shows parallel connection of two 3-wire generators. The neutral

provides for load at half the generator voltage, and both motor and lighting loads can be supplied from one generator.

Approximate Weights, Dimensions and Efficiencies of representative high-speed D.C. belt-driven generators are given in Table 8. Table 9 gives similar information for representative direct-›nnected generators. Table 10 gives data for engine-driven D.C. generators.

SHUNT-WOUND GENERATORS are excited from a single winding, of comparatively igh resistance, connected across the brushes of the generator, in parallel with the arma-ure and the load. With a given field rheostat setting, voltage drops as load is applied. uch generators are used where load remains unchanged over long periods as in electro-ytic plants, furnaces, electro-chemical and exciter services. Shunt generators also may e separately excited by connecting the shunt field to a source independent of the arma-ure circuit. Such generators are stable from zero to rated voltage and should be used where stable operation is required over a wide voltage range. Typical applications are oosting, Ward Leonard systems, where the voltage is carried through zero from maximum ositive to maximum negative for reversing purposes and speed control of motors by rmature or voltage control.

COMPOUND-WOUND GENERATORS are the most commonly used D.C. gen-rators for all kinds of service. They have a shunt winding connected across the generator erminals, and a series winding of fewer turns of heavy wire through which the entire urrent output flows. The shunt winding supplies most of the excitation; the series

Table 8.—Dimensions and Efficiencies of Standard D.C. Generators for Belt Drive, 125 Volts

Kilowatts	Revolutions per Minute	Net Weight, lb.*	Dimensions, in.			Full Load Efficiency, per cent
			Long	Wide	High	
1	1750	160	18	10	10	76.5
	1450	200	20	11	11	78.5
	1150	200	20	11	11	74.8
1½	1750	200	20	11	11	79.2
	1450	220	21	11	11	80.7
	1150	275	25	13	12	76
2	1750	220	21	11	11	81
	1450	275	25	13	12	79
	1150	275	25	13	12	75
	850	355	28	14	14	78
3	1750	275	25	13	12	79
	1450	275	25	13	12	79
	1150	355	28	14	14	80
	850	570	34	18	18	76
5	1750	355	28	14	14	84
	1450	355	28	14	14	80
	1150	570	34	18	18	81
	850	640	36	18	18	80
10	1750	570	34	18	18	85
	1450	640	36	18	18	84
	1150	640	36	18	18	85
	850	830	40	21	21	84
	575	1310	45	24	24	84
20	1750	830	40	21	21	86
	1450	930	41	21	21	86.5
	1150	1280	45	24	24	87
	850	1310	45	24	24	86
30	1750	1280	45	24	24	87
	1450	1320	45	24	24	87

* Includes base and pulley.

Table 9.—Standard D.C. Generators for Direct Connection, 125 Volts

Kilowatts	Revolutions per Minute	Net Weight, lb.	Dimensions, in.			Full Load Efficiency, per cent
			Long	Wide	High	
10	600	1200	50	27	28	83.3
20	900	1200	50	27	28	86.2
	600	1740	57	30	31	85.6
30	1200	1200	50	27	28	87.1
	900	1740	57	30	31	87
	600	2580	69	33	34	87.5
50	1800	1325	51	27	28	89.6
	1200	1740	57	30	31	88.4
	900	2580	69	33	34	88.5
	600	3100	70	33	35	87.8
75	1800	1915	60	30	31	89.9
	1200	2580	69	33	34	89.5
	900	2770	70	33	34	89.6
	600	4625	85	39	40	88.8
100	1200	2,770	70	33	34	90.2
	900	3,500	75	33	35	89.8
	600	6,200	82	41	43	89.2
200	1200	6,520	68	47	40	90.8
300	900	9,200	77	53	42	91.2
400	750	13,000	89	63	50	91.4
500	750	16,550	92	69	53	91.5
*600	900	12,950	86	64	50	93.5
*750	900	15,100	95	64	50	93.5
*1000	750	21,700	99	79	58	93.5
*1250	750	24,200	103	81	64	93.1

* 250 Volts.

Table 10.—Sizes and Efficiencies of Engine-driven Direct-current Generators
125 volts, 2-wire, compound-wound

Kilowatts	R.p.m.	Efficiencies at 1.25 Load	Full Load	3/4 Load	1/2 Load	Approx. Net Weight, lb.	Kilowatts	R.p.m.	Efficiencies at 1.25 Load	Full Load	3/4 Load	1/2 Load	Approx. Net Weight, lb.
75	450	87.2	88.3	89.0	89.5	4200	175	327	89.3	90.2	91.0	91.2	8,800
75	400	87.1	88.2	89.1	89.6	4650	175	300	90.9	91.5	91.7	91.2	9,400
75	360	88.2	89.0	89.6	90.0	4960	200	450	91.0	91.6	92.0	91.9	7,900
75	327	87.8	88.9	89.7	90.0	5200	200	400	90.7	91.4	91.8	91.7	8,500
75	300	87.5	88.7	89.7	90.2	5500	200	360	89.8	90.7	91.4	91.5	9,150
100	450	89.2	90.1	90.6	90.7	5200	200	327	91.7	92.2	92.2	92.0	9,550
100	400	88.7	89.5	90.1	90.2	5500	200	300	90.4	91.2	91.7	91.7	10,100
100	360	88.6	89.5	90.2	90.4	5900	250	400	92.0	92.6	92.6	92.2	9,650
100	327	89.6	90.3	90.8	90.9	6200	250	360	91.5	92.2	92.6	92.5	10,350
100	300	88.6	89.6	90.4	90.7	6550	250	327	91.0	91.7	92.2	92.3	11,000
125	450	89.9	90.5	90.8	90.9	5900	250	300	90.2	90.8	91.1	90.7	11,700
125	400	90.8	91.4	91.7	91.3	6300	300	400	91.5	92.1	92.6	92.5	10,800
125	360	90.1	90.8	91.4	91.4	6750	300	360	91.3	91.7	91.8	91.3	11,600
125	327	90.8	91.6	92.2	92.1	7200	300	327	92.0	92.4	92.6	92.2	12,300
125	300	90.7	91.3	91.8	91.7	7650	300	300	91.8	92.3	92.5	92.3	13,200
150	450	91.2	91.8	92.2	91.9	6650	350	360	91.8	92.1	92.1	91.7	12,900
150	400	90.4	91.2	91.8	91.8	7150	350	327	91.7	92.1	92.5	92.3	13,800
150	360	90.3	91.2	91.7	91.9	7600	350	300	91.3	92.0	92.5	92.5	14,700
150	327	91.6	92.1	92.5	92.5	8200	400	360	91.4	91.9	92.2	91.9	14,100
150	300	89.3	90.0	90.6	90.5	8400	400	327	91.7	92.5	92.8	92.6	15,100
175	450	91.5	92.0	92.3	92.0	7400	400	300	91.7	92.3	92.7	92.7	16,200
175	400	91.2	91.7	92.0	91.7	7850	500	327	92.5	92.9	93.1	92.7	17,700
175	360	90.4	91.0	91.4	91.1	8400	500	300	91.8	92.3	92.7	92.5	19,200

winding increases excitation with increase of load. Compound-wound generators are used to supply constant voltage with continuous change in load. Compounding generally is flat, but in mining and railway service generators often are over-compounded to give higher voltage at full load than at no load, thus compensating for increased line drop as load is applied. The voltage of a flat-compounded D.C. generator averages 3 to 4% higher near half load than at either no load or full load.

5. VOLTAGE REGULATION

To avoid drop in terminal voltages of generators when load is applied, they must be so compounded as to increase total excitation, or field excitation must be adjusted either manually or automatically. Field current required to maintain full rated voltage of standard A.C. generators, 0.8 power factor, at no load is 50% of that required at full load.

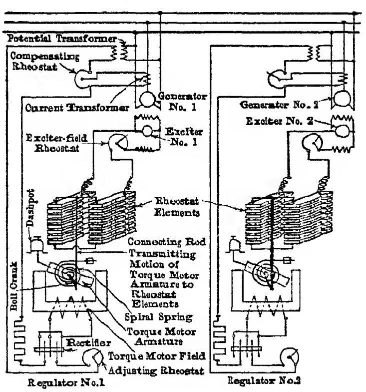


FIG. 15. Connection Diagram of Two Direct-acting Field Rheostat Type Generator Voltage Regulators Controlling Two Paralleled A.C. Generators

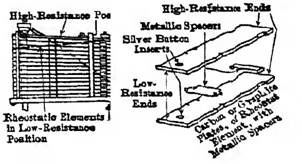


FIG. 16. Elements of Direct-acting Generator Voltage Regulator

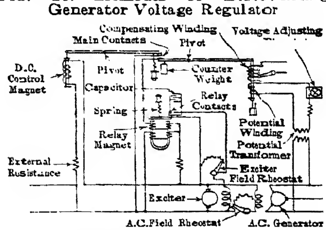


FIG. 17. Connection Diagram of Vibrating-contact Type Generator Voltage Regulator

AUTOMATIC GENERATOR VOLTAGE REGULATORS, in general, fall into three classes as follows:

Direct-acting Field Rheostat Type, which operates in shunt field of self-excited exciters, will control A.C. generators, synchronous condensers and small and medium size synchronous motors. Fig. 15 is a diagram of this type controlling two paralleled A.C. generators. The regulator operates only when a change in excitation is required. The torque motor operates against a spiral spring and moves the rheostat element, Fig. 16, to vary the field strength of the exciter. While designed to control but one exciter, it can control several generators in parallel served by the same exciter.

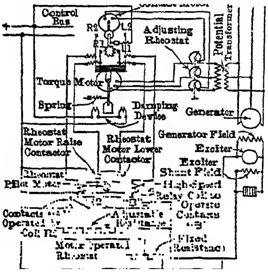


Fig. 18. Connection Diagram of Exciter Field Rheostat Type Generator Voltage Regulator

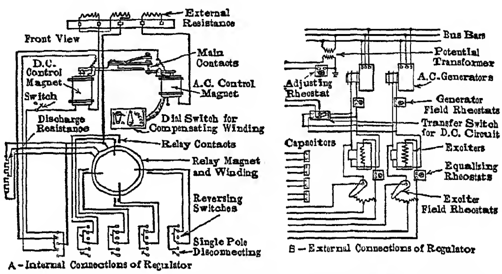


Fig. 19. Connection Diagram of Single Vibrating-contact Type Voltage Regulator Controlling Two Generators with Individual Non-paralleled Exciters

Vibrating Contact Type Regulators, Fig. 17, for small and medium capacity A.C. machines, operate in the shunt field of self-excited exciters. These regulators are rated 100/125 volts A.C., 125 or 250 volts D.C., and operate on any frequency between 25 and 125 cycles, over an exciter voltage range of 50 to 125 volts for 125-volt exciters, and 100 to 250 volts for 250-volt exciters. The two main parts are a D.C. control and an A.C. control. The former is a D.C. regulator with a main control magnet and relay magnet connected across the exciter lines. The relay contacts shunt the exciter field rheostat. The operation maintains voltage which varies with the demands of the A.C. control magnet connected to the A.C. bus.

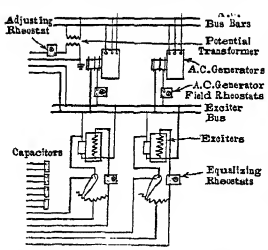


Fig. 20. External Connecti ns ingle Voltage Regulator Controlling Two Generators with Individual Paralleled Exciter

Exciter Field Rheostat Type with High-speed Features is suitable for any size or type of A.C. synchronous machine. It has no continuously-vibrating contacts. It operates only on A.C. voltage changes. Excitation is controlled by automatic adjustment of the exciter field rheostat, and by high-speed relays which cut in or out the entire regulating resistance in the exciter field with large changes in A.C. voltage. It is used widely in central stations to maintain system stability. In industrial use, it usually is limited to special cases of relatively large transient loads where rapid generator voltage recovery is required. Fig. 18 shows typical connections.

AUTOMATIC VOLTAGE CONTROL OF PARALLELED A.C. MACHINES.—Machine ratings and type of excitation system determine the type of regulator and method of control to be used. The three methods commonly used are:

Unit Exciters with Individual Regulators.—See Fig. 15. The exciters are operated non-parallel. The voltage of each A.C. machine is automatically and independently controlled by its own regulator. Division of reactive kva. among machines is controlled by A.C. compensation of the individual regulators, and there is no problem of load division among exciters. All three types of regulators may be applied. Voltage regulation is completely automatic, and no manual adjustment of field rheostats is required. Exciter or regulator trouble affects only one machine.

Unit Exciters with Common Regulator.—Each A.C. machine has an individual exciter operated non-parallel, but a common regulator controls all A.C. machines and exciters. See Fig. 19. The vibrating-contact regulator is the only standard type designed for such operation. A.C. potential for the regulator is taken from the station bus, and the D.C. supply is taken from any one exciter at a time, with transfer arrangement for connection

to any exciter. Proper division of reactive kva. among A.C. machines, and division of load among exciters, require manual adjustment of exciter equalizing rheostats, or of main generator field rheostats, or of both.

Common Exciter Bus and a Common Regulator.—A.C. machine fields are excited in parallel from a variable-voltage exciter bus, supplied by one or more exciters. All exciters

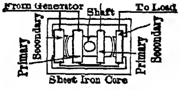


FIG. 21. Diagram of Induction Type Feeder Voltage Regulator

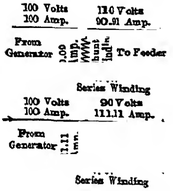


FIG. 22. Voltage and Current Relations in Single-phase Induction Feeder Voltage Regulator

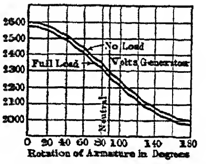


FIG. 23. Boosting and Lowering of Feeder Voltage by Induction Regulator

and A.C. generators are controlled by one regulator. If two or more exciters operate in parallel, only the vibrating contact type regulator can be used. With only one exciter, the regulator may be of any type, depending on machine ratings. This method is not entirely automatic. A.C. bus voltage is controlled automatically, but division of reactive kva. and exciter load must be manually adjusted. Fig. 20 with Fig. 19A show the regulator connections. Difficulties which may be encountered are: Exciters, operating in parallel, with different rates of response (voltage build-up) to regulator action, will cause unequal loading of exciters during load changes; exciters may not share light loads properly; paralleling an incoming exciter with the fluctuating voltage of the exciter bus may be difficult; continual manual adjustment of rheostats may be required to maintain proper division of exciter load and reactive kva.

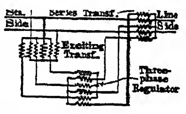


FIG. 24. Regulation of 3-phase High-voltage Circuit

VOLTAGE REGULATION OF D.C. GENERATORS.—Voltage of small and medium capacity D.C. generators may be controlled by the direct-acting regulator, Fig. 15. The rectifier is omitted and the torque motor is operated directly from the generator bus. The rheostatic type regulator, with or without high speed features, is suitable for any size of machine, and for any range of voltage, speed or current control. It also is suitable for control of separately-excited D.C. machines.

INDUCTION FEEDER VOLTAGE REGULATOR is a variable-ratio auto-transformer with primary and secondary windings. The primary is connected across the feeder to be controlled; the secondary is in series with the feeder. Fig. 21 is an elementary diagram. The primary winding is placed on a core which can be rotated to produce a voltage, variable in value and direction, in the secondary winding. Fig. 22 shows current and voltage relations in a single-phase regulator, for a 10% boost and lower. In maximum boost position, about 9% of the generator current flows through the primary winding and back to the generator. This amount is deducted from the current in the feeder, but the voltage on the feeder is boosted 10%. In the maximum lowering position, current in the regulator primary winding is reversed, increasing feeder current 11% and lowering voltage 10%. Fig. 23 shows the boosting and lowering of feeder voltage caused by rotation of an induction feeder regulator armature.

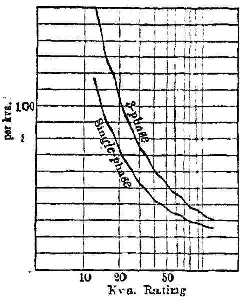


FIG. 25. Approximate 1935 Prices of Automatic Indoor Type Induction Feeder Voltage Regulator

Regulators can be built for any voltage and current for which it is practicable to build motors or generators of corresponding sizes. However, regulators are used on feeders where their kva. capacity is only about 10% of feeder capacity, and they must be wound for voltages and currents corresponding to much larger machines. Exciting and series transformers may be required. To control a circuit of high current but small kva. capacity, a series transformer is used in the regulator secondary only. To control a circuit of high voltage but of small kva. capacity both exciting and series transformers are required. Fig. 24 shows connections for these auxiliary devices for a 3-phase circuit.

Automatic Operation of Induction Feeder Regulators.—A modern distributing station may deliver power over a large number of individual feeders. Since the amount of voltage adjustment required depends on the load and voltage drop on any one feeder, automatic regulation of individual feeders is essential to good regulation on the entire system. Regulators for automatic service are motor operated.

Approximate Prices (1935) of standard 2400-volt automatic indoor type induction regulators which will vary feeder voltage ± 10%, are shown in Fig. 25. Outdoor equipments run about 15% higher.

Weights and Dimensions are given, approximately, in Table 11.

Table 11.—Dimensions and Weights of Standard Induction Feeder Regulators

Single-phase					3-phase				
Kva. Rating	Weight, lb.	Approximate Dimensions, in.			Kva. Rating	Weight, lb.	Approximate Dimensions, in.		
		Width	Depth	Height			Width	Depth	Height
12	1475	19	27	64	10.4	1400	19	27	62
18	1785	21	33	60	20.8	2300	21	33	72
24	1785	21	33	60	31.2	2425	21	33	74
36	2360	21	33	70	41.6	2600	23	33	76
48	2535	23	35	72	62.4	3575	28	38	82
60	2855	25	34	76	83.2	4300	28	38	90
72	3295	26	37	79	104	5025	38	48	90
96	4335	31	42	80	125	5425	38	48	93
120	4935	32	43	85					

CONVERSION EQUIPMENT

Power generated as alternating current frequently must be changed to some other form before it can be utilized for industrial application. It may be changed in voltage, phase, frequency, from alternating to direct, or from direct to alternating current.

VOLTAGE CONVERSION on A.C. circuits is by means of transformers; on D.C. circuits, motor-generators are used.

PHASE CONVERSION.—Single-phase loads of sufficient magnitude to cause objectionable unbalance on a 3-phase system are best converted by use of a motor-generator. Phase conversion between polyphase systems, as 3-phase to 2- or 6-phase may be accomplished by transformers with special windings and taps (see p. 15–18).

FREQUENCY CONVERSION, necessary for interchange of power between two systems of different frequency, is most simply done by a motor-generator set. Another application is machinery operated by high-speed motors, requiring a frequency higher than the power supply; for example wood-working machinery.

1. MOTOR-GENERATOR SETS

CLASSIFICATION AND USES.—1. D.C. to D.C., used as balancer sets on 3-wire systems supplied by 2-wire generators, battery-charging sets, and booster sets for boosting voltage. 2. A.C. to A.C., used for frequency and phase conversion. 3. A.C. to D.C., used for all types of service, battery charging from A.C. supply, generator and motor excitation, railways, electrolytic work, material handling equipment, speed control, machine tool circuits and testing. Standard sets have a wide range of rating, with either induction or synchronous motor drive. 4. D.C. to A.C., used to a limited extent for special applications.

INDUCTION MOTOR-GENERATOR SETS for general purpose use are standardized for 25-, 50- and 60-cycle systems. They have a 2-bearing, squirrel-cage induction motor with a short shaft extension, connected through a solid coupling to the single-bearing generator. Foundations to prevent deflection of bases are necessary. Sets larger than 125 kw. have speed limit devices. Generators for 3-wire service have two collector rings for deriving the neutral.

Voltage Ratings are the same as for standard motors and generators used separately.

Temperature Rise Rating is 40° C. for continuous operation, based on 40° C. ambient temperature.

Overloads of 150% of rated generator current, in amperes, are permissible with successful commutation as defined by the A.I.E.E.

Service Factor is 1.15 for all except 50-cycle sets.

Used as Exciters for A.C. generators, rating should be not less than 110% of excitation required for all generators to be excited.

Standard kw. Ratings of 25-, 50- and 60-cycle sets are: 1, 1.5, 2, 3, 5, 7.5, 10, 15, 20, 25, 30, 40, 50, 60, 75, 100, 125 and 150 kw.

Standard Speeds are: 60-cycle, 1 to 50 kw., inclusive, 1800 r.p.m.; 60 to 150 kw., inclusive, 1200 r.p.m.

50-cycle, 1 to 50 kw., inclusive, 1500 r.p.m.; 60 to 150 kw., inclusive, 1000 r.p.m.

25-cycle, 1 to 50 kw., inclusive, 1500 r.p.m.; 60 to 150 kw., inclusive, 750 r.p.m.

Approximate List Prices (1935) of induction motor-generator sets for general purposes are given in Fig. 1.

Efficiencies, Dimensions, Weights and Power Factors are given in Table 1.

SYNCHRONOUS MOTOR-GENERATOR SETS are built in small sizes as follows: 50 to 150 kw., 1200 r.p.m., 60 cycles; 1000 r.p.m., 50 cycles; 750 r.p.m., 25 cycles. Motors can be wound for either 2-phase or 3-phase operation. Standard power factor is 0.8 leading. Motor voltages are 220, 440, 550, and 2200 volts. Standard large sizes range from 200 to 1000 kw. at 125 volts; to 4000 kw. at 250 volts; from 300 to 6000 kw. at 600 volts. Three-unit, 4-bearing sets, with two generators each of half the set rating, are used for sizes of 300 kw. and over at 125 volts. Three-unit sets are used at 250 and 600 volts for 2000 kw. and over.

Temperature Rise.—Usual ratings are: 150 kw. and less, continuous operation, 40° C.; 200 kw. and over, 2-hr. rating at 125% of load, 55° C.

Standard kw. Ratings are: Small size, 25-, 50- and 60-cycle, 50, 60, 75, 100, 125 and 150 kw.; large size, 125 volts, 60-cycle, 200, 250, 300, 400, 500, 600, 800, and 1000 kw.; 250 volts, 60-cycle, 200, 250, 300, 400, 500, 600, 750, 1000, 1250, 1500, 2000, 2500, 3000 and 4000 kw.; 600 volts, 60-cycle, 300, 500, 750, 1000, 1250, 1500, 1750, 2000, 2500, 3000, 3500, 4000, 5000 and 6000 kw. Standard 25-cycle ratings are similar.

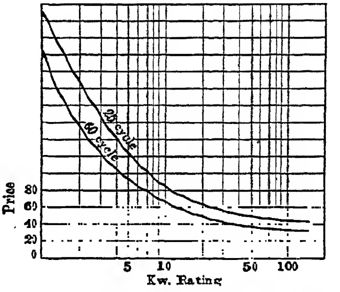


FIG. 1. Approximate 1935 Prices of Induction Motor-generator Sets

Table 1.—Characteristics of General-purpose Induction Motor-generator Sets

125 volts, D.C., 220,440 and 550 volts A.C.

Rating, kw.	3-phase, 60-cycle, A.C. Full Load Efficiency	Full Load Power Factor	Net Weight, lb.	Overall Dimensions, in. Long	Wide	High	3-phase, 25-cycle, A.C. Full Load Efficiency	Full Load Power Factor	Net Weight, lb.	Overall Dimensions, in. Long	Wide	High
1	62.3	81.0	282	30	16	14	62.0	84.0	325	31	16	14
2	69.2	80.0	385	34	17	15	67.0	71.0	491	40	19	16
5	74.0	83.5	612	43	20	19	68.0	86.0	810	45	23	21
10	75.5	86.5	1005	50	23	23	71.7	86.5	1,295	54	25	25
20	77.0	89.5	1450	58	26	25	77.4	86.0	1,880	64	27	27
30	78.3	92.0	2095	65	30	30	78.3	86.0	2,570	69	30	31
50	82.1	92.5	3110	74	27	34	78.3	90.0	3,480	76	29	36
75	83.4	85.5	5720	89	42	41	80.3	90.0	8,420	95	39	46
100	84.2	92.0	6140	94	42	42	82.2	91.5	9,900	103	42	48
125	85.0	91.0	7420	103	42	40	83.5	90.0	11,400	116	44	48
150	84.2	91.0	8850	108	42	42	83.8	92.0	13,250	119	48	50

Table 2.—Characteristics of Synchronous Motor-generator Sets, 150 kw. and Smaller

125-volts, D.C., 220,440, and 550-volts, A.C. 0.8 power factor motor

Rating, kw.	3-phase, 60-cycle, A.C. Full Load Efficiency	Net Weight, lb.	Approx. Dimensions, in. Long	Wide	High	3-phase, 25-cycle, A.C. Full Load Efficiency	Net Weight, lb.	Approx. Dimensions, in. Long	Wide	High
50	80.3	4170	83	30	38	79.1	7,200	101	34	41
60	80.8	4550	86	32	39	79.0	8,300	100	39	47
75	82.2	5480	93	33	41	80.0	9,550	105	39	47
100	83.8	6300	100	33	41	81.2	10,600	109	42	48
125	84.5	7250	106	37	40	82.5	12,500	120	49	50
150	85.3	9050	108	41	42	83.3	14,350	122	50	52

Approximate Prices (1935) of small synchronous motor-generator sets are given in Fig. 2. In the large sizes, prices average between $20 and $25 per kw. at 60 cycles with 2300- or 4000-volt motors. 25-cycle sets average about 20% higher.

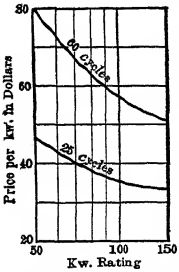


Fig. 2. Approximate 1935 Prices of Synchronous Motor-generator Sets, 125 Volts

Approximate Efficiencies, Weights and Dimensions of representative A.C. to D.C. motor-generator sets for continuous, constant-voltage service are given in Tables 2 and 3. Efficiencies are determined by including I^2R losses in armature and field windings, and in field rheostats for both motor and generator, bearing windage and friction, core, stray load, brush friction and brush contact losses. Segregated losses are measured in accordance with A.I.E.E. Standards.

REVERSED OPERATION OF SYNCHRONOUS MOTOR-GENERATOR SETS.—With the A.C. motor acting as a generator, at 0.8 power factor, the A.C. output will be 80% of the rated D.C. output, for the same temperature rise as specified for A.C. to D.C. operation.

PARALLEL OPERATION OF MOTOR-GENERATOR SETS usually is practicable. For satisfactory operation, special adjustments may be necessary after installation. See Parallel Operation of D.C. Generators, p. 15–31.

PARALLEL OPERATION OF MOTOR-GENERATORS WITH SYNCHRONOUS CONVERTERS in the same station is possible only when the machines are connected by several thousand feet of feeders, and the converters are not subject to serious A.C. voltage fluctuations. There must be sufficient minimum resistance to maintain some feeder drop from the interconnected substation to the loads. The machines inherently are unsuited for parallel operation, and careful study of A.C. supply voltage and D.C. load is necessary.

MOTOR-GENERATOR SETS FOR ELECTROLYTIC SERVICE usually are rated for continuous operation. Load is at constant current, and generator output varies with voltage. The kw. rating is determined by maximum voltage and current required continuously. A voltage range of 20% below normal usually can be obtained by field control on a shunt-wound generator; separate excitation is required for wider ranges.

D.C. TO A.C. MOTOR-GENERATOR SETS.—An important application is provision of variable-frequency A.C. supply. Within a specified range an infinite number of speeds is possible, because of the speed characteristics of the D.C. motor. A typical use of a variable frequency is in the manufacture of machine tools or business machines, where motors of various frequencies must be tested. The equipment will include a variable-speed D.C. to A.C. motor-generator set, auto-transformers with suitable taps for obtaining the required voltages at the required frequencies, a generator control and tap changing panel, and a

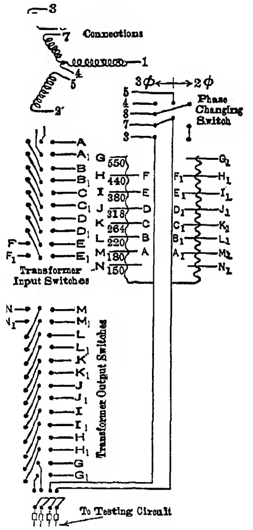


Fig. 3. Diagram of Variable Frequency, Variable Voltage Testing Set for A.C. Motors

Table 3.—Characteristics of Large Synchronous Motor-generator Sets

3-phase, 60-cycles, 2300- or 4000-volt motors

Rating, kw.	125 Volts Full Load Effic.	125 Volts Net Weight, lb.	125 Volts Approx. Dimensions, in. Long	Wide	High	250 Volts Full Load Effic.	250 Volts Net Weight, lb.	250 Volts Approx. Dimensions, in. Long	Wide	High	600 Volts Full Load Effic.	600 Volts Net Weight, lb.	600 Volts Approx. Dimensions, in. Long	Wide	High
200	85.3	12,000	112	50	49	86.8	9,900	111	42	42	.....	.....	.....	.....	.....
300	86.0	17,300	113	52	50	.....	.....	.....	.....	.....	88.0	17,400	139	59	57
400	86.4	24,200	130	67	66	87.9	17,800	124	57	54	.....	.....	.....	.....	.....
500	87.0	29,600	131	73	72	.....	.....	.....	.....	.....	89.0	27,200	129	68	58
600	86.5	29,300	180	70	61	88.7	24,400	132	69	65	.....	.....	.....	.....	.....
1000	87.5	50,500	210	79	73	89.1	38,200	141	84	75	89.6	40,625	140	85	76
1500	.....	.....	.....	.....	.....	.....	.....	.....	.....	.....	90.0	72,080	154	104	88
2000	.....	.....	.....	.....	.....	89.6	108,000	190	154	102	90.2	87,240	170	120	92
3000	.....	.....	.....	.....	.....	90.2	120,700	306	118	92	90.3	131,240	192	164	111
4000	.....	.....	.....	.....	.....	90.3	206,000	320	140	106	90.9	159,450	310	120	94
6000	.....	.....	.....	.....	.....	.....	.....	.....	.....	.....	91.0	243,600	344	164	112

speed controller for the D.C. motor. Fig. 3 shows a connection diagram for a test set. The generator can be wound for 2- or 3-phase output, and change in connections made with a triple-pole, double-throw switch, which also connects two single-phase auto-transformers for either 2-phase, 4-wire, or 3-phase, open delta, 3-wire, output. Single-phase power also is available at the same frequencies and voltages. Single-phase capacity of the generator at any given frequency is 66% of the 3-phase capacity. Generator excitation remains constant, and voltage and kw. capacity vary directly with speed.

2. SYNCHRONOUS CONVERTERS

SYNCHRONOUS CONVERTERS comprise a synchronous motor and a D.C. generator combined in one machine, with an armature winding common to both A.-C. and D.C. circuits. The effective A.C. voltage, V_{ac}, between collector rings, for a given D.C. voltage, V_{dc}, is $V_{ac} = 0.707 V_{dc} \sin (\pi/n)$, where n = number of collector rings. For a single-phase converter, $n = 2$; for a polyphase converter n = number of phases. Fig. 4 shows arrangement of windings, and Fig. 5 connections to control apparatus.

Converters are widely used where the D.C. load remains constant, as in electrolytic work. Converters, with transformers, occupy about the same floor space as motor-generator sets of the same rating, but the efficiency of the former is higher. The transformers can be located away from the converter, which often is an advantage. Since the A.C. and D.C. circuits are electrically connected through the converter, fluctuations in the A.C. supply circuit are immediately reflected in the D.C. load circuit. With wide A.C. voltage fluctuations, flashovers may occur, and the motor-generator then is preferable to the converter.

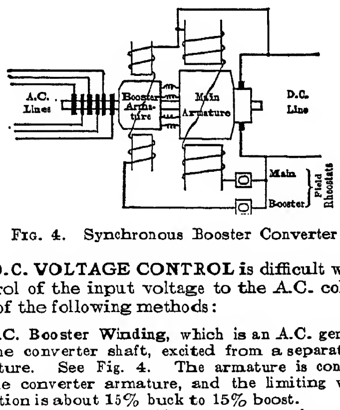


FIG. 4. Synchronous Booster Converter

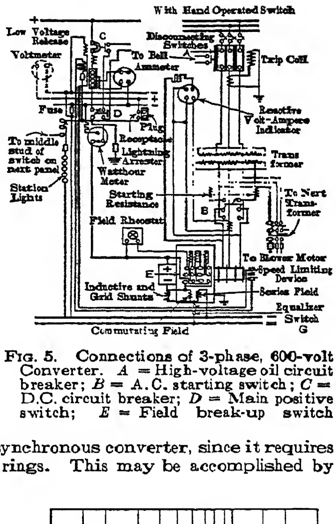


FIG. 5. Connections of 3-phase, 600-volt Converter. A = High-voltage oil circuit breaker; B = A.C. starting switch; C = D.C. circuit breaker; D = Main positive switch; E = Field break-up switch

D.C. VOLTAGE CONTROL is difficult with a synchronous converter, since it requires control of the input voltage to the A.C. collector rings. This may be accomplished by one of the following methods:

A.C. Booster Winding, which is an A.C. generator on the converter shaft, excited from a separate field structure. See Fig. 4. The armature is connected to the converter armature, and the limiting voltage variation is about 15% buck to 15% boost.

Transformer Taps.—Since power must be removed to change taps, this method is useful only to compensate for line drop. Load ratio control permits changing taps under load, and can be applied to synchronous converters. Rapid changes in voltage are not possible, as two steps are required to change from one tap to another.

Induction Regulators permit a continuous variation in D.C. voltage, but seldom are used because of high cost. If continuous variation is important, the cost of induction regulators may be warranted. Although apparatus for voltage variation reduces the efficiency of synchronous converters, it still is higher than that of any other type of conversion equipment.

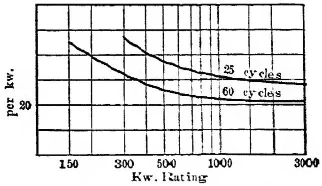


FIG. 6. Approximate 1935 Prices of Standard Synchronous Converters and Transformer Banks for Industrial Application. 2300, 4000 volts A.C.

STARTING SYNCHRONOUS CONVERTERS.—Methods are: 1. Star-delta connection of transformer primary winding, giving about 58% voltage for starting. 2. Low-

voltage secondary tap; common for manual control; a 50% tap is used to give a starting kva. of about 25% of full voltage value. 3. Separate starting transformer or auto-transformer. 4. Starting motor; a direct-connected motor, specially designed for high starting torque per ampere; is expensive, and not used unless system limitations demand it. 5. D.C. starting; the converter is started as a D.C. motor from an available D.C. supply.

Approximate Prices (1935) of 250-volt compound-wound synchronous converters with banks of three single-phase transformers, are given in Fig 6.

Efficiencies, Weights and Dimensions of synchronous converters of standard ratings for industrial service are given in Table 4.

Table 4.—Characteristics of Synchronous Converters for Industrial Service

250 volts

Kw. Rating	60 Cycle							25 Cycle						
	Speed, r.p.m.	Weight, lb.	Trans-former kva.	Full Load Effic.	Dimensions, in. Long	Wide	High	Speed, r.p.m.	Weight, lb.	Trans-former kva.	Full Load Effic.	Dimensions, in. Long	Wide	High
75	1800	2,100	83	91.0	57	32	32	......	......	......	......	......	......	......
100	1200	3,400	110	92.1	66	38	39	......	......	......	......	......	......	......
150	1200	3,800	165	92.7	75	38	40	......	......	......	......	......	......	......
200	1200	4,900	219	93.1	80	41	42	......	......	......	......	......	......	......
300	1200	6,500	330	93.7	86	42	42	750	12,200	330	94.0	96	57	55
500	1200	13,800	549	94.4	96	57	56	750	18,000	549	94.4	111	61	63
750	900	21,000	825	94.1	127	73	71	750	25,000	825	94.5	123	70	67
1000	720	26,500	1100	93.7	125	81	81	500	37,000	1100	94.0	143	88	80
1250	600	33,000	1374	94.0	140	88	83	......	......	......	......	......	......	......
1500	514	46,000	1650	93.7	148	103	93	375	52,000	1650	94.2	145	104	92
2000	450	54,000	2199	93.6	148	112	103	300	67,000	2199	93.8	147	120	104
2500	360	74,000	2750	93.8	152	130	118	250	85,000	2750	94.6	170	138	123
3000	300	94,000	3300	94.3	155	148	132	214	97,000	3300	94.6	163	150	118

3. POWER RECTIFIERS

A rectifier is a device which converts alternating to direct current, by permitting flow of current in but one direction. The essential parts are the electrodes (anode and cathode) and a medium for conducting current between them. Current flows from the anode into the arc and from it into the cathode. A starting anode is used to start the initial arc. Its grids are placed in the arc stream, to which a voltage control can be applied. Excitation anodes maintain an auxiliary arc in the vacuum tank.

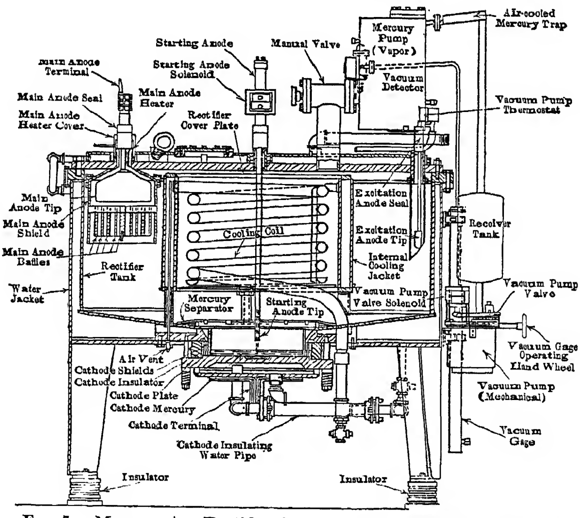


Fig. 7. Mercury Arc Rectifier, Metallic Tank, 3000 kw., 625 Volts

MERCURY ARC RECTIFIERS depend on ionized mercury vapor to conduct the current between electrodes. Fig. 7 shows the arrangement of a metallic tank mercury arc rectifier. Fig. 8 shows typical connections of a rectifier and its associated apparatus as used for railway service.

Railway Applications.—The advantages of the mercury arc rectifier for electric railway service are its high efficiency at light loads and at D.C. voltages above 550, its lack of rotating parts, and its low maintenance. Standard rectifiers for railway service are 6- and 12-phase, and range in capacity from 500 to 6000 kw. at 600, 750, 1500 and 3000 volts. A complete installation usually will comprise power transformer, interphase transformer, auxiliary and insulating transformers, rectifier, ignition and excitation equipment, switchboard and cooling system. A low-voltage, high-current transformer and loading resistor also are required to bake out the rectifier and remove foreign gases from it before placing it in service. Transformers with 6- or 12-phase secondaries are used to cut down the A.C. ripple in the D.C. output and to get the most efficient use of the transformer.

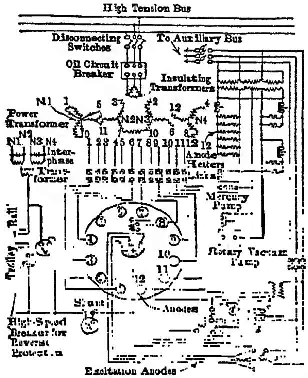


Fig. 8. Connection Diagram of 12-phase Rectifier Using Quadruple-delta Zigzag Main Transformer

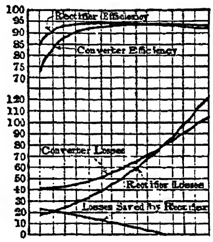


Fig. 9. Relative Efficiencies, 1000-kw. Rectifier and 1000-kw. Synchronous Converter at 600 Volts

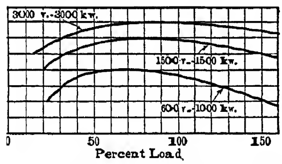


Fig. 10. Overall Station Efficiencies of Rectifiers at Various Voltages

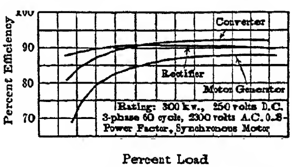


Fig. 11. Efficiency Characteristics of Motor-generator Sets, Synchronous Converters and Mercury Arc Rectifiers, with Transformer Losses

Industrial Applications usually require direct current at 250 volts, where the mercury arc rectifier is at a disadvantage. The principal loss in a rectifier is the loss in the arc. Arc drop varies with load, ranging from 20 to 35 volts, and is independent of rated D.C. output voltage. Hence, for high D.C. voltages, efficiency is high, but is relatively low for low voltages. Higher efficiencies at part loads are an advantage, and for low load factor applications, 250-volt rectifiers may be warranted. The relative advantages of a rectifier, a motor-generator set or a converter in industrial service depend on the type of load and the D.C. voltage required. Three-wire operation is not so readily obtained with a rectifier as with a motor-generator or converter. A balancer set is necessary, which increases cost and lowers efficiency. Switch-gear control equipment can be manual or automatic.

Efficiencies of mercury arc rectifiers in comparison with other types of conversion apparatus are shown in Figs. 9, 10, and 11.

Power Factor of a mercury arc rectifier is shown in Fig. 12, which is representative. Since power factor is due almost entirely to wave distortion, it does not result in appreciable increase in reactive current in the system, and is less important than in rotating apparatus.

Voltage Regulation of a mercury arc rectifier is about 5%, from 5% to 100% load. For most installations this is satisfactory.

Approximate Prices (1935) of 250-volt mercury arc rectifiers are shown in Fig. 13.

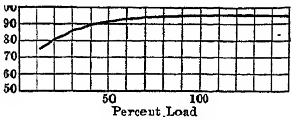


FIG. 12. Power Factor of 600-volt, 1000-kw., 12-phase Rectifier at Various Loads

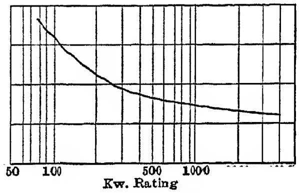


FIG. 13. Approximate 1935 Prices of Mercury Arc Rectifiers, 3-phase, 60-cycle, 2300 Volts A.C., 250 Volts D.C.

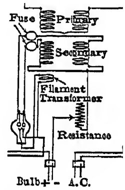


FIG. 14. Internal Wiring for Full-wave Tungar Rectifier, 30 Volts, 0.3–0.5 Amperes

TUNGAR RECTIFIERS are used for small battery-charging installations. They use a tube rectifier with transformer and means of regulating the D.C. output. Current capacities are small. A few standard equipments are: 6–90 volts, 6 and 12 amperes; 120–175 volts, 0.5–2 amperes and 2–6 amperes; 24–30 volts, 0.5–2.5 amperes; 40–60 volts, 0.5–2.5 amperes. Fig. 14 shows typical connections for a full wave tungar charger.

SWITCHBOARD EQUIPMENT

1. PANEL CONSTRUCTION AND EQUIPMENT

TYPES OF PANELS.—Two general classifications are vertical type and bench type. The vertical type is applicable to most conditions. Protective equipment may be mounted directly on panels or mounted remotely and electrically controlled, with only the control on panels. Bench boards are used largely in high-capacity A.C. stations, where switching conditions are complicated, and electrically-operated switchboard apparatus is employed.

SWITCHBOARD PANEL MATERIALS ordinarily used are steel, ebony-asbestos, and slate. Characteristics and advantages of these materials are:

Steel Panels are almost universally used for all types of switchboards, except where live equipment is mounted on the front of panels. They are formed in one piece of 1/8-in. panel steel, with angle sections integral with the panel at top, bottom and sides. The bottom angle is bolted to the channel-iron sill. See Fig. 1. Standard panel widths are 16, 20, 24, 28 and 32 in. for panel heights of 64, 76 and 90 in.

Ebony-asbestos consists of asbestos fiber and other insulating and waterproofing materials. It will withstand high heat without disintegration, is impervious to moisture and oil, is hard and of permanent form, but easily machined and worked, has dull-black finish, weighs less than slate, and can be used for any kind of switchboard service.

Slate is suitable for insulating up to and including 1200 volts A.C. or D.C. Its present use is limited to matching existing installations.

Marble, as a panel material, has almost disappeared, except where demanded by the user.

STANDARD SWITCHBOARD PANEL SIZES for ebony-asbestos and slate boards are shown in Fig. 2. For ebony-asbestos panels, the one-section panel is recommended as it provides a larger unbroken mounting surface.

Channel Iron Sills are recommended for all panels with sub-bases.

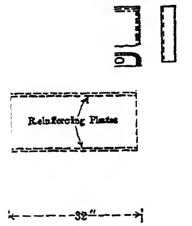
Grounding.—All switchboard panel supports should be grounded.

ALTERNATING-CURRENT SMALL-PLANT AND INDUSTRIAL SWITCHBOARDS.—Manufacturers list standardized panels, using knife switches with fuses for 60-cycle service. These control generators and feeders for general power and lighting

service in small plants. The maximum capacity of a single generator or feeder panel is 600 amperes for 240- or 480-volt, 3-phase, 3-wire non-grounded service. Construction includes a one-section ebony-asbestos panel 48 in. high, mounted on 76-in. angle-irons.

GENERATOR PANELS are of two classes: 1. Isolated Installation, to control one generator, one exciter and one or two feeder circuits. 2. Switchboard Assembly to control one or more generators and one or more feeders.

Feeder Panels have knife switches and fuses for one or more circuits, and are arranged to line up and connect to a common generator bus for switchboard assembly. Panels for isolated installation are shown in Fig. 3*a*. Each panel controls one generator, one exciter and one or two feeder circuits. Maximum capacity available in standard equipments is 250 kva. at 240 volts, or 500 kva. at 480 volts.



[F]IG. 1. Panel

Panel Devices consist of: 1 ammeter; 1 voltmeter; 3 ground detector lamp receptacles; 1 push button switch for ground detector lamps (480-volt panels); 1 double-pole, single-throw discharge switch, mounted on back of panel with operating lever in front; 1 nameplate and card holder; 1 three-pole, single-throw 250-volt or 500-volt knife switch, with fuse cutouts on front of panel, including 1 set of N.E.C.S. fuses; 1 current transformer; 1 potential transformer 440/110 volt, with fusible cutouts (for 480-volt panels), and all main connections, small wiring and cable terminals. The framework is angle-iron with floor braces. For switchboard assembly, panel devices are essentially the same except that a main bus is required, in back of the panels, to which each generator and feeder circuit is connected. A synchroscope receptacle is added to each generator panel.

NECESSARY ACCESSORIES include:

Main 3-phase bus for each panel in an assembly of two or more panels, to carry current continuously with a temperature rise not over 30° C., based on 40° C. ambient, size to be as follows:

Ampere capacity, 25-60 Cycles	Up to 750	751-1200	1201-1675	Up to 1300	1301-2050	2051-2500
Bus bars, 1/4 in. thick for each leg of bus...	1 2-in.	2 2-in.	3 2-in.	1 4-in.	2 4-in.	3 4-in.

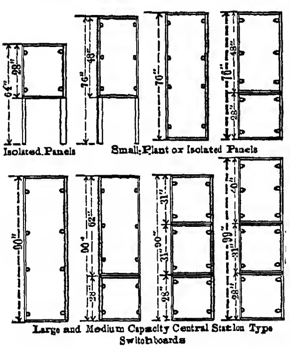


FIG. 2. Standard Sizes of Ebony-asbestos and Slate Switchboard Panels

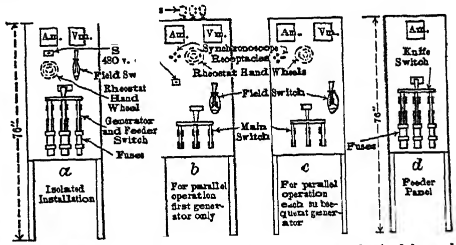


FIG. 3. A.C. Small Plant and Industrial Switchboard

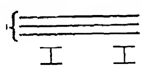


FIG. 4. Usual Order of Panel Assembly

Current that a bus must carry back of a specific switchboard panel is determined as follows: Assume the usual order of panel assembly recommended, as shown in Fig. 4. For the most economical distribution of copper, if the bus has more than one bar per leg the smaller-capacity panels of the same kind should be placed nearest the right-hand end of the board. Let I_g, I_f = respectively, current in bus back of the panels under consideration, and I_{rg} and I_{rf} = their rated ampere

Table 1.—Net Weights of Panel Materials.

Material	Lb. per sq. ft.			
	1/8 in.	1 in.	1 1/2 in.	2 in.
Steel	5.1	....	....	....
Ebony-asbestos	....	10.6	15.9	21.2
Slate	....	....	21.8	29.1
Marble	....	...	21.6	28.8

capacity; I_{g1}, I_{g2}, I_{f1}, I_{f2} = respectively, rated capacity of each generator and feeder panel to the left of the ones under consideration. Then

$$I_g = I_{rg} + I_{g1} + I_{g2} + \ldots, \text{ etc.}; \quad I_f = I_{rf} + I_{f1} + I_{f2} + \ldots, \text{ etc.}$$

If this gives a result exceeding the current which the bus is required to carry back of the extreme right-hand generator panel, use the latter value as the current the bus is required to carry back of the feeder panel. Incoming lines should be regarded as generators. If the panels are arranged in the reverse order from that shown in Fig. 4, interchange the values for right and left in the formulas.

Synchronizing Equipment is necessary if the switchboard controls two or more generators operating in parallel. This includes a swinging bracket, synchroscope and synchronizing plug.

A Governor Motor Control Switch is required for turbine- or engine-driven generators, and the prime mover speed is adjusted from the switchboard.

Field Rheostats and operating mechanisms are considered as generator and not switchboard equipment. The combined generator and exciter panels are designed for the exciter rheostat to be supported back of the panel and the generator rheostat mounted separately, operated from the front of the panel by concentric hand wheels.

Heating of switchboard equipment should not exceed limits prescribed by the A.I.E.E. Standardization Rules.

APPROXIMATE PRICES AND WEIGHTS applying to a 240-volt panel (Fig. 3*a*) are given in Table 2. Table 3 gives similar information for a 480-volt panel (Fig. 3*b*).

Table 2.—Approximate Prices and Weights of 240-volt Panels.

Generator, kva. rating	0–25	26–41	42–83	84–166	167–250
Panel ampere rating	0–60	61–100	101–200	201–400	401–600
Approximate weight, lb.	220	225	230	245	260
Approximate price (1935)	$204	211	217	247	279

Table 3.—Weights and Prices of Panels for Assembly with 480-volt Feeder Panels

Generator kva. Rating	Panel Ampere Rating	Approximate Price, Dollars (1935)	Approximate Weight, lb.	Feeder Panel (Fig. 11*c*) Approx. Price, Dollars (1935)	Feeder Panel Approximate Weight, lb.	Feeder Panel Feeder Ampere Rating
0–50	0–60	264	270	125	170	0–200
51–83	61–100	266	270	157	190	201–300
84–166	101–200	273	275	162	200	301–400
167–330	201–400	292	295	193	235	401–600
331–500	401–600	316	315			

Table 4.—Bus Spacing for Different Voltages

Operating Voltage	Distance Between: Bus Centers, in. A*	Bus Centers, in. B†	Live Side and Ground, in., min. A*	Live Side and Ground, in., min. B†	Opposite Potentials, in., min. A*	Opposite Potentials, in., min. B†
250	1 1/2	2 1/2	3/4	1 1/2	1	2
600	2	3	1	2	1 1/2	2 1/2
1,100	4	5	1 1/2	2 1/2	2 1/2	3 1/2
2,300	5	6 1/2	2	2 3/4	2 3/4	4
4,000	6	7 1/2	2 1/4	3	3	4 1/2
6,600	7	8	2 1/2	3	3 1/2	4 1/2
7,500	8	9	2 3/4	3 1/4	4	4 1/2
9,000	9	10	3	3 1/2	4 1/4	4 1/2
11,000	9	11	3 1/4	3 3/4	4 1/2	4 3/4
13,200	9	12	3 1/2	4 1/4	4 3/4	5
15,000	9	14	3 3/4	4 1/2	5	5 1/2
16,500	10	14	4 1/2	5	5 1/2	6
18,000	11	14	5	6	6	7
22,000	12	15	6	7	7 1/2	9

Operating Voltage	Distance Between: Bus Centers, in. A*	Bus Centers, in. B†	Live Side and Ground, in., min. A*	Live Side and Ground, in., min. B†	Opposite Potentials, in., min. A*	Opposite Potentials, in., min. B†
26,000	14	16	8	9	10	12
35,000	18	22	10	12	12	15
45,000	22	27	13 1/2	15	16	18
56,000	28	31	16	17 1/2	17 1/2	19
66,000	34	38	18 1/2	23	22	24
75,000	36	42	25	27 1/2	26	30
90,000	46	54	27	29	32	35
104,000	54	60	28 1/2	32	34 1/2	39
110,000	60	72	33	36	38	41
122,000	66	78	35 1/2	39	42	47
134,000	74	84	39	41	48 1/2	56
148,000	82	96	45	50	59	67
160,000	88	105	53	63	70	85

* Based on a factor of safety of 3 1/2 between live parts of opposite polarity, and of 3 between live parts and grounds. † Good practice for larger plants.

2. ALTERNATING-CURRENT SWITCHBOARDS

MANUALLY-OPERATED A.C. SWITCHBOARDS WITH OIL CIRCUIT BREAKERS control generators, transformers and feeders for general power and lighting service in public service, municipal, and industrial plants. Panels are primarily for switchboard assembly, connected to a single set of buses. Standard equipments are listed for 240, 480 and 2400 volts, 60-cycle, 3-phase, non-grounded, 3-wire circuits. Maximum capacities provided for are: Generators, 3100 kva. at 2400 volts; exciters, 125 kw. (250 kw. at

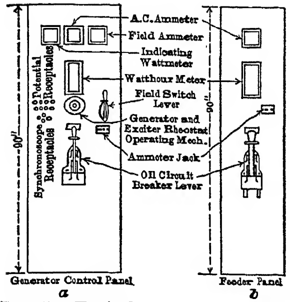


FIG. 5. Typical 90-in. A.C. Panels, Manually Operated

250 volts); feeders, 800 amperes. Standard panel material is steel, but ebony-asbestos or slate can be used.

Fig. 5a shows a typical 90-in. panel, including devices for average operating conditions, to control one generator, parallel or non-parallel operated, with individual belted or direct-connected exciter.

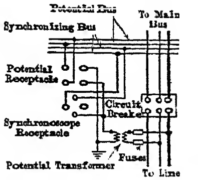


FIG. 6. Connections Showing Potential and Synchronizing Receptacles Added to Fig. 5b to Serve as Incoming Line Panel

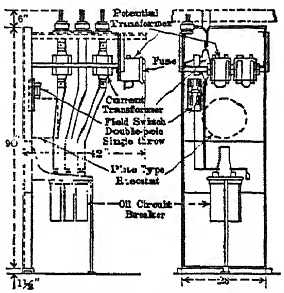
Generator Panels may include A.C. ammeter, indicating wattmeter and field ammeter, a 3-pole ammeter transfer jack and plug for reading in each phase on one meter and an 8-point potential receptacle for reading voltage across all phases.

Feeder Panels may have one ammeter and 3-pole ammeter jack and plug. A typical 90-in. feeder panel is shown in Fig. 5b; it also is suitable for an incoming line panel. In stations where generators are installed, synchronizing equipment must be added to the feeder panel to make it serve as an incoming line panel as shown in Fig. 6.

Watthour Meters, when desired, usually are included with the generator panel rather than the feeder panel, where they are most likely to operate continuously at full load for highest efficiency.

Swinging Brackets are necessary for mounting the synchroscope used in paralleling two or more generators, or where it is necessary to parallel a generator with an incoming line. It is usual to mount voltmeter plugs and receptacles on each generator and incoming panel. In synchronizing, it then is possible for the operator to see the voltage of station bus and incoming generator at the same time.

PROTECTION OF GENERATORS AND FEEDERS is provided by automatic disconnection of feeder circuits when load current exceeds a predetermined value. Improved operation often is obtained by adding time delay to such protection, so that service on a feeder is not interrupted for momentary overloads. Automatic protection is not applied to generator field or exciter circuits, as continuity of service usually is more important, and possibility of trouble in these circuits is remote.



FI . 7. Construction and Assembly of A.C. Generator and Feeder Panels

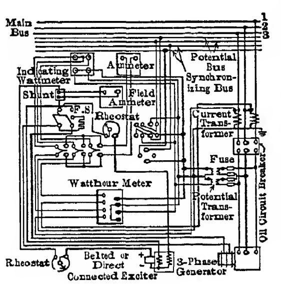


FIG. 8. Connection Diagram of A.C. Generator Panel

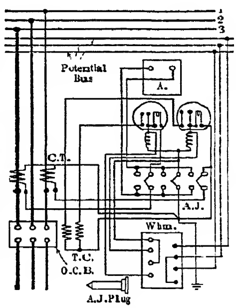


FIG. 9. Connection Diagram of A.C. Feeder Panel

A = A.C. Ammeter; A.J. = Ammeter jack; C.T. = Current transformers; D.R. = discharge resistors; F = Fuses; F.A. = Field ammeter; F.S. = Field discharge switch; I.W. = Polyphase indicating wattmeter; O.C.B. = Oil circuit breaker; P.T. = Potential transformers; Pe.Re. = Potential receptacle; Re = Relays; Rh = Field Rheostat; S.Re = Synchronizing receptacle; Sh = Ammeter shunt; T.C. = Trip coils; Whm = Polyphase integrating watt-hour meter

Disconnecting Switches provide for convenient isolation of oil circuit breakers from the bus for inspection, changing oil, and adjustments. They are recommended particularly for switchboards under 24-hr. loads that cannot be shut down at intervals.

Selection of Oil Circuit Breakers for switchboard assembly should not be made until proper ratings are determined from investigation of short circuit conditions. (See p. 15-63.)

Construction and Assembly of typical 90-in. generator and feeder switchboard is shown in Fig. 7. Figs. 8 and 9 show typical connection diagrams.

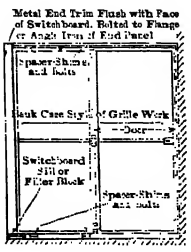


FIG. 10. Typical End Screen for Switchboards

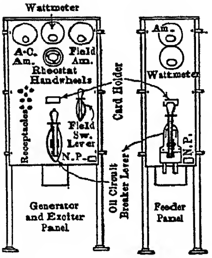


FIG. 11. Typical A.C. Panels for Small Plants

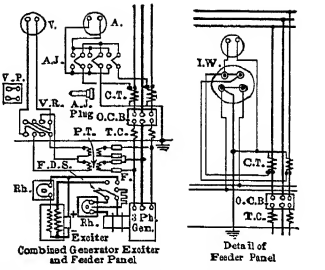


FIG. 12. Standard Connections for Small Plant A.C. Panels. See Key under Fig. 8

Grille-work End Screens provide complete enclosure for switchboards installed parallel to, and within a few feet, of a station wall. Accidental contact with live parts is prevented. Fig. 10 shows typical construction.

Ground Detector Equipments are required by the National Electrical Safety Code.

SMALL-PLANT INDUSTRIAL-TYPE A.C. SWITCHBOARDS are standardized for 240, 480, 600 and 2400 volts, 3-phase, 3-wire circuits. They use oil circuit breakers instead of fused knife switches. Manufacturers list them for 60-cycle service, but they are available for all standard frequencies. Listed panels are limited generally to generators, not exceeding 332 kva. at 240 volts, 665 kva. at 480 volts, and 750 kva. at 600 volts.

Typical panels are shown in Fig. 11, and typical connections in Fig. 12. Construction is shown in Fig. 13, *a* and *b*.

3. DIRECT-CURRENT SWITCHBOARDS

D.C. SMALL-PLANT AND INDUSTRIAL MANUALLY-OPERATED SWITCHBOARDS, limited to generators of 150 kw. at 125 volts, or 300 kw. at 250 volts and 1200 amperes maximum feeder capacity, are shown in Fig. 14. Feeder circuits may be included on the generator panel or on separate panels. The panels provide for isolated or parallel operation, and when furnished with proper bus connections can be assembled for control of two or more generators and feeders. Knife switches generally are used for lighting, and circuit breakers for power service. Circuit breakers are required where feeder circuits may be subjected to abnormal conditions of dangerous character, and it may be necessary to open the feeder circuits to relieve the connected apparatus.

Usual panel dimensions are: Length 36 and 48 in.;

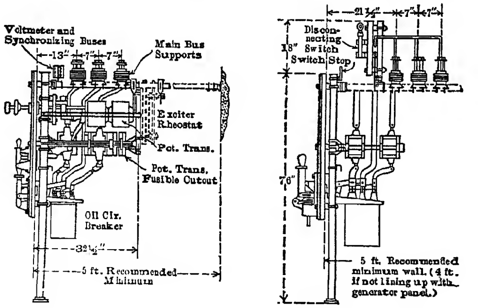


FIG. 13. Construction of Small Plant Generator and Feeder Panels

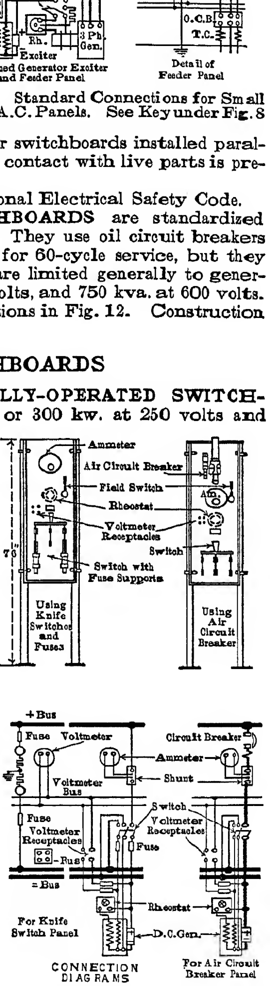


FIG. 14. Typical Small Plant D.C. Generator Panel and Connection Diagrams

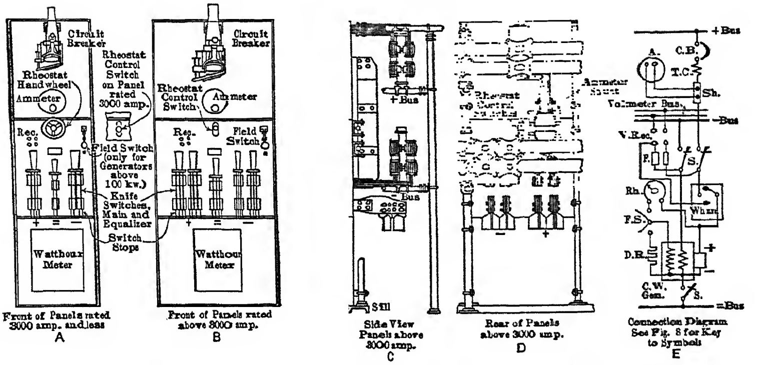


FIG. 15. D.C. Generator Switchboard for 2-wire Service and Parallel Operation

width 20, 24, 28, 32 in. These panels are mounted on 64- and 76-in. angle-iron supports as required. For small isolated generators, panels can be wall mounted.

D.C. SWITCHBOARDS FOR GENERAL POWER AND LIGHTING SERVICE.—Standard individual panels designed for switchboard assembly, provide control for generators and circuits as follows: 2-wire generators, 25 to 1500 kw. (on both 3-wire and 2-wire systems); 2-wire, shunt- or compound-wound synchronous converters; 3-wire generators 50 to 1500 kw.; 3-wire balancer sets, 14 to 1000 amperes neutral current; 2- and 3-wire feeder circuits, 30 to 6000 amperes, and for 125- and 250-volt, 2-wire non-grounded, or 125/250-volt, 3-wire systems.

Typical D.C. Panel Construction for 2-wire Generators for assembly in 2-wire non-grounded switchboards is shown in Fig. 15. Each panel controls one compound-wound D.C. generator operating in parallel with others. Three-wire generator panels are shown in Fig. 16.

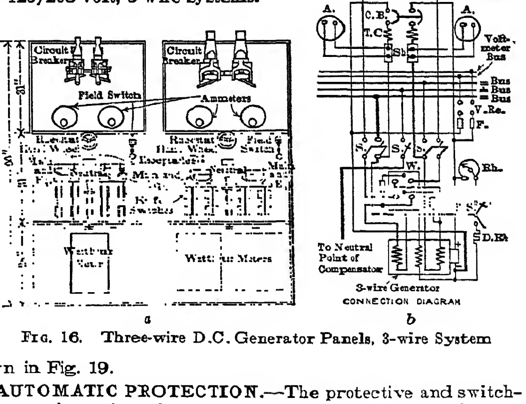


FIG. 16. Three-wire D.C. Generator Panels, 3-wire System

Complete D.C. Switchboard Connections.—Fig. 17 is typical for a 2-wire system. A 3-wire system for 3-wire generators is shown in Fig. 18. Typical connections for a 3-wire system using balancer sets are shown in Fig. 19.

D.C. SWITCHING AND AUTOMATIC PROTECTION.—The protective and switching schemes upon which the control panels and connections are based, in general, are:

Two-wire Systems.—Over-current protection for 2-wire generators and converters is obtained by single-pole, single-coil circuit breakers connected in the positive lead. The series fields of 2-wire,

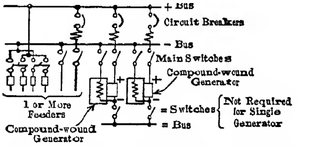


FIG. 17. Connections for D.C. Switchboard with One or More Compound-wound Generators, 2-wire System

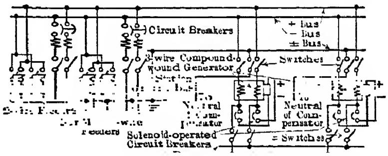


FIG. 18. Connections for D.C. 3-wire System with 3-wire Generators

compound-wound, generators and converters are equalized on the negative sides and the breaker trip coil is actuated by full armature current for both compound-wound and shunt-wound machines. This is important when compound-wound generators operate in parallel, because otherwise unnecessary disconnection from the bus, or injurious overloading may occur, depending on the direction of equalizing currents. In synchronous converters, additional protection is afforded in the control of the A.C. end.

Speed limit devices on converters or motor-generator sets also actuate circuit breakers through under-voltage devices, to prevent parallel-operated machines from motoring from the D.C. bus at dangerous speeds, when the converter or motor supply is accidentally disconnected. Speed limit devices usually are set at about 15% over-

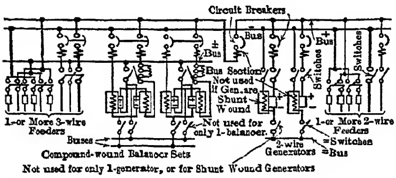


Fig. 19. Connections for D.C. 3-wire System with 2-wire Generators and Balancer Sets, Non-grounded System

When paralleling compound-wound generators or converters, less voltage disturbance occurs if series fields are established before connecting the machines to the bus. The connection diagrams show knife switches arranged to permit this procedure.

Three-wire Generators on 3-wire Systems usually are compound-wound, and operated in parallel. Switchboard panels are standardized on this basis. Generators have divided series fields, connected on opposite ends of the armature. This is necessary for equalization. Circuit breakers are so connected between generator brushes and equalizer switches that they are actuated only by armature current. Knife switches are so arranged that series fields may be established in the proper direction and the generators brought to full voltage before final connection to main bus. Incidentally, this affords a means of correcting the polarity of a generator. With other switching arrangements voltage disturbances may occur at paralleling, with possibility of heavy armature short circuits. Automatic circuit breaker elements are mechanically or electrically interlocked, so that when one side opens the other also must open.

Two-wire Generators with Balancer Sets and Bus Sectionalizing on 3-wire Systems.—On 3-wire systems, when the neutral is derived from a balancer set, unbalancing may exceed the capacity of the balancer set and shift the neutral enough to burn out lamps on the lightly loaded side. A balanced voltage differential relay, which functions when unbalanced voltage exceeds a predetermined limit affords protection against this contingency.

Control is provided on panels whereby the relay trips the breakers on all 2-wire generators, shutting the system down completely, or a bus section breaker can be used on a separate panel (see Fig. 19) which disconnects the balancers and 3-wire load, leaving the generators and 2-wire load in operation. Switches on 2-wire generator panels on 3-wire systems provide the same operating sequence as on 2-wire systems.

Balancer panels have double-pole, double-coil breakers, a starting and positive switch, a neutral switch, and for parallel-operated balancers, a rheostat switch and two equalizer switches. To start a balancer set which operates singly, the neutral switch must be open; if parallel-operated, the rheostat switch also must be open when starting.

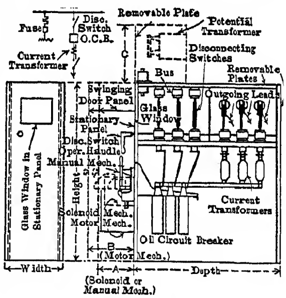


Fig. 20. Single-bus Cubicle with Group-operated Switches for Isolating Oil Circuit Breaker. See Table 4

Feeder Circuits on 3-wire Systems.—For 2-wire and 3-wire feeders on grounded neutral systems, double-pole, double-coil circuit breakers are used with single-pole knife switch in series with each breaker pole. If the neutral is not grounded, the equipment for 3-wire feeders is the same as if the neutral is grounded except, to comply further with the Code, a triple-pole switch is used, to make sure that outside legs are open when the neutral is opened.

D.C. Circuit Breakers are equipped to operate instantaneously on over-current. Immediate interruption of abnormal current is necessary.

4. CUBICLE SWITCHGEAR

INDOOR A.C. CUBICLE SWITCHGEAR consists of steel-enclosed switchboard units, in which all apparatus for one panel, including potential transformers, current transformers, oil circuit breaker, disconnecting switches, meters and instruments, and mounting structures are compactly assembled in one cubicle. Typical construction is shown in Fig. 20 with various devices assembled. The hinged door in the front forms the meter panel. All switching operations can be made without opening the cubicle.

Indoor cubicles are used in central station, substations, industrial plants and office buildings. They control generators, motors, incoming lines, feeders, bus-sectionalizing and bus-tie circuits. Either manually- or electrically-operated circuit breakers can be used.

Advantages of Cubicle Construction are: They are completely assembled when shipped; are designed for switchboard assembly; to put them in operation it is necessary only to set each unit in place, and connect the leads; breakers can be isolated by disconnecting switches; all live parts are enclosed; high-voltage compartment is completely isolated from low-voltage compartment; they economize space; installation cost is low.

Standard Ratings are available with breakers having current ratings from 200 to 2000 amperes and voltage ratings up to 15,000 volts. The interrupting ratings range from 20,000 to 500,000 kva. All ampere ratings are based on the maximum current which

Table 5.—Dimensions of Standard A.C. Switchgear Cubicle

Interrupting kva.	Oil Circuit Breaker		Dimensions, in. (See Fig. 20)						Approximate Net Weight, lb.	
	Volts	Amperes	A	B	C †	Wide	Deep	High	Base Cubicle *	Front Enclosure
20,000	5,000	200	15	19	24	24	54	76	1150	200
	5,000	400	15	19	24	24	54	76	1150	200
25,000	7,500	600	15	19	24	24	54	76	1300	200
	2,500	800	15	19	24	24	54	76	1300	200
50,000	15,000	600	15	19	30	32	64	90	2000	250
	7,500	1200	15	19	24	32	64	90	2150	250
	5,000	2000	15	19	24	40	64	102	3000	300
50,000	15,000	600	15	19	30	32	64	90	1850	250
	7,500	1200	15	19	24	32	64	90	1950	250
	5,000	2000	15	19	24	40	64	102	3100	300
100,000	15,000	600	15	19	30	32	64	90	2300	250
	15,000	1200	15	19	30	32	64	90	2450	250
	7,500	2000	15	19	24	40	64	102	3150	300
150,000	15,000	600	15	19	30	32	64	90	2350	250
	15,000	1200	15	19	30	32	64	90	2500	250
250,000	15,000	600	15	19	30	32	64	90	2350	250
	15,000	1200	15	19	30	32	64	90	2500	250
	15,000	2000	19	23	30	40	64	102	4300	300
500,000	15,000	600	19	23	30	40	64	102	3650	300
	15,000	1200	19	23	30	40	64	102	3850	300
	15,000	2000	19	23	30	40	64	102	4350	300

*Net weight of cubicle with solenoid-operated breaker, connections, buses, and supports.
† Potential transformers rated 2000–3000 volts and below are mounted in cubicle structure.

Table 6.—Dimensions of Outdoor Switch Houses

Interrupting kva.	Oil Circuit Breakers		Switch House Dimensions, in. (See Fig. 21)								
			No Disconnecting Switches			With One Set of Indoor Group-operated Disconnecting Switches			With Superstructure for 3, 6, or 9 Hook-operated Outdoor Disconnecting Switches		
	Volts	Amperes	High	Wide	Deep	High	Wide	Deep	High	Wide	Deep
20,000	5,000	200									
	5,000	400	76	36	48	88	36	48	...	..	..
25,000	7,500	600									
	2,500	800									
25,000	7,500	600									
50,000	15,000	600									
	7,500	1200									
	15,000	600									
	7,500	1200									
100,000	15,000	600	84	48	52	96	48	58	142	48	52
	15,000	1200									
150,000	15,000	600									
	15,000	1200									
250,000	15,000	600									
	15,000	1200									
50,000	5,000	2000									
	5,000	2000									
100,000	7,500	2000									
250,000	15,000	2000	96	60	66	108	60	66	154	60	66
500,000	15,000	600									
	15,000	1200									
	15,000	2000									

the cubicles will carry continuously without overheating. Equipment should be selected which has a rating at least equal to the maximum rating of the circuit.

Weights and Dimensions applying to the type of cubicle shown in Fig. 20 are given in Table 5.

OUTDOOR SWITCH HOUSES are self-contained switching and metering equipments, enclosed in weather-proof housings. They are used as temporary or permanent feeder or line protective installations for manual or automatic operation. Any type of standard metering, relay or control equipment can be used.

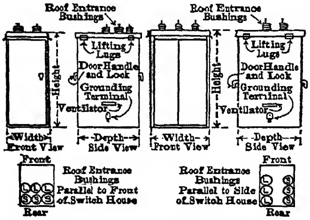


FIG. 21. Outdoor Switch Houses. See Table 5

To install an outdoor switch house it is only necessary to anchor the house, connect the ground terminal and the incoming and outgoing leads to the entrance bushings. A concrete pad, or corner pieces, are sufficient for mounting. If system changes require relocation of the switch house, it is easily disconnected and moved as a unit.

Typical construction is shown in Fig. 21. Dimensions of standard units are given in Table 6.

BATTERIES

A battery comprises two or more cells which convert chemical energy to electrical energy. In each cell two unlike electrodes are immersed in an electrolyte. The electrolyte is a solution of water and acid, alkalies or salts.

Primary Cells generate current by consuming one element, usually zinc. The electrodes may be copper or carbon and zinc. Such cells become exhausted in use and have to be replaced, because the chemical reactions are not reversible.

Secondary or Storage Cells convert chemical energy to electrical energy by chemical reactions which are reversible. Such cells can be recharged by passing direct current through in the opposite direction.

1. TYPES OF STORAGE BATTERIES

Storage Batteries comprise one or more storage cells, of which there are two major types:

THE LEAD-ACID CELL.—The positive plate contains peroxide of lead; the negative plate contains finely-divided pure lead. The electrolyte is sulphuric acid and pure water, with a specific gravity ranging from 1.200 to 1.280 according to type of cell. In the lead battery, specific gravity falls during discharge and rises during charge, thus indicating the condition of discharge of the cell.

THE NICKEL-ALKALI CELL (Edison cell) uses a positive plate containing an oxide of nickel and a negative plate containing metallic iron. The alkaline electrolyte does not change in specific gravity during charge or discharge. The container is a nickel-plated steel can, and individual cells must be insulated

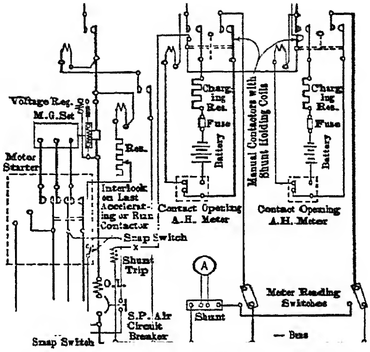
2. METHODS OF CHARGING

To fully charge a storage battery, current must be passed through the cells in a direction opposite to that of discharge. The ampere-hours of current required for charge must equal the discharge plus an excess of 5 to 20% to make up losses. In general, any charging rate is permissible which does not produce excessive gassing or a cell temperature exceeding 110° F.

THE VARIABLE-RESISTANCE METHOD (series-resistance or constant-current method) uses an adjustable, charging rheostat in series with each battery to give manual control of the charging rate. For a lead battery the starting rate is maintained until battery voltage rises to about 2.5 volts per cell. The rate then is reduced to a finishing rate, of about 40% of the starting rate, until battery voltage rises to 2.5 volts per cell. With an Edison battery, this method results in a constant current instead of a starting and finishing rate; the battery volts per cell will, however, be different. This method of

charging is applicable to, and almost necessary for charging different batteries of a different number of cells from the same D.C. supply.

THE MODIFIED-CONSTANT-POTENTIAL METHOD maintains a constant bus voltage of about 2.6 volts per cell for lead batteries, and between 1.85 and 2.0 volts per cell for Edison batteries, with a fixed resistance in series with each battery to be charged. For lead batteries, each battery must have the same number of cells but with Edison batteries it is possible to adjust each fixed resistor to a proper value for the particular battery to be charged, irrespective of the number of cells. The fixed resistance usually is provided with several taps for seasonal adjustment. More resistance will be required when batteries are warm, because required charging voltage is lower, than when they are cold.



. 1. Connections for Modified Constant-potential Charging of Two Batteries

The modified-constant-potential method of charging, if conditions warrant it, is considered the best method. It automatically regulates current to battery requirements, and makes it unnecessary for the operator to determine current setting for each circuit. This method is used when fully automatic equipment is required for two or more charging circuits. Fig. 1 shows typical connections.

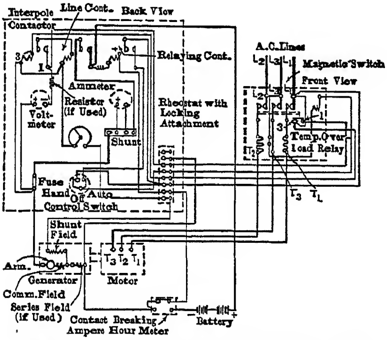
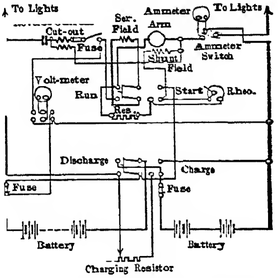


Fig. 2. Connections of Automatic Taper Charging Equipment for Vehicle Batteries, Showing Starting Switch

TAPER CHARGING EQUIPMENT for single batteries usually is automatic, using a motor-generator set designed to give the modified-constant potential characteristics, but without fixed resistance in the circuit between battery and generator. Fig. 2 shows connections for such equipment. A typical equipment includes a motor-generator set and a control panel: a single pole contactor, closing on battery voltage; shunt-trip relay, hand-reset, operated by ampere-hour meter preferably located at the battery; ammeter; voltmeter; generator field rheostat; fuse for over-current protection; snap switch for motor starting.

A battery often is used as an emergency source of power. Under normal operating conditions it floats on the line and is kept charged by the generator. After long periods of discharge it is necessary to recharge the battery in a short time. This can be accomplished manually with connections as shown in Fig. 3, where the battery is split into two sections. When floating, or on discharge, the sections are connected in series. When on charge they are connected in parallel and charged through a series resistance.



IG. 3. Arrangement for Manually Charging Emergency Battery from Engine-driven Generator, also Used as Starting Motor for Engine

BATTERY VOLTAGES.—A Lead Battery starts discharging at about 2.05 volts per cell and finishes at about 1.75 volts. It starts charging at about 2.1 volts per cell and finishes at 2.5 volts per cell. Normal charging time is 8 hr.

Edison Batteries on a 5-hr. complete discharge start discharging at about 1.45 volts per cell, and finish at 1.1 volts. On charge, the starting voltage is about 1.55 volts per cell, finishing at 1.80 volts per cell.

Average Current Requirements for charging typical batteries of both types are given in Table 1. These charging current rates apply to batteries for industrial trucks and tractors, street trucks, and electric locomotives.

Table 1.—Charging Rates per Cell

Lead (Ironclad)					Edison			
Type	Variable Resistance Method		Modified Constant Potential Method		Type	Variable Resistance Method	Modified Constant Potential Method	
	Start, amp.	Finish, amp.	Start, amp.	Finish, amp.		Normal Rate, amp.	Start, amp.	Finish, amp.
7 MV	19	8	22	8	A–4, G–4	30.0	38	25
9 MV	24	10	29	10	A–5	37.5	47	31
11 MV	30	12	36	12	A–6, G–6	45.0	57	37
13 MV	35	14	43	14	A–7, G–7	52.5	66	43
15 MV	40	16	50	16	A–8	60.0	75	49
17 MV	45	18	57	18	G–9	67.5	85	55
19 MV	51	20	65	20	A–10	75.0	94	62
21 MV	56	22	72	22	G–11	82.5	103	68
23 MV	61	24	79	24	A–12	90.0	113	74
25 MV	66	26	86	26				

3. CHARGING SETS

BATTERY CHARGING MOTOR-GENERATOR SETS (Variable resistance method). —Standard equipment is available, designed particularly for multiple charging of two or more electric industrial trucks, street trucks and electric locomotives. To select the proper set the following information is necessary: Voltage, frequency and phases of power supply, number and sizes of batteries, lead or Edison, number of cells in each. number of plates per cell and maker of lead batteries, number of batteries to be charged at one time.

Generator Voltage must be higher than that of the highest-voltage battery. The highest charging voltage of lead cells is 2.60 volts; of Edison cells, 1.85 volts. With long lines from control board to batteries, allow for line drop. To determine voltage of generator, multiply number of cells by volts per cell. For example, a 44-cell lead battery requires $44 \times 2.60 = 114.4$ volts; a 60-cell Edison battery, $60 \times 1.85 = 111.0$ volts. A 115-volt generator would be used in either case.

Current Rating of Generator.—Add starting rates of all batteries to be charged at one time. See Table 1 for charging rates for 8-hr. charging of lead, 7-hr. charging of Type A and 4 3/4-hr. charging of Type G Edison batteries.

Kilowatt Capacity.—After determining voltage and current rating of generator, amperes $\times$ volts/1000 = kw. capacity of motor-generator set.

MODIFIED CONSTANT-POTENTIAL CHARGING requires the same voltages per battery cell, but generator must have close voltage regulation under all operating conditions of load and temperature. Regulation must not exceed $\pm 30\%$. A compound-wound generator with voltage regulator is recommended.

Current Rating of Generator.—Add starting rates of all batteries to be charged (see Table 2), and multiply sum by 0.8.

Kilowatt Capacity.—Amperes $\times$ volts/1000 = kw. generator capacity required.

Automatic Battery-charging Motor-generator Sets and Control Panels provide the following automatic features: Independent cut-off for each battery when fully charged; shut-down of motor generator set when last battery has cut off; shut down of motor-generator set in case of line interruption; restart of the set upon return of power supply after interruption; control of charging rate by modified-constant-potential method.

Standard kw. Ratings of Charging Motor-generator Sets are as follows:

32 and 40 volts—3, 5, 7.5, 10, 15, 20, 25, 30.

48 and 63 volts—3, 5, 7.5, 10, 15, 20, 25, 30, 40, 50.

84 volts—3, 5, 7.5, 10, 15, 20, 25, 30, 40, 50, 60.

115 volts—3, 5, 7.5, 10, 15, 20, 25, 30, 40, 50, 60, 75.

Standard Motor Voltages are 110, 220, 440, 550, and 2200 volts

Generator Voltages for Lead Batteries should not be higher than the following:

Hour charge	5	6	7	8	9	10	11	12
Volts per cell, Type MV or MVA	2.53	2.56	2.59	2.63	2.69	2.745	2.835	2.97
Volts per cell, Type TL	....	2.494	2.512	2.55	2.552	2.574	2.607	2.643

Edison batteries normally are charged in 8 hours with generator voltage based on 2 volts per cell.

Motor-generator Prices and Efficiencies will be approximately the same as for standard sets under Conversion Equipment. See p. 15-37.

General.—In fixing the time required for complete battery charge all the time available should be used, since it will result in lower charging rates, smaller capacity of charging apparatus and circuits, and lower battery temperatures. For vehicle or other cyclically-charged batteries, automatic charging is recommended.

Trickle Charging Rates are those sufficient to keep an idle battery in condition.

A Floating Charge is sufficient to bring a battery back to full charge after a partial discharge. Rates are determined as required by operating conditions. Typical installations of floating batteries are emergency lighting, and control power for circuit breaker operation. These applications consist of a steady load with occasional peaks, or occasional periods when the battery is the only source of supply. Generator capacity is determined by the normal steady load plus that necessary to restore the battery to full charge in a specified time after a period of discharge. During normal operation, the battery floats on the line at line voltage. To charge the battery, generator voltage is raised to give the required charging current.

Portable Charging Equipments are available for charging batteries used with air conditioning apparatus on railway cars. Automatic or manual equipment can be provided.

POWER FACTOR

1. POWER FACTOR EFFECTS

CURRENT AND E.M.F. RELATIONS IN A.C. CIRCUITS.—In an alternating-current circuit, maximum values of current and voltage will occur simultaneously if the circuit contains only ohmic resistance. Current and voltage then are *in phase*. If the circuit contains inductive reactance, as will be produced by induction motors, electric welders, etc., voltage will attain its maximum value in advance of the maximum value of the current; that is, the current *lags* the voltage. If the circuit contains capacitive reactance, as produced by a condenser, the maximum value of the voltage will occur after the maximum value of the current; that is, the current *leads* the voltage. These relations are shown in Fig. 1*a*.

The current flowing in the circuit will be the resultant of the power component, in phase with the voltage, and of the reactive component, lagging or leading the voltage 90 deg. The power in the circuit at any instant is the product of the simultaneous values of current and electromotive force at that instant. It is the product (volts × amperes) only when the circuit contains ohmic resistance alone, or when inductive and capacitive reactance exactly balance each other. For all other cases the power in the circuit equals the product (volts × amperes × power factor).

POWER FACTOR is the proportion of the energy in an A.C. circuit, represented by the product (volts × amperes) that can be applied to useful work. Power factor can be represented graphically by a vector diagram. See Fig. 1. Diagram *b* shows a circuit containing a resistance R, an inductive reactance X_l, and with an electromotive force E_0 impressed on it. A capacitive reactance X_c may be added to the circuit. If the inductance of reactance X_l be L henrys and the frequency of the circuit be f, then $X_l = 2\pi fL$. The reactive component of the current will be $I_{xl} = E/X_l$, lagging the voltage 90 deg. The power component, in phase with the voltage, will be $I_r = E/R$. E and R are taken in volts and ohms respectively. In vector diagram *c*, let vector OE represent electromotive force E_0, and OI_r the power component of the current I_r. Let vector OI_{xl} at 90 deg. from OE represent the reactive component of the current. Then, completing the parallelogram, OI_0 is the resultant of I_r and I_{xl}, and represents the actual current I_0 in the circuit as shown by an ammeter. Current I_0 lags voltage E_0 by angle θ. Power factor, $\cos\theta = I_r/I_0$. The power component of the current I_0 is kw $= I_0 \cos\theta$,

and the reactive component is $I_0 \sin\theta$. Heating in the circuit is produced by current I_0, but useful energy is produced only by the power component I_r.

If the inductive reactance X_l, diagram *b*, be replaced by an equivalent dielectric resistance consisting of a condenser X_c, the reactive component of the current will be $I_{xc} = 1/(2\pi fC)$, and will lead the voltage. C is the capacity in farads. In the vector diagram I_r is laid out as before, and OI_{xc} represents the reactive component. The total current is I_{01}, represented by the vector OI_{01}, leading the voltage by angle θ_1. Power factor is $\cos\theta_1 = I_r/I_{01}$.

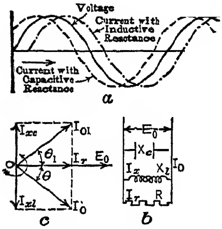


FIG. 1. Power Factor Diagrams

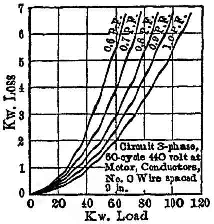


FIG. 2. Line Loss per 1000 Ft. at Various Power Factors

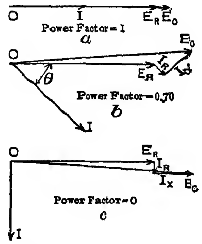


FIG. 3. Voltage Drop with Varying Power Factor

If X_c and X_l are equal and are both connected in the circuit, they will neutralize each other, and $I_{xl} + I_{xc} = 0$. I_r, I_o, and I_{01} will coincide in the vector diagram *c*, I_r will be the power in the circuit, and kva. = kw.

With a given power factor, the amount of copper required in a circuit is the same whether power factor lags or leads. A capacitive load will cause voltage to rise; inductive load will cause it to drop. No gain results in changing reactive kva. from lagging to leading at the same power factor. At unity power factor, kw. = kva.; at other power factors, the current for a given kilowatt load varies inversely as the power factor.

EFFECT OF LOW POWER FACTOR.—The kva. rating specifies the current that safely can be carried by motors, transformers, and other apparatus connected to the circuit, regardless of power factor. A 1000-kva. load at 0.5 power factor will deliver 500 kw., but will produce the same heating effect in generator, transformers and line as 1000 kva. at 0.8 power factor, delivering 800 kw. For the same kilowatt load, the investment in electrical apparatus varies inversely as power factor.

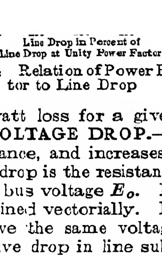


FIG. 4 Relation of Power Factor to Line Drop

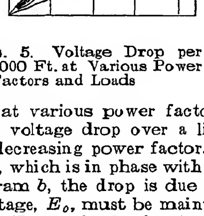


FIG. 5. Voltage Drop per 1000 Ft. at Various Power Factors and Loads

LINE LOSS.—Power is lost as heat in a conductor carrying current, and equals I^2R, where I is current in amperes, and R is resistance in ohms. For constant kilowatt load, power loss in any circuit varies inversely as (power factor)^2. Fig. 2 shows actual kilowatt loss for a given line at various power factors.

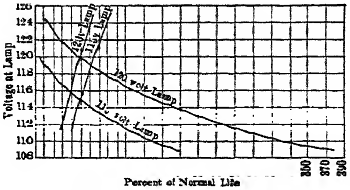
VOLTAGE DROP.—Actual voltage drop over a line depends on its resistance and reactance, and increases with decreasing power factor. See Fig. 3. In diagram *a*, the only drop is the resistance drop, which is in phase with voltage and current and subtracts from bus voltage E_o. In diagram *b*, the drop is due to resistance and reactance drops combined vectorially. Bus voltage, E_o, must be maintained at a value higher than in *a* to give the same voltage drop E_R at the load. At zero power factor, in diagram *c*, reactive drop in line subtracts almost directly from bus voltage. Fig. 4 shows relation between power factor and line drop in percent of line drop at unity power factor for constant kilowatt load. Fig. 5 shows effect of reduced power factor.

EFFECT OF POOR VOLTAGE REGULATION ON LOAD APPARATUS.—**Induction Motors.**—Voltage drop below normal increases motor slip. Maximum torque varies as (voltage)^2, *i.e.* motor loses torque rapidly with dropping voltage. A drop much below 10% may cause a motor to fail to develop the torque required by its load. If a

motor develops 200% maximum torque on full rated voltage, it will develop barely 100% torque at 70% voltage, speed being reduced to approximately 85% of normal. Loaded motors take increased current at reduced voltage. This causes increased heating, thereby limiting capacity to carry peak loads because of excessive heating and reduced torque.

Synchronous Motors do not drop in speed on reduced voltage, but pull-out torque varies directly with voltage.

Lamps vary in life and efficiency with voltage. See Fig. 6. Over-voltage produces greater light intensity, but reduces life. Under-voltage prolongs life at a sacrifice in efficiency, and less light is available for a given investment in equipment and distribution circuits.



Effect of Voltage Variation on Lamps

Heating Devices.—Proper temperature control of electrical heating equipment is not possible with variable voltage. Improving plant power factor will improve voltage regulation on plant distribution circuits. Maintenance of voltage on circuits of low power factor means greater investment.

2. POWER FACTOR IMPROVEMENT

ADVISABILITY OF IMPROVING POWER FACTOR.—Supply of reactive current, incident to low power factor loads, is considered by public utilities an additional service which justifies an additional charge. If power is generated by the industrial plant, additional investment in equipment would be necessary for the same quality of service.

High power factor has corresponding advantages. The benefits of high power factor must be balanced against the cost of corrective equipment for low power factor. A power factor survey of an individual plant will determine the advisability of power factor correction, and will balance the cost of corrective equipment against lower public utility charges, smaller investment in the plant, and improved operating conditions. In a new industrial plant, initial economies can be obtained by planning for high power factor.

In existing plants operating conditions may become critical if the load grows to the limit of generator and distribution system capacity. For instance, the installation of a 300-kva. and a 500-kva. synchronous condenser at different locations in the same plant may make possible an additional load of 500 Hp. without changes in existing equipment. Otherwise, a new generator, a new line and two additional transformer banks may be necessary.

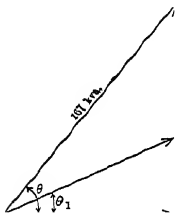


Fig. 7. Correction of a 100-kw., 0.6 Power Factor Load to 0.9 Power Factor

SOLUTION OF POWER FACTOR PROBLEM.—To improve power factor, it is necessary to reduce the ratio of reactive kva. to kw. In the vector diagram, Fig. 7, *OA* represents a load of 100 kw. at 0.6 power factor. The kva. load = (100/0.6) = 167, or $(\sqrt{3} \times E \times I)/1000$ if 3-phase, where E = line-to-line voltage and I = current, amperes, in each line. Since $\cos\theta$ = (100/167) = 0.6, represents power factor, reactive kva. =

$$167 \times \sin\theta = 167 \times \sqrt{(1 - \cos^2\theta)} = 167 \times \sqrt{1 - (0.6)^2} = 133 = AB, \text{ Fig. 7.}$$

For a power factor of 0.9 with the load at 100 kw., kva. = (100/0.9) = 111 kva. In the vector diagram, the reactive kva., *OC*, will be drawn at the angle θ_1, cosine of which is 0.9. The reactive kva. will be *AC* which scales to 48.4, and which is mathematically equal to $111 \times \sqrt{1 - (0.9)^2}$ = 48.4. Capacity of corrective equipment required will be (133 − 48.4) = 84.6 kva. leading. That is, 84.6 kva. must be neutralized by connecting to the line apparatus which will draw leading current. Lagging kva. in Fig. 7 is plotted from *A* to *B*, and leading kva. from *B* to *A*.

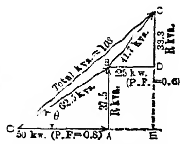


Fig. 8. Correction of Power Factor of Combined Loads

Fig. 8 shows a combined vector diagram for obtaining overall power factor and total kva. A 50-kw. load at 0.8 power factor on one feeder circuit, and a 25-kw. load at 0.6 power factor lagging on another feeder are assumed. The line *OA* drawn to scale represents 50 kw. At 0.8 power factor, kva. = (50/0.8) = 62.5. *OB* is drawn equal to 62.5 at the angle $\cos^{-1}\theta = 0.8$. Then AB = 37.5 = the reactive kva. for the 50-kw. load. From point *B*, *BD* is drawn to scale to represent 25 kw., and following the same procedure, the reactive kva. *DC* is found to be 33.3 kva. The total kva. is found to be 103 at a power factor of 0.73 and the combined

reactive kva. is 70.8 with a total load of 75 kw. The capacity of corrective apparatus to improve power factor to any other value is determined by the same procedure as outlined for Fig. 7.

POWER FACTOR IMPROVEMENT TABLE.—In Table 1, the tabulated figures multiplied by the kw. input will give the leading kva. necessary to correct from the original to the desired power factor.

Table 1.—Power Factor Corrections

Original Power Factor, percent	Desired Power Factor, percent					Original Power Factor, percent	Desired Power Factor, percent				
	100	95	90	85	80		100	95	90	85	80
	Kva. input						Kva. input				
50	1.732	1.403	1.248	1.112	0 982	75	0.882	0.553	0.398	0.262	0.132
51	1.687	1.358	1.202	1.067	.936	76	.855	.527	.371	.235	.105
52	1.643	1.314	1.158	1.023	.892	77	.829	.500	.344	.209	.078
53	1.600	1.271	1.116	0.980	.850	78	.802	.474	.318	.182	.052
54	1.559	1.230	1.074	.939	.808	79	.776	.447	.292	.156	.026
55	1.518	1.189	1.034	.898	.768	80	.750	.421	.266	.130	......
56	1.479	1.150	0.995	.859	.729	81	.724	.395	.240	.104	......
57	1.442	1.113	.957	.822	.691	82	.698	.369	.214	.078	......
58	1.405	1.076	.920	.785	.654	83	.672	.343	.188	.052	......
59	1.368	1.040	.884	.748	.618	84	.646	.317	.162	.026	......
60	1.333	1.004	.849	.713	.583	85	.620	.291	.136	......	......
61	1.299	0.970	.815	.679	.549	86	.593	.265	.109	......	......
62	1.266	.937	.781	.646	.515	87	.567	.238	.082	......	......
63	1.233	.904	.748	.613	.482	88	.540	.211	.056	......	......
64	1.201	.872	.716	.581	.450	89	.512	.183	.028	......	......
65	1.169	.840	.685	.549	.419	90	.484	.155	......	......	......
66	1.138	.810	.654	.518	.388	91	.456	.127	......	......	......
67	1.108	.779	.624	.488	.358	92	.426	.097	......	......	......
68	1.078	.750	.594	.458	.328	93	.395	.066	......	......	......
69	1.049	.720	.565	.429	.298	94	.363	.034	......	......	......
70	1.020	.691	.536	.400	.270	95	.329	......	......	......	......
71	0.992	.663	.507	.372	.241	96	.292	......	......	......	......
72	.964	.635	.480	.344	.214	97	.251	......	......	......	......
73	.936	.608	.452	.316	.186	98	.203	......	......	......	......
74	.909	.580	.425	.289	.158	99	.142	......	......	......	......

1.—Wattmeter indicates a plant load of 100 kw. at 0.7 power factor. Required the leading reactive kva. necessary to correct power factor to 0.95. In Table 1, line 70, column 95, find factor 0.691. Reactive kva. = 100 × 0.691 = 69.1 kva. If capacitors are used, the nearest standard unit will be selected.

Example 2.—Assume the power factor correction required in Example 1 to be made by substituting synchronous motors of 0.8 power factor for induction motors. For each kw. of induction motors operating at 0.7 power factor, Table 1 shows a 1.020 lagging reactive kva. For each kw. input to 0.8 power factor synchronous motors, the table shows leading kva. of 0.75. If the induction motors are replaced by synchronous motors, each kilowatt in synchronous motors reduces lagging reactive kva. by (1.020 + 0.75) = 1.77 kva. Total reduction to correct power factor to 0.95 is 6.91 kva. (Example 1). Required capacity of synchronous motors is 69.1/1.77 = 39 kw. Nearest standard synchronous motor would be 50 Hp.

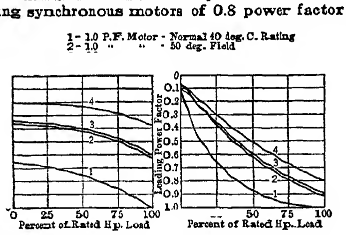


Fig. 9. Corrective Kva. and Power Factor of General-purpose Synchronous Motor with Normal Full-load Excitation

CORRECTIVE KVA. AVAILABLE FROM SYNCHRONOUS MOTORS.—The approximate amount of kva. available from, and the power factor of, general purpose synchronous motors at any load, with full load excitation, is shown in Fig. 9.

METHODS OF POWER FACTOR IMPROVEMENT.—1. A power factor survey often will show several induction motors to be operating below rated output, at least part of the time. Kva. input to an induction motor has a lower power factor when the motor is operating below rated load than it has at full load. Interchanging motors to give each more nearly its rated load will improve conditions. Often it will pay to substitute new motors with better operating characteristics. 2. Replace induction motors with synchronous motors of unity or 0.8 power factor. 3. Install capacitors at points where loads are at low power factor. Maximum improvement in loading on generators, transformers, and distribution lines is obtained by installing corrective equipment near the terminals

of low power factor loads. If correction is made solely for rate reduction, corrective equipment can be installed at the terminus of the public utility line supplying the plant. 4. Install synchronous condensers.

TYPICAL PLANT POWER FACTOR SURVEYS.—**Example 1.**—Plant is supplied with power at 11,000 volts, 3-phase, transformed to 460 volts and distributed to the power circuits. A second transformer bank reduces voltage to 220 volts, 3-phase, for power, with a 3-wire single-phase arrangement from one transformer for 110/220 volts for lighting. Fig. 10 shows arrangement of circuits. The rate schedule allows a bonus for high power factor, power bill being reduced 0.5% for each percent average power factor is held above 80%. Test showed the average load to be 62 kw. and the reactive load to be 89 kva., corresponding to a power factor of 0.67. Existing equipment includes an air compressor, driven by an induction motor rated at 40 Hp., 1160 r.p.m., 220 volts. Compressor is driven at 115 r.p.m. through a speed reducer and belt, although suitable for operation at 200 r.p.m.

Two methods of improving power factor are available. 1. Installation of capacitors, the approximate savings being determined by test and from power bills by means of Table 2. 2. If the additional air can be used, it can be obtained together with plant power factor correction by replacing the induction motor driving the air compressor with a 75-Hp. 0.8 power factor, 440-volt synchronous motor. This will remove the inductive load and provide an equivalent correction of 65 kva. The substitution of the synchronous motor will cost, including reduced voltage starter, base and V-belt drive, $1850.00. The results are compressed air supply increased about 80%, and reduced transformer and drive losses. The cost of a 65-kva., 440-volt capacitor is approximately $650.00. The advisability of purchasing a new synchronous motor and changing the drive depends on the need for the additional compressed air capacity that would be available.

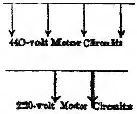


FIG. 10. Typical Industrial Plant Distribution System

Conclusion.—Table 2 shows a greater saving from the use of 130-kva. 440-volt capacitor equipment with incidental benefits of improved voltage regulation, increased transformer and feeder capacity, improved motor operation at peak loads and more uniform motor speed. Capacitors, either floor, ceiling, or wall mounted, installed at the point of origin of load will give a flexible arrangement and result in greatest benefit from power factor correction.

Table 2.—Capacitor Savings

Capacitor rating, kva......	0	50	60	70	80	90	100	110	120	130	140	150
Approx. delivered cost, $...	0	510	612	714	816	918	1020	1122	1224	1326	1428	1530
Expected power factor *....	0.57	0.78	0.825	0.865	0.895	0.92	0.95	0.97	0.99	0.997	0.999	1.00
Approx. annual savings, $..	0	‡	70	185	270	340	425	485	540	560	565	570
Investment return, %......	0	‡	12	26	33	37	42	43	44	43	40	37
Approx. savings in 5 yrs., $†	0	‡	¶	560	940	1240	1620	1880	2080	2170	2110	2080

* Based on average monthly power factor using ratcheted reactive meters. † Savings after deducting cost of capacitors with depreciation at 10% per year. ‡ Negligible. ¶ Small.

Example 2.—A flour mill driven entirely by electric motors may obtain a discount of 0.5% of its power bill for every 1% increase in power factor over 85%. The motors in the mill are loaded as follows, the total load being 74.0 kw.:

Motor Hp......................	200	40	20	15
Kw. input......................	103.8	9.0	6.4	3.2
Hp. output....................	126	9.0	7.3	3.5
Power factor..................	0.77	0.635	0.675	0.58

Recommendations.—To reduce energy consumption, the 15-Hp. motor should be changed to drive the grain elevator where it would approximately carry 10 Hp. maximum. A new 10-Hp. motor is recommended to drive the corn mill and feed plant and a new 5-Hp. motor for the wheat elevator. The motors would operate at near full load and effect the following savings in energy and improvements in power factor:

Load	Motor Rating	Load	Power Factor	Energy Saving per Month
Grain Elevator..............................	15 Hp.	9.0 Hp.	0.84	134 kw.-hr.
Corn Mill and Feed Plant..................	10 Hp.	7.3 Hp.	.83	36 kw.-hr.
Elevator in Mill..............................	5 Hp.	3.5 Hp.	.81	38 kw.-hr.
Total Load..................................	......	......	.775	208 kw.-hr.

In determining energy saving, 200 hr. of operation per month were used for the 5- and 10-Hp. motors and 100 hr. per month for the 15-Hp. motor.

No energy would be saved by using a smaller main motor. Even at less than full load, its efficiency is a little better than a smaller motor at full load. Of three possible methods of power factor correction, only synchronous motors or capacitors are considered, because for small amount of correction a synchronous condenser is the most expensive. In this case, a capacitor was lowest in first cost. A synchronous motor would require special design to get torque required for mill service and its efficiency would be no better than that of a slip ring motor. Starting characteristics of the slip ring motor are better for the present type of load.

Demand meter charts show loads of 120 kw. for grinding soft wheat and 136 kw. for grinding

hard wheat. For power factor correction a 60 kva. and a 90 kva. capacitor were considered with the following results:

	60-kva. Capacitor			90-kva. Capacitor		
Conditions	Present Power Factor	Expected Power Factor	Discount Earned	Present Power Factor	Expected Power Factor	Discount Earned
No Motor Changes						
Soft wheat............	0.74%	0.92	3 1/2%	0.74%	0.98	6 1/2%
Hard wheat...........	77%	.93	4%	77%	.985	6 3/4%
With Motor Changes						
Soft wheat............	74%	.95	5%	74%	Unity	7 1/2%
Hard wheat...........	77%	.96	5 1/2%	77%	Unity	7 1/2%

Cost of Equipment for Recommended Changes		Alternative	
1 new 10-Hp. motor and starter, 1 new 5-Hp. motor and starter } Total cost.........	$ 225	1 90 kva., 2200-volt, 3-phase, 60-cycle, capacitor equipment....	$1375
1 60-kva., 2200-volt, 3-phase, 60-cycle capacitor with racks and oil switches—Total cost.........	1040	Cost of new motors.......	225
Total cost complete......................	$1265	Total cost...........	$1600

3. EQUIPMENT FOR POWER FACTOR CORRECTION

SYNCHRONOUS MOTORS often can be used instead of induction motors, and the cost of corrective kva. then is chargeable to the difference in price between a synchronous motor with controls and an induction motor with controls. The cost of leading kva. obtained from synchronous motors increases rapidly for sizes below 75 Hp. See Fig. 11. Above 100 Hp., synchronous motors provide power factor correction at very low cost.

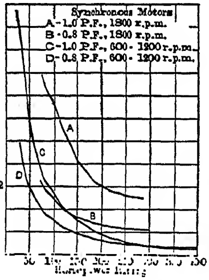


Fig. 11. Average 1935 Unit Prices of Leading Kva. Obtained by Synchronous Motor

CAPACITORS or static condensers are manufactured in small units; size depends on use. A capacitor unit comprises one or more flattened rolls of paper and aluminum foil pressed together and enclosed in a steel case with a fluid of good dielectric properties. Capacitors are applicable for power factor correction to four general types of service conditions: 1. To improve power factor of individual or groups of motors rated at 230, 460, or 575 volts. These units are for indoor installation and are enclosed. 2. To improve power factor at 230, 460, 575, 2300, 4000, or 4600 volts in small blocks at one or more points in a plant. These units are for indoor installation and are assembled in stacks. 3. Similar to (2) but for outdoor installation and pole mounting, for power factor correction at 2300, 4000, and 4600 volts. 4. To improve power factor of a large block of power at one point on circuits from 230 to 6900 volts. Suitable for either indoor or outdoor installation.

A capacitor equipment will include: 1. A group or stack of individual units, number depending on bank rating required. 2. Fuses for each individual unit to protect against a fault within the unit. 3. A circuit breaker that can serve as a main switch and provide over-current protection. The breaker must be capable of clearing successfully a short circuit between itself and the capacitor bank. 4. A discharge device which will allow charge on capacitor to leak off when bank is disconnected from the power circuit. Fig. 12 shows typical connections.

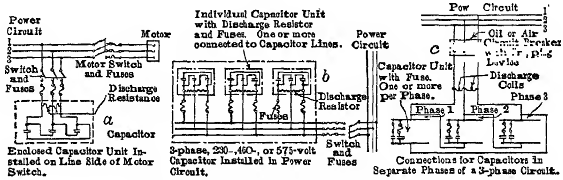


Fig. 12. Typical Capacitor Connections

Losses of capacitors, in kilowatts, will not exceed 1/3 of 1% of the kva. rating, including losses in individual units and all accessories.

Ambient Temperature.—Standard capacitors are designed for a maximum allowable temperature of 40° C.

Voltage Range.—Permissible working voltages for standard capacitors are:

Voltage Rating	230	460	575	2300	4000	4600	6900
Maximum Permissible Voltage	264	528	660	2640	4565	5280	7920

Installation and Maintenance.—The only foundation necessary for a capacitor bank is that necessary to support its own weight. Capacitors can be set on existing floors and natural ventilation is sufficient. They can be installed in isolated locations and inspected monthly. No adjustment or frequent attention is necessary. Low maintenance is an important factor in the economics of capacitor installations.

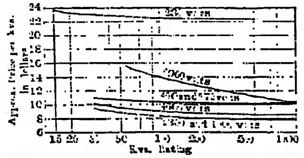


FIG. 13. Approximate 1935 Prices of Standard Capacitors, Including Racks, Indoor Type

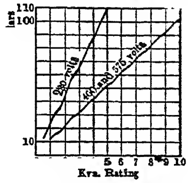


FIG. 14. Approximate 1935 Prices of Standard Individual Capacitor Units for Small Loads

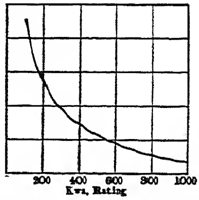


FIG. 15. Approximate 1935 Prices of Standard Synchronous Converters

Sizes of Capacitor Units Available.—Enclosed units for application to individual motors, or other small loads, are made in sizes of 0.5 to 5 kva. at 230 volts, 1-, 2-, or 3-phase, 60-cycle, and 1 to 10 kva. at 460 and 575 volts. For higher voltages and when power factor correction is applied to the distribution bus, or primary feeder, individual 5-, 10-, or 15-kva. units are assembled in racks and a sufficient number connected in parallel to give the required bank capacity. Figs. 13 and 14 show approximate 1935 prices of capacitors.

SYNCHRONOUS CONDENSERS are similar in construction to A.C. generators. They are designed to float on the line, fully excited, without load, and to supply leading kva. The rotor has a squirrel cage winding to assist in starting and to prevent hunting while in operation.

Standard Sizes range from 100 kva. up, with direct-connected exciters. Ratings below 100 kva. can be obtained, but for power factor correction below 100 kva., a capacitor usually is used. Control can be manual or automatic. A manual equipment will include one each of the following: Starting auto-transformer, condenser field rheostat chain operated, exciter field rheostat, indicating lamp for pilot light, ammeter, voltmeter, field ammeter, reactive-volt-ampere meter with auto-transformer, field switch, concentric operating mechanism for condenser and exciter field rheostats, main and starting oil circuit breaker. Separate main and starting breakers may be necessary.

Cost.—Fig. 15 shows the approximate 1935 price of synchronous condensers between 100 and 1000 kva. including simple manual control.

Synchronous Condenser Losses in kilowatts loss percent of kva. rating from 100 to 1000 kva. are shown in Fig. 16.

CHOICE OF SYNCHRONOUS CONDENSERS OR CAPACITORS FOR POWER FACTOR CORRECTION.—Power factor clauses in rate schedules usually are classified as: 1. Where power factor is determined during period of maximum demand. 2. Where power factor is determined as monthly average. If the power factor clause is in Group 1, corrective equipment, which will supply a large amount of leading kva. when the demand is maximum, is necessary. During periods of light loads, power factor is not important, which favors the use of synchronous motors or synchronous condensers. The cost of leading kva. is low and losses and maintenance are not important, as corrective equipment may operate only a few hours per day.

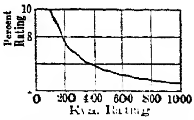


FIG. 16. Synchronous Condenser Losses

When the power factor clause is in Group 2, capacitors are preferred. Power factor will be lower during periods of light loads, because ratio of magnetizing current to load current then is higher than at full load. To improve average power factor, corrective equipment always must be connected to the line. While the cost per leading kva., is higher for a capacitor, its low loss and negligible maintenance often are the deciding factors.

The amount of leading kva. that economically can be provided in a given plant is limited. Reactive kva. meters cannot run backwards, and there is no economy in providing equipment for power factor improvement from lagging to leading.

4. METHODS OF POWER FACTOR MEASUREMENT

INDICATING POWER FACTOR METER.—Fig. 17 shows the connections for this type of meter. Its principle is based on the two-wattmeter method of measuring power in a 3-phase circuit, where the power factor or

where W_1, W_2 are the two wattmeter readings. The only variable in this equation is the ratio W_2/W_1. The indicating meter is designed to indicate this ratio and is calibrated to read directly in power factor.

THE RECORDING METER works on the same principle as the indicating meter, but includes a chart and a stylus for making a permanent, continuous record of power factor.

INTEGRATING METERS are preferred for billing purposes because a more accurate record may be obtained which is simpler to analyze. These meters do not require the attention necessary for recording instruments.

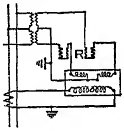


FIG. 17. Connections for a 3-phase Indicating Power Factor Meter

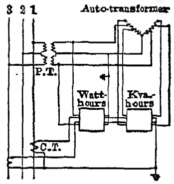


FIG. 18. Meter Connections for Measuring Reactive Volt-amperes or Reactive Volt-ampere-hours

REACTIVE VOLT-AMPERE-HOUR METERS.—Power factor can be obtained readily from integrating meters by the combined use of a watt-hour meter and a reactive-volt-ampere-hour meter. Fig. 18 shows connections for leading reactive-volt-amperes, or reactive-volt-ampere-hours.

CALCULATIONS OF POWER FACTOR FROM WATT-METER AND REACTIVE-VOLT-AMPERE-METERS.—The readings of reactive volt-amperes, or reactive-volt-ampere-hours cannot be used to determine power factor unless wattmeter or watt-hour readings also are taken. If V = volts, A = amperes, VA_r = reactive volt-amperes, W = watts, $W/VA = \cos\theta$, or power factor, which is the quantity desired; $W/VA_r = \tan\theta$, or power factor $= \cos\tan^{-1}(VA_r/W)$.

Apparent power, or volt-amperes, $VA = \sqrt{W + (VA_r)^2}$.

POWER DISTRIBUTION

1. PRINCIPLES OF DESIGN

Design of a distribution system must take into account the physical layout of the plant, future growth, kind of power required, type of loads in various departments, economy, continuity of service, voltages, protection and code requirements. For any given system several arrangements are possible, ranging between maximum economy and maximum continuity of service. Comparative data should be tabulated, and the final design then can be made incorporating desired features at minimum cost.

PRIMARY DISTRIBUTION.—The extent of primary distribution depends on plant layout. Where distribution is confined to one building and small loads, the most economical distribution will be at utilization voltage. If power must be distributed over a large area, and in relatively large blocks, it is economical to run primary feeders from the main substation or generating station to load centers and there step down voltage for secondary distribution.

Primary Voltage will depend on load carried by primary feeders, total number of feeders, and distance of transmission from power source to load centers. Generally, higher voltage decreases power loss, but increases cost of line construction. A common primary distribution voltage for power is 2300 volts, but for any given installation the most economical voltage must be determined by comparing the cost of cable and wire. Whether primary distribution or distribution at utilization voltage is best can be decided by a cost comparison of several plans, which consider: 1. Character of loads. 2. Cost of low-voltage feeders plus capitalization of higher line losses as compared with lower line losses plus line costs for primary distribution at a higher voltage, plus cost of additional switching apparatus and transformers to step down to utilization voltage at load centers. 3. Size of motors. 4. Physical layout, which determines area over which power must be distributed, and to some extent, voltage of the power source. In small industrial plants, secondary distribution only is possible, as power is supplied from public utility lines through transformers at 230 or 450 volts.

Secondary Voltages are determined by the apparatus being supplied. It is difficult to build satisfactory small motors for high voltage. Small and medium size motors are

standardized at 220, 440 and 550 volts. Also in many applications of small motors, high voltages are undesirable because of hazards to operators. As motor size increases, low voltage becomes less desirable because of the large currents to be handled. Larger size motors are standardized at 2200 volts, and often can be operated directly from primary feeders. Large quantities of power at low voltage cannot be transmitted over any great distance without a voltage drop below the minimum allowable at the receiving end of the circuit. See Tables 10 and 12. Hence, it is economical in larger plants to use primary distribution to supply a secondary system through transformers or conversion equipment at load centers. Too many different voltages will complicate distribution and increase cost. Usual practice is to operate all small equipments at 220 or 440 volts, and sometimes at 550 volts. Common practice is 440 volts for motors of 50 Hp. and less, and 2200 volts for larger motors.

ARRANGEMENT AND DESIGN OF FEEDER CIRCUITS.—Under normal operating conditions, voltage at the load should not vary more than $\pm$ 10% from normal, which is the maximum allowable deviation recognized by the A.I.E.E. and maximum for which motor manufacturers' guarantees will apply. Feeder circuits, therefore, must be designed on the basis of load requirements.

Individual feeder circuits in industrial plants may be subject to considerable load fluctuation, especially if load consists of a few large units. A load of 10 or more units of uniform size usually can be considered as constant, even with wide fluctuations on individual units, unless peak loads on several units occur simultaneously.

Laying Out a Feeder Circuit requires determination of the overall load cycle. The feeder must be designed with ample capacity to carry the root-mean-square value of current represented by the fluctuating load without overheating, and to deliver sufficient voltage under peak load conditions. The various departments of an industrial plant should have an estimated daily load curve (see p. 15–22) for each season of the year. This will give the average yearly load curve, which, with proper consideration of future growth, will determine the capacities of departmental feeders.

Heating often is the limiting factor in short feeders. Thermal protection usually is not provided for feeder circuits. Occasional temperature measurements usually are sufficient to indicate circuit conditions.

Voltage Drop more often is the limiting factor in longer circuits. It is first necessary to determine what voltage drop can be tolerated. Peak loads may be due either to a combination of loaded units, no one of which operates above normal rating, or to overloads on one or more units. A greater voltage drop can be tolerated in the first case than in the second. It is difficult to establish definite voltage limits, since much depends on peak loads to be carried by motors and the importance of feeder circuits.

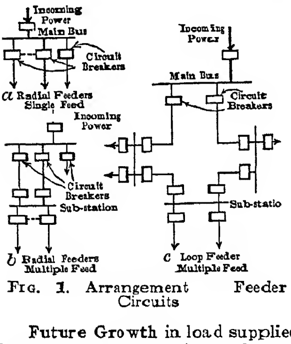


FIG. 1. Arrangement Feeder Circuits

Motor Starting Loads must be considered in design of feeder circuits. Many motors, including medium sizes, are started on full voltage if power system conditions permit. Circuits supplying such motors must be more liberally designed to limit voltage drop. Starting kva. required by motors may be from approximately five to nine times normal at full voltage, and starting power factor may vary between 20% and 60% depending on type of motor; under such conditions voltage drop in the feeder circuits may be excessive. Voltage must not drop so low that the motor being started is unable to develop sufficient torque, nor so low that under-voltage devices operate to shut down other motors connected to the feeders.

Future Growth in load supplied by the feeder circuit can be provided for initially by a feeder of larger capacity, or by providing for addition of duplicate feeders.

Continuity of Service required depends on character and importance of load. In continuous processes, continuity of service is highly important and more than one feeder will be required. With two feeders, either must be able to carry the total load. With three feeders, any two must carry the total load. Feeders can be taken over separate routes to eliminate possibility of trouble involving more than one feeder. Selection of high quality cable and equipment, careful observance of codes, and approved type of construction will give greater continuity of service.

CIRCUIT ARRANGEMENTS for distribution generally are: 1. Stub or radial feeders, single or double, Fig. 1 *a*. 2. Parallel feeders, Fig. 1 *b*. 3. Loop feeders, Fig. 1 *c*. 4. Network, or combination of radial and loop; power is supplied to the loop at several

points and radial feeders are taken off for individual loads. Radial and loop feeder systems cover most industrial plant requirements.

A stub or radial feeder circuit is separated from all others and prevents feed back; load can be supplied at any point on the line. Continuity of service is limited unless duplicate feeders are provided as in the parallel system of Fig. 1 *b* where the degree of continuity is high. The loop feeder circuit leaves the power source, loops through several distribution points and returns to the source. Continuity of service is about the same as for the parallel feeder. Its use depends on the physical layout of the plant and may prove economical in supplying power over wide areas. The fundamental circuit arrangement can be modified to meet conditions in any particular plant. Some flexibility is desired and often necessary to insure uninterrupted power, which is obtained in the switching arrangement and number of circuits. On some feeder circuits power can be shut off long enough for inspection of apparatus, but on others power always must be available. Switching arrangements must provide for isolation of apparatus without interrupting power.

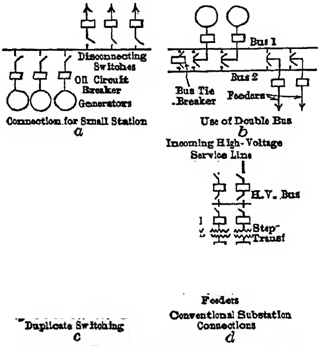


FIG. 2. Typical Distribution System Connections

DEPARTMENTAL SUBSTATIONS are required: 1. Where utilization voltage differs from distribution voltage. 2. Where direct current must be provided. The substation should be located at the load center, and the switching apparatus used will depend on flexibility, degree of reliability and inspection required. Transformers should be located at load centers. Usual connections for power are 3-phase, delta-delta, and for lighting single-phase or 3-phase, 4-wire.

Transformer Capacity Required depends on load characteristics and future growth. All-day efficiency should be considered in selecting transformers. If core loss is low and copper loss high, efficiency will be highest at part load. If the reverse is true, efficiency will be higher at full load. If average demand is low, select a transformer with highest efficiency at low loads.

All-day Efficiency of a distribution transformer is $E = (100P_2h)/(P_2h + Ah_1 + hI_2^2R_2)$, where E = all-day efficiency, percent; h = hours per day of secondary load; h_1 = hours per day transformer is on line; A = core loss, watts; P_2 = secondary output, watts; I_2 = secondary current, amperes.

Total resistance = $R_2 = (r_1^2/a^2) + r_2$, where a = ratio of transformation; r_1, r_2 = primary and secondary resistance.

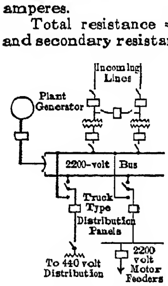


FIG. 3. Flexible Connections for Purchased and Generated Power

CONTROL EQUIPMENT required depends on protection of feeders, tie lines, transformers and rotating apparatus, and short circuit conditions. When the distribution circuits have been laid out, it is necessary to connect them in a distribution system. If solidly connected, trouble on any one circuit would cause complete loss of power. System connections, therefore, are made through control devices. Beginning at the main generating station, or main substation, generators or transformer banks should be provided with control equipment which will permit isolation of any one unit. Feeder circuits leaving the main station should have similar control. Main distribution centers will require, in general, a bus structure, feeder switches and switchboard and metering equipment for each circuit. If power is purchased, the typical incoming panel will include disconnecting switches, circuit breaker, a watthour meter, a curve-drawing demand meter and, if the power contract includes a power-factor clause, a kva. meter.

TYPICAL SYSTEM CONNECTIONS of Fig. 2 represent common practice and are flexible enough to meet most service conditions. Fig. 2 *a* is the simplest, each generator being connected through a circuit breaker, and disconnecting switch to a common bus serving various feeders. Fig. 2 *b* uses a double generator bus with disconnecting switches to permit connection of any generator or feeder to either bus. The bus tie breaker facilitates transfer of apparatus from one bus to the other. Fig. 2 *c* shows a more elaborate scheme using duplicate switching throughout. Fig. 2 *d* shows connections for a conventional substation. This arrangement can be modified to increase flexibility. Fig. 3 shows an arrangement used in a distribution system where part of the power is

generated and the balance purchased. The various loads can be supplied from either the plant generator or the public utility. The feeder switchboard consists of truck type panels which can be disconnected from the switchboard assembly and rolled out for inspection of circuit breaker and associated devices.

2. SHORT CIRCUITS

No power system should be laid out without consideration of possible short circuit currents. The magnitude of short circuit currents should be determined at all strategic points on the system, and arrangments made to remove automatically from service any circuit or piece of apparatus in which a short circuit occurs. It also is essential that the faulty circuit or apparatus be disconnected from the power source without interrupting service on other circuits. This is done with differential and other selective relaying schemes with circuit breakers.

EFFECT OF FUTURE GROWTH ON SHORT CIRCUIT CURRENTS.—Short circuit currents increase simultaneously with system capacity. Present circuit interrupting devices and bus structures may be inadequate within a few years. The increase in short circuit duty with plant growth is greater at the main power station. The number of circuits increases with increased power demand, with a proportionate increase in the number of breakers. At the same time the short circuit duty on all breakers in the plant increases. Two remedies are possible when power requirements reach the point where circuit breakers are inadequate: 1. New circuit breakers to meet requirements. 2. Current limiting devices to keep short circuit currents at about the original value. In small systems, it usually is less expensive to meet increased short circuit duty with new breakers. As the system further increases in size, the cost of current limiting equipment becomes less than the cost of new breakers. The size to which a system must increase to reach this critical point increases with system voltage and depends largely on the type and number of circuits, which should be considered when selecting the original system voltage.

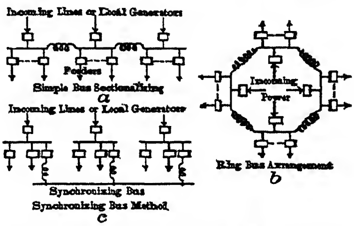


FIG. 4. Methods of Reducing Short Circuits

LIMITATION OF SHORT CIRCUIT CURRENTS.—Short circuit currents are limited by increasing the circuit impedance between power source and point of short circuit. Series reactance added to the circuit would produce the desired result, but at the expense of increased voltage regulation under normal operation. Since full feeder kva. would be transmitted through the reactance, feeder loss would be increased. A more efficient method is to split the power system into independent generator and load sections, tied together through current limiting reactors so located as to give a maximum of short circuit protection with a minimum of interference with normal system operation. Fig. 4 *a* shows such an arrangement. If power fails on one bus section, essential load can be supplied from adjacent sections through the reactors without serious voltage drop. With load properly proportioned on each bus section, little or no current flows through the interconnecting reactors and the system operates as if solidly tied together. If a short circuit occurs on any bus section or its feeders, the reactors are effective in limiting short circuit current fed from adjacent sections. Three general arrangements for applying current limiting reactors are: Bus sectionalizing, Fig. 4 *a*; the ring bus arrangement, Fig. 4 *b*; the synchronizing bus, Fig. 4 *c*. Heavy short circuit currents largely are limited to systems of 2200 volts and above. On circuits of 230 or 450 volts, the reactance of distribution network, circuit breakers, bus structures and the short circuit arc itself is effective in limiting maximum short circuit current, which experience indicates will not exceed 20,000 amperes.

MAGNITUDE OF SHORT CIRCUIT CURRENT.—Considering only A.C. circuits and apparatus, the current flowing into a short circuit is limited by the impedance between generator and point of short circuit. Impedance $= Z = \sqrt{r^2 + X^2}$, where r = resistance, and X = reactance to point of short circuit. For usual circuits and apparatus for power generation and distribution, resistance can be neglected. Results still will be within the required accuracy. The short circuit current then is $I = E/X$, where E = generator voltage; X = instantaneous value of reactance to point of short circuit.

SHORT CIRCUIT OF A.C. GENERATORS.—At the instant of short circuit of an A.C. generator, current flow will be considerably higher than at a fraction of a second later. Initial current will decrease exponentially with time until sustained at a lower value. The reactance exhibited at the instant of short circuit, called the sub-transient reactance, is used in short circuit calculations. It ranges from approximately 8 to 15% for high-speed turbine generators, from 15 to 25% for salient-pole, medium-speed gen-

erators, and 25 to 35% for slow-speed, engine-type generators. The effective short circuit current may be a maximum of 1.73 times that determined by the sub-transient reactance, depending on the point of the voltage wave at which short circuit occurs. As time elapses, current decreases so that when circuit breakers are able to operate, 1/4 to 1/3 second later, the current will have decreased to approximately 30 to 60% of maximum value, depending on amount of reactance external to the generator, and between generator and point of short circuit.

CALCULATION OF SHORT CIRCUIT CURRENT IN A.C. CIRCUITS.—It is customary to express reactances of generators and transformers in percent of rated capacity; *i.e.*, a generator with 10% sub-transient reactance would deliver 100/10 = 10 times rated current if short circuited at its armature terminals with full load excitation. It is more convenient to use percent reactance in short circuit calculations, and assume a common base kva. to which all individual reactances are converted. Line reactances usually are given in ohms, and when two parts of a distribution system are connected through transformers, these reactances must be put on a common voltage basis as follows: Let X_1 = actual ohms reactance; X_2 = ohms at assumed common voltage; E_1 = actual circuit voltage; E_2 = common voltage chosen. Then $X_2 = X_1\ (E_2/E_1)^2$.

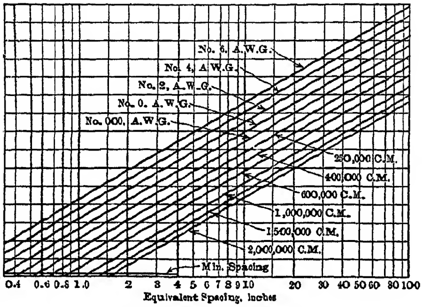


Fig. 5. Reactance Chart

To convert ohms reactance to percent reactance: $X\% = (X_2 \times \text{kva. base})/(\text{kv.}^2 \times 10)$, where X_2 = ohms reactance and kv. = (line voltage/1000).

Fig. 5 is a chart for determining the reactance at 60 cycles of conductors of various sizes, spaced at various distances. The chart is entered at the bottom on the ordinate of equivalent spacing. Opposite the intersection of this ordinate with the curve of conductor size, read reactance on the scale at the left. The equivalent spacing of 3-phase conductors in the same plane at a distance a apart is 1.26 a. The equivalent spacing of 3-phase conductors not in the same plane is $\sqrt[3]{a \times b \times c}$, where a, b and c are the distances between pairs of conductors.

To obtain reactance at other than 60 cycles, multiply the reactance at 60 cycles by the following factors: 25 cycles, 0.4167; 50 cycles, 0.8333; 120 cycles, 2.0; 180 cycles, 3.0; 240 cycles, 4.0; 300 cycles, 5.0.

Example.—Fig. 6 represents a 3-phase, 60-cycle system, power being generated at 6600 volts on one 2000-kva. and one 5000-kva. generator, with 12% and 8% sub-transient reactance, respectively, and transmitted over two parallel feeders to a substation where voltage is stepped down to 2200 volts. Required value of short circuit current at points A and B.

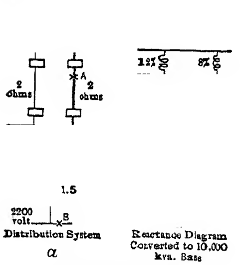


Fig. 6. Data for Short Circuit Calculation

Assume transformer to have 6% reactance on its rating of 500 kva. Reactance of each transmission line will be 2 ohms per conductor, and of 2200-volt feeder, 1.5 ohms per conductor. Assume 10,000 kva. as a convenient base, and a common voltage of 6600 volts. Equivalent reactance of 2200-volt feeder = $1.5(6600/2200)^2 = 4.5$ ohms. Fig. 5 b shows all reactances converted to percent on the common base of 10,000 kva.

Reactance of generator No. 1	$= 12(10{,}000/2000)$	= 60%
Reactance of generator No. 2	$= 8(10{,}000/5000)$	= 16%
Reactance of transmission line	$= (2 \times 10{,}000)/(6.6^2 \times 10)$	= 46%
Reactance of transformer	$= 6\,(10{,}000/500)$	= 120%
Reactance of 2200-volt feeder	$= (4.5 \times 10{,}000)/(6.6^2 \times 10)$	= 103%

t at A is fed by two generators in parallel having equivalent reactance of + 16) = 12.6%. Since a generator with 100% reactance will deliver normal current on short circuit, a generator of 12.6% reactance will deliver 100/12.6 = 7.9 times normal current on short circuit. With base kva. of 10,000,

$$\text{Initial current} = \{(7.9 \times \text{kva. base})/1.73E_2\} \times 1000$$
$$= (7.9 \times 10{,}000 \times 1000)/(1.73 \times 6600) = 6940 \text{ amperes minimum.}$$

Maximum value = 1.73 × 6940 = 12,000 amperes. Since current is limited by only reactance

of generators, interrupting duty on the circuit breakers will be about 33%, assuming an operating time of 1/4 sec. Circuit breaker must interrupt 0.33 × 12,000 = 3960 amperes, but must have mechanical strength to withstand an initial current of 12,000 amperes.

Total reactance to point B is: Two generators in parallel, 12.6%; two lines in parallel, 23.0%; reactance of transformers, 120%; reactance of 2200-volt feeder, 103.0%; total, 258.6%. Short circuit current = 100/258.6 = 0.386 × normal current = 0.386 × 10,000 × 1000/2200 = 1756 amperes maximum. Interrupting duty = 0.57 × 1756 = 1000 amperes.

Calculations are laborious on complex distribution systems. The larger manufacturers of electrical apparatus have available calculating tables for making short circuit studies of complicated systems.

SHORT CIRCUIT PRECAUTIONS.—1. Construction should be such as to confine any fire resulting from a short circuit to a small part of the installation. 2. All apparatus should have sufficient mechanical strength to resist stresses resulting from current flow during short circuit. 3. Short circuit current should not be allowed to flow any longer than necessary. A great amount of heat is generated by abnormal currents which quickly damages insulation. 4. All circuit breakers that are required to open on short circuit should have ample capacity to interrupt safely the maximum current flowing at the instant they operate.

3. CIRCUIT CONSTANTS

Every circuit or transmission line has definite electrical characteristics or constants depending on the material of conductors, their cross-sectional areas, outside diameters and spacing. Electrically, each line consists of resistance and inductance in series, through which current must flow, and a capacitance between conductors into which, in A.C. circuits, charging current flows.

Resistance of a unit length of line is a function of the material of the conductors and cross-sectional areas.

Inductance of a unit length of line is a function of the outside diameter of conductors and the distance between them. The inductance of one conductor of a single- or 3-phase circuit may be determined from:

$$L = \{1.41 \log_{10}(s/r) + 0.1524\} \times 10^{-4}, \quad [1]$$

where (s/r) is large as in overhead lines, or

$$L = [1.41 \log_{10}\{(s - r)/r\} + 0.1524 + 0.304(d/s)] \times 10^{-4}, \quad [2]$$

where (s/r) is small as in multi-conductor cables. L = inductance of one conductor, henrys per 1000 ft.; s = spacing between center lines of conductors, in.; r = radius of conductor, in.; d = diameter of conductor, in. If conductors of a 3-phase circuit are horizontally spaced, $s = \sqrt[3]{a \times b \times c}$, where a, b, c = respectively distance between phases 1 and 2, 2 and 3, and 1 and 3.

Capacitance of a line is the function of the outside diameter of the conductors and their spacing. Where (s/r) is large

$$C = \{7.354/\log_{10}(s/r)\} \times 10^{-9}, \quad [3]$$

where C = capacitance, farads per 1000 ft., of one conductor to neutral. In industrial distribution systems, circuits are too short to cause appreciable error if capacitance is neglected.

Kva. Power, Voltage and Current Relations in various types of circuits are given in Table 1.

Table 1.—Formulas for Amperes, Horsepower, Kilowatts and Kilovolt-Amperes *

Desired Data	Alternating Current			Direct Current
	Single-phase	2-phase, 4-wire †	3-phase	
Kilowatts	$IEF/1000$	$2\,IEF/1000$	$1.73 \times IEF/1000$	$IE/1000$
Kilovolt-amperes	$IE/1000$	$2\,IE/1000$	$1.73\,IE/1000$	....
Horsepower	$IEeF/746$	$2\,IEeF/746$	$1.73 \times IEeF/746$	$IEe/746$
Amperes	Hp. × $746/EeF$	Hp. × $746/2\,EeF$	Hp. × $746/1.73\,EeF$	Hp. × $746/Ee$
Amperes	Kw. × $1000/EF$	Kw. × $1000/2EF$	Kw. × $1000/1.73EF$	Kw. × $1000/E$
Amperes	Kva. × $1000/E$	Kva. × $1000/2E$	Kva. × $1000/1.73E$	....

* I = amperes; E = volts; e = efficiency; F = power factor; Hp. = horsepower; Kw. = kilowatts; Kva. = kilovolt-amperes.

† In 3-wire, 2-phase circuits, the current in the common conductor is 1.41 times current in either of the other conductors.

VOLTAGE DROP, POWER LOSSES AND POWER FACTOR IN POWER TRANSMISSION easily can be determined if line or cable constants and characteristics of power supplied to or received from the line are known. The most common methods of trans-

mission are: Direct current, and single-phase, 2-phase, 4-wire, and 3-phase alternating current.

Notation.—E = voltage between conductors; e = voltage line to neutral = $E/1.73$; f = frequency, cycles per sec.; I = current, amperes; kva. = kilovolt-amperes; L = inductance of one conductor, henrys; P = power, kilowatts; R, r = resistance of one conductor, ohms; X = reactance of one conductor, ohms = $2\pi fL$; Z = impedance of one conductor, ohms $= \sqrt{r^2 + X^2}$; θ = displacement angle between voltage and current (see p. 15–53); $+\theta$ = lagging current; $-\theta$ = leading current; $\cos\theta$ = power factor. Subscripts g, r, l, respectively, indicate: at generating end, at receiving end, and loss in line.

DIRECT CURRENT.—$P_g = E_gI/1000$; $P_r = E_rI/1000$. $E_l = 2RI = E_g - E_r$.

$$_l = E_l I/$$

$$\text{Percent regulation} = (E_l/E_r) \times 100;$$

$$\text{Percent power loss} = (P_l/P_g) \times 100.$$

ALTERNATING CURRENT.—**Single Phase.**—

$$E_g = \sqrt{(E_r \cos\theta_r + 2rI)^2 + (\pm E_r \sin\theta_r + 2XI)^2};$$

$$\text{Percent voltage regulation} = \{(E_g - E_r)/E_r\} \times 100;$$

$$\cos\theta_g = (E_r \cos\theta_r + 2rI)/E_g;$$

$$\text{Kva}_g = E_g I/1000 = 2e_g I/1000;$$

$$P_g = E_g I \cos\theta_g/1000 = 2e_g I \cos$$

$$P_l = P_g - P_r = 2rI^2/1000;$$

$$\text{Percent power loss} = (P_l/P_g) \times 100.$$

Three Phase.—

$$E_g \quad 1.73\sqrt{(e_r \cos\theta_r + rI)^2 + (\pm e_r \sin\theta_r}$$

$$\text{Percent voltage regulation} \quad \{(E_g - e_r)/E_r\} \times 100;$$

$$\cos\theta_g - (e_r \cos\theta_r + rI)/e_g;$$

$$\text{K}\ _g = 1.73E_g I/1000 = 3e_g I/1000;$$

$$P_g = 1.73E_g I \cos\theta_g/1000 = 3e_g \cos\theta$$

$$P_l = P_g - P_r = 3rI^2/1000;$$

$$\text{Percent power loss} = (P_l/P_g) \times 100.$$

E.—A substation operates at 2200 volts, 60 cycles, 3-phase for a load of 250 kw. at 80% power factor lagging. Power is received from a generating station, 5000 ft. distant over a line of three No. 4 copper cable conductors spaced 24 in. apart in a horizontal plane.

From Table 4, R per conductor = 0.258 × 5 = 1.29 ohms. Equivalent equilateral spacing is $S = \sqrt[3]{a \times b \times c} = \sqrt[3]{24 \times 24 \times 48} = 30.2$ in. From Fig. 5, for an equivalent spacing of 30.2 in., reactance = 0.135 ohms per 1000 ft. per conductor at 60 cycles. For 5000 ft., X = 5 × 0.135 = 0.675 ohms. Reactance also may be calculated from the inductance and frequency by means of equation [1]. From equation [1], inductance per conductor =

$$L = \{1.41 \log_{10}(30.2/0.117) + 0.1524\} \times 5 \times 10^{-4} = 0.00178$$

Reactance = $2\pi f L = 2 \times 3.1416 \times 60 \times 0.00178 = 0.675$ ohms per conductor.

Impedance per conductor = $Z = \sqrt{r^2 + X^2} = \sqrt{1.29^2 + 0.675^2} = 1.454$ ohms.

$I = (250 \times 1000)/(1.73 \times 2200 \times 0.8) = 82$ amperes.

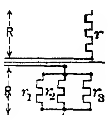
$\cos\theta_r = 0.8$; $\theta_g = 36°\ 52'$; $\sin\theta_r = 0.6$.

Voltage at receiving end = $e_r = 2200/1.73 \quad 1270$ volts.

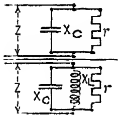
Voltage at generating end = E_g

$$1.73\sqrt{(1270 \times 0.8 + 1.29 \times 82)^2 + (1270 \times 0.6 + 0.675 \times 82)^2} = 2400 \text{ volts}$$

Table 2.—Relation between Resistance and Reactance



$R = 1/(1/r_1 + 1/r_2 + 1/r_3)$



$1/\sqrt{(1/r^2) +}$

Percent voltage regulation = 100 × (2400 − 2200)/2200 = 9.1%.
Power factor at generating end of line is,

$$\cos\theta_g = (1270 \times 0.8 + 1.29 \times 82)/(2400/1.73) = 0.807$$

Kva. at generating end = Kva._g = (1.73 × 2400 × 82)/1000 = 341
Line loss = P_l = $(3 \times 1.29 \times 82^2)/1000$ = 26 kw.
Power supplied to line = P_g = (1.73 × 2400 × 82 × 0.807)/1000 = 276 kw.
Percent power loss = (26/276) × 100 = 9.4%.

Two-phase, Four-wire.—Calculations for transmission of power 2-phase, 4-wire, can be made by considering the 2-phase circuit as two independent single-phase circuits, each carrying half the power, and making use of the single-phase formulas.

Relation between Resistance, Inductive Reactance and Capacitive Reactance in A.C. circuits are shown by the formulas and diagrams of Table 2, in which R, r = resistance; X_L, X_c = inductive and capacitive reactance, respectively; Z = impedance.

4. METHODS OF DISTRIBUTION

OVERHEAD DISTRIBUTION has the advantages of low initial cost, high overload capacity and ease of modification or change in circuit arrangement. Tap-offs easily can be made at any point. Overhead lines may be a hazard to life. They are subject to damage and require lightning arresters for protection. Wood poles, usually spaced 120 to 125 ft. apart, are most common in overhead distribution. Pole height will depend on required clearances. Usual practice is to creosote poles to about two feet above ground to prevent deterioration and damage from insects. Pole hardware is standardized and includes cross arms and braces, insulators and guys where necessary.

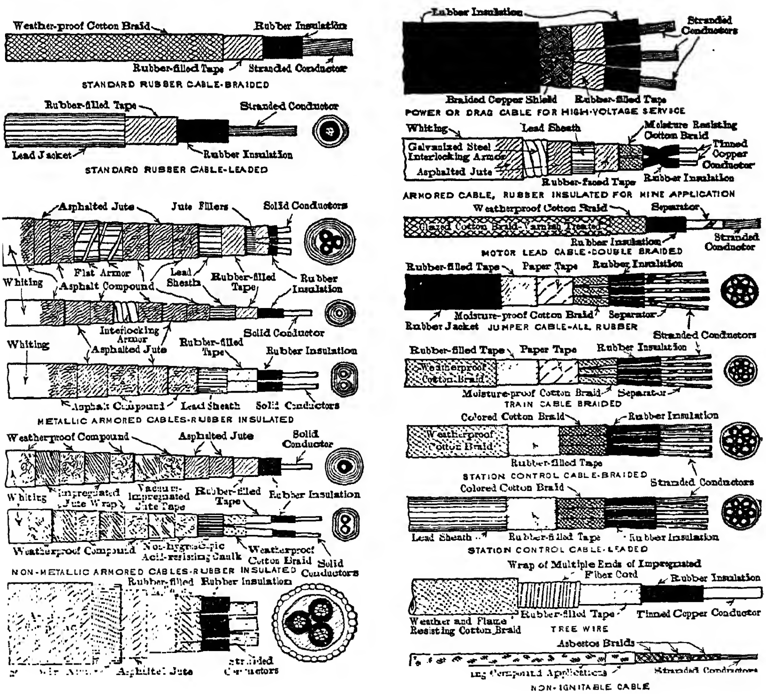


FIG. 7. Types of Cable Construction

UNDERGROUND DISTRIBUTION depends on permanency of installation, flexibility, physical layout of plant and comparative costs. If an industrial plant requires underground tunnels for steam, water and gas, they also can be used for electrical circuits, providing proper physical separation is maintained. Underground distribution requires cable which may be carried in fiber or tile conduit embedded in concrete with concrete separation between ducts. Spare ducts are provided for future requirements. For short cable runs within buildings, such as runs from generating or conversion apparatus to switchboards, common practice is to install cable in pipe conduits in the floor. For distribution to isolated points, or for outdoor lighting circuits, parkway type cable can be used. Table 11 gives data on conduits.

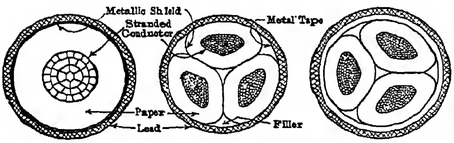


Fig. 8. Shielded and Non-shielded Cable

CABLE consists of solid or stranded conductor, which may be bare for overhead line construction or insulated for underground and conduit wiring. Common insulating materials are: Asbestos, paper, rubber and varnished-cambric. Common finishes are: Asbestos, braid, braided-wire armor, cotton, flat-band armor, compound sheath, interlocked armor, asphalted jute, lead sheath, rubber, tape and wire armor. Fig. 7 shows various cable constructions.

Paper-insulated Cable is generally accepted for transmission of large blocks of power, underground, overhead or underwater, and also for interconnection of apparatus in generating stations, substations and industrial plants. The two types, solid and oil-filled, are insulated with wraps of paper impregnated with oil. They may be shielded or non-shielded. See Fig. 8. A metal shield comprising a thin tape of copper is wound tightly over the paper insulation to protect it from corona or ionization damage. Single-conductor shielded cable is recommended for circuits of from 40 to 69 kv., but it can be obtained for voltages up to 75 kv. The multiple-conductor shielded cable is recommended for circuits of from 10 to 35 kv. and is adaptable to small and medium size conductors. Non-shielded cable has a belt of insulation over the insulated conductors. Single-conductor unshielded cable is recommended for any voltage up to 39 kv. and is adaptable to large conductor sizes. Unshielded multiple-conductor cable is recommended for circuits of any voltage up to 10,000 kv.

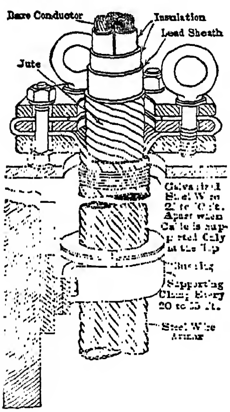


Fig. 9. Methods of Supporting Vertical Cable

Paper-insulated cables are finished with a lead sheath. Armor also is applied to paper-lead cables to prevent corrosion of the lead and prevent mechanical injury. The armor may be flat band steel applied over jute. Cables buried directly in the ground frequently have an overall layer of asphalted jute to protect against corrosion. Steel wire is used for armor if tensile strength and resistance to mechanical injury are required. The lead sheath may be omitted on paper insulated cable for voltages up to 17 kv. and a rubber hose jacket substituted. Such cable is suitable for high voltage overhead construction, where the weight of a lead sheath is undesirable.

Varnished-cambric-insulated Cable is highly moisture resisting but not completely waterproof. It is used for a wide variety of conditions, but it is important that the correct finish be used. In well ventilated exposed positions, braid finish, either weatherproof or flame-resisting as required, can be used. The use of non-metallic finishes without lead or metallic shields (grounded) is not recommended for circuits above 7500 volts except where cable is supported free on insulators.

APPLICATIONS OF VARNISHED-CAMBRIC-INSULATED CABLE.—Central Stations and Substations.—Varnished-cambric cable in all sizes is used for general station wiring and interconnection of apparatus. Single-conductor cable is usual, although multiple-conductor cable often is used for control and auxiliary circuits, handling small blocks of power.

Industrial Plants.—Cambric cable can withstand sharper bends than paper cable without injury to the insulation and it will stand higher temperatures than rubber cable. Varnished-cambric cable is ideal for small systems such as industrial plants in which both single- and multiple-conductor cable are widely used. Any of the standard finishes can be used depending on the service requirements.

Mines.—Varnished-cambric cable with lead sheath is used for horizontal runs in mines and for vertical mine-shaft runs when finished with jute and wire-armor over the lead. It can be suspended from the top of the shaft by means of the wire armor without additional support, but, where possible, good practice requires a support every 25 or 50 ft., depending on length of run. See Fig. 9.

Tall Buildings require high voltage lines run vertically to a substation about half-way up. The height may be 200 to 300 ft. Varnished-cambric cable or a combination rubber and varnished-cambric cable without plastic fillers is recommended.

Parkway Cable Installations, where cable is run in the ground without ducts, is a common use for varnished-cambric cable, when furnished with a lead sheath and with flat-band armor and jute or jute finish over lead.

Transformer Leads.—Varnished-cambric cable is recommended for incoming lines to transformers of all types. It is equally suitable for connections to motors and generators. Single-conductor or multiple-conductor cable of this type is available for circuits of any voltage up to 26 kv., leaded or shielded, or 7.5 kv. non-shielded. Varnished-cambric cable is not injured by oil and therefore it can be run directly into oil switches and oil-filled transformers. Rubber insulation under these conditions would not last.

Rubber-insulated Cables are used widely for all types of exterior and interior wiring. They are available braided and leaded, or armored. Rubber-insulated armored cable can be applied without ducts to general use, power, lighting and control circuits, street-lighting circuits in outlying sections, lighting and power circuits in parks, estates, industrial plants and on bridges, connecting residences to mains, railroad yard and airport lighting, circuits for lighting of baseball and amusement parks, railway signal circuits, crossings under lakes and streams, horizontal runs in mines for power, light and telephone circuits. Rubber-insulated armored cable is resistant to weather conditions, is not affected by frost, and can be buried in the ground from 12 to 18 in. below surface.

Rubber-insulated cable is recommended for portable use with mining machinery, and locomotives, for supplying power to shovels, dredges, welding sets, and for equipment used in quarries, gravel pits, shipyards, and for logging operations. For portable applications, and where a strong, flexible cable is required, cable insulated with tellurium rubber compound is recommended.

Rubber non-metallic cables are used in ducts or aerial applications for low and medium voltage distribution circuits. For overhead line construction in residential districts where lines must be run through tree foliage, and tree cutting minimized, tree wire is used.

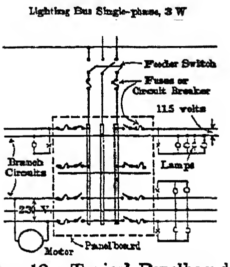


Fig. 10. Typical Panelboard Connections for Lighting and Small Motors

Non-ignitible Cable is designed for use as field rheostat leads and similar low voltage systems where non-combustible covering is desired.

DISTRIBUTION WITHIN BUILDINGS usually will start with one or more main service feeders from the main switchboard, depending on required reliability of service. Depending on the size of load, short circuit duty and voltage, the main feeder may be controlled by: *a*, fused knife switch; *b*, manually-operated air circuit breaker. For lighting and small power loads, fused knife switches are suitable for service switches. Air circuit breakers are more expensive, but are more accurate in operating overloads and short circuits, and do not require re-fusing. Main feeders terminate in motor control devices or in panelboards which serve branch circuits to lights, small motors and heating units. Usually one panelboard controls branch circuits feeding the same type of load. Panelboards usually are metal cabinets with hinged covers, which can be installed flush with the wall in finished buildings. The cabinet contains copper buses to which the main feeder is connected, and outlets and fused knife switches or circuit breakers for each branch circuit. Fig. 10 shows typical panelboard wiring. The remainder of the distribution system consists of conductors for branch circuits and connections between panelboards and feeders. Codes require either armored wire and cable or wires run in metal or fiber conduits. Conduits or armored cable can be run inside walls, in concrete floors, or overhead on beams. Rigid steel conduit is widely used in wiring systems.

Conduit Size for wires depends on number and sizes of wires to be drawn in the conduit without stretching of the conductor or damage to the insulation. Table 11 shows recommended conduit sizes for wires.

Capacity and Sizes of Wire or Cable for feeder or branch circuits depends on current to be carried and type of insulation. Heating of conductor and voltage drop are the limiting factors. Heating largely depends on conditions surrounding the conductor as they affect transfer of heat away from it. The National Board of Fire Underwriters publish tables showing allowable capacities of conductors from which the proper size can be determined. See Table 6.

Table 3.—Wire Gages

Gage No.	American Wire Gage B. & S.		Steel Wire Gage (Washburn & Moen)		Birmingham Wire Gage (Stubs' Iron)		Old English Wire Gage (London)		Stubs' Steel Wire Gage		British Standard Wire Gage		U. S. Std. Sheet Gage	Gage No.
	Diam., in.	Diam., mm.	Diam., in.	Diam., mm.	Diam., in.	Diam., mm.	Diam., in.	Diam., mm.	Diam., in.	Diam., mm.	Diam., in.	Diam., mm.	Diam., in.	
0 000 000	.....	.....	0.4900	12.4	.....	.....	.....	.....	.....	.....	0.500	12.7	0.5000	0 000 000
000 000	.....	.....	.4615	11.7	.....	.....	.....	.....	.....	.....	.464	11.8	.4687	000 000
00 000	.....	.....	.4305	10.9	.....	.....	.....	.....	.....	.....	.432	11.0	.4375	00 000
0 000	0.460	11.7	.3938	10.0	0.454	11.5	0.454	11.5	.....	.....	.400	10.2	.4062	0 000
000	.410	10.4	.3625	9.2	.425	10.8	.425	10.8	.....	.....	.372	9.4	.3750	000
00	.365	9.3	.3310	8.4	.380	9.7	.380	9.7	.....	.....	.348	8.8	.3437	00
0	.325	8.3	.3065	7.8	.340	8.6	.340	8.6	.....	.....	.324	8.2	.3125	0
1	.289	7.3	.2830	7.2	.300	7.6	.300	7.6	0.227	5.77	.300	7.6	.2812	1
2	.258	6.5	.2625	6.7	.284	7.2	.284	7.2	.219	5.56	.276	7.0	.2656	2
3	.229	5.8	.2437	6.2	.259	6.6	.259	6.6	.212	5.38	.252	6.4	.2500	3
4	.204	5.2	.2253	5.7	.238	6.0	.238	6.0	.207	5.26	.232	5.9	.2344	4
5	182	4.6	.2070	5.3	.220	5.6	.220	5.6	.204	5.18	.212	5.4	.2187	5
6	.162	4.1	.1920	4.9	.203	5.2	.203	5.2	.201	5.11	.192	4.9	.2035	6
7	.144	3.7	.1770	4.5	.180	4.6	.180	4.6	.199	5.05	.176	4.5	.1875	7
8	.128	3.3	.1620	4.1	.165	4.2	.165	4.2	.197	5.00	.160	4.1	.1719	8
9	.114	2.91	.1483	3.77	.148	3.76	.148	3.76	.194	4.93	.144	3.66	.1562	9
10	.102	2.59	.1350	3.43	.134	3.40	.134	3.40	.191	4.85	.128	3.25	.1406	10
11	.091	2.30	.1205	3.06	.120	3.05	.120	3.05	.188	4.78	.116	2.95	.1250	11
12	.081	2.05	.1055	2.68	.109	2.77	.109	2.77	.185	4.70	.104	2.64	.1094	12
13	.072	1.83	.0915	2.32	.095	2.41	.095	2.41	.182	4.62	.092	2.34	.0937	13
14	.064	1.63	.0800	2.03	.083	2.11	.083	2.11	.180	4.57	.080	2.03	.0781	14
15	.057	1.45	.0720	1.83	.072	1.83	.072	1.83	.178	4.52	.072	1.83	.0703	15
16	.051	1.29	.0625	1.59	.065	1.65	.065	1.65	.175	4.45	.064	1.63	.0625	16
17	.045	1.15	.0540	1.37	.058	1.47	.058	1.47	.172	4.37	.056	1.42	.0562	17
18	.040	1.02	.0475	1.21	.049	1.24	.049	1.24	.168	4.27	.048	1.22	.0500	18
19	.036	0.91	.0410	1.04	.042	1.07	.040	1.02	.164	4.17	.040	1.02	.0437	19
20	.032	.81	.0348	0.88	.035	0.89	.035	0.89	.161	4.09	.036	0.91	.0375	20
21	.0285	.72	.0317	.81	.032	.81	.0315	.80	.157	3.99	.032	.81	.0344	21
22	.0253	.64	.0286	.73	.028	.71	.0295	.75	.155	3.94	.028	.71	.0312	22
23	.0226	.57	.0258	.66	.025	.64	.0270	.69	.153	3.89	.024	.61	.0281	23
24	.0201	.51	.0230	.58	.022	.56	.0250	.64	.151	3.84	.022	.56	.0250	24
25	.0179	.45	.0204	.52	.020	.51	.0230	.58	.148	3.76	.020	.51	.0219	25
26	.0159	.40	.0181	.46	.018	.46	.0205	.52	.146	3.71	.018	.46	.0187	26
27	.0142	.36	.0173	.439	.016	.41	.01875	.48	.143	3.63	.0164	.42	.0172	27
28	.0126	.32	.0162	.411	.014	.36	.01650	.42	.139	3.53	.0148	.38	.0156	28
29	.0113	.29	.0150	.381	.013	.330	.01550	.394	.134	3.40	.0136	.345	.0141	29
30	.0100	.25	.0140	.356	.012	.305	.01375	.349	.127	3.23	.0124	.315	.0125	30
31	.0089	.227	.0132	.335	.010	.254	.01225	.311	.120	3.05	.0116	.295	.0109	31
32	.0080	.202	.0128	.325	.009	.229	.01125	.286	.115	2.92	.0108	.274	.0101	32
33	.0071	.180	.0118	.300	.008	.203	.01025	.260	.112	2.84	.0100	.254	.0094	33
34	.0063	.160	.0104	.264	.007	.178	.00950	.241	.110	2.79	.0092	.234	.0086	34
35	.0056	.143	.0095	.241	.005	.127	.00900	.229	.108	2.74	.0084	.213	.0078	35
36	.0050	.127	.0090	.229	.004	.102	.00750	.191	.106	2.69	.0076	.193	.0070	36
37	.0045	.113	.0085	.216	.....	.....	.00650	.165	.103	2.62	.0068	.173	.0066	37
38	.0040	.101	.0080	.203	.....	.....	.00575	.146	.101	2.57	.0060	.152	.0062	38
39	.0035	.090	.0075	.191	.....	.....	.00500	.127	.099	2.51	.0052	.132	.....	39
40	.0031	.080	.0070	.178	.....	.....	.00450	.114	.097	2.46	.0048	.122	.....	40
41	.....	.....	.0066	.168	.....	.....	.....	.....	.095	2.41	.0044	.112	.....	41
42	.....	.....	.0062	.157	.....	.....	.....	.....	.092	2.34	.0040	.102	.....	42
43	.....	.....	.0060	.152	.....	.....	.....	.....	.088	2.24	.0036	.091	.....	43
44	.....	.....	.0058	.147	.....	.....	.....	.....	.085	2.16	.0032	.081	.....	44
45	.....	.....	.0055	.140	.....	.....	.....	.....	.081	2.06	.0028	.071	.....	45
46	.....	.....	.0052	.132	.....	.....	.....	.....	.079	2.01	.0024	.061	.....	46
47	.....	.....	.0050	.127	.....	.....	.....	.....	.077	1.96	.0020	.051	.....	47
48	.....	.....	.0048	.122	.....	.....	.....	.....	.075	1.90	.0016	.041	.....	48
49	.....	.....	.0046	.117	.....	.....	.....	.....	.072	1.83	.0012	.030	.....	49
50	.....	.....	.0044	.112	.....	.....	.....	.....	.069	1.75	.0010	.025	.....	50

Selection of Wire Size and Fuses for motor branch circuits can be made from Table 7.
Relation between Load, Distance, Loss and Size of Conductor is given in Table 10.

DISTRIBUTION FOR LIGHTING depends on the required intensity and the type of control, which further depends on the service, office or factory, and whether individual or group switching of the lights is required. Usual lighting distribution systems are: 115-volt, single-phase, 2-wire, A.C. or D.C.; 115/230-volt, single-phase, 3-wire, A.C. or D.C.;

Table 4.—Dimensions and Resistance of Copper Wire and Cable

U.S. Bureau of Standards

American Wire Gage (A.W.G.)						Bare concentric cables of standard annealed copper							
Gage No. A.W.G.	Diam., mils	Cross Section		Ohms per 1000 ft. at 25° C. (77° F.)*	Lb. per 1000 ft.	Size, 1000 cir. mils or A.W.G.	Ohms per 1000 ft. at 25° C. (77° F.) †	Lb. per 1000 ft.†	Standard Strands		Flexible Strands		Outside Diam., mils
		Cir. mils	Sq. in.						No. of Wires	Diam. of Wires, mils	No. of Wires	Diam. of Wires, mils	
0000	460	212,000	0.166	0.0500	641	2000	0.00539	6180	127	125.5	169	108.8	1632
000	410	168,000	.132	.0630	508	1750	.00616	5410	127	117.4	169	101.8	1527
00	365	133,000	.105	.0795	403	1500	.00719	4630	91	128.4	127	108.7	1413
0	325	106,000	.0829	.100	319	1250	.00863	3860	91	117.2	127	99.2	1289
1	289	83,700	.0657	.126	253	1000	.0108	3090	61	128.0	91	104.8	1153
2	258	66,400	.0521	.159	201	950	.0114	2930	61	124.8	91	102.2	1124
3	229	52,600	.0413	.201	159	900	.0120	2780	61	121.5	91	99.4	1094
4	204	41,700	.0328	.253	126	850	.0127	2620	61	118.0	91	96.6	1063
5	182	33,100	.0260	.320	100	800	.0135	2470	61	114.5	91	93.8	1031
6	162	26,300	.0206	.403	79.5	750	.0144	2320	61	110.9	91	90.8	999
7	144	20,800	.0164	.508	63.0	700	.0154	2160	61	107.1	91	87.7	965
8	128	16,500	.0130	.641	50.0	650	.0166	2010	61	103.2	91	84.5	930
9	114	13,100	.0103	.808	39.6	600	.0180	1850	61	99.2	91	81.2	893
10	102	10,400	.00815	1.02	31.4	550	.0196	1700	61	95.0	91	77.7	855
11	91	8,230	.00647	1.28	24.9	500	.0216	1540	37	116.2	61	90.5	815
12	81	6,530	.00513	1.62	19.8	450	.0240	1390	37	110.3	61	85.9	773
13	72	5,180	.00407	2.04	15.7	400	.0270	1240	37	104.0	61	81.0	729
14	64	4,110	.00323	2.58	12.4	350	.0308	1080	37	97.3	61	75.7	682
15	57	3,260	.00256	3.25	9.86	300	.0360	926	37	90.0	61	70.1	631
16	51	2,580	.00203	4.09	7.82	250	.0432	772	37	82.2	61	64.0	576
17	45	2,050	.00161	5.16	6.20	0000	.0510	653	19	105.5	37	75.6	533
18	40	1,620	.00128	6.51	4.92	000	.0643	518	19	94.0	37	67.3	471
19	36	1,290	.00101	8.21	3.90	00	.0811	411	19	83.7	37	60.0	420
20	32	1,020	.000802	10.4	3.09	0	.102	326	19	74.5	37	53.4	374
21	28.5	810	.000636	13.1	2.45	1	.129	258	19	66.4	37	47.6	333
22	25.3	642	.000505	16.5	1.94	2	.163	205	7	97.4	19	59.1	296
23	22.6	509	.000400	20.8	1.54	3	.205	163	7	86.7	19	52.6	263
24	20.1	404	.000317	26.2	1.22	4	.258	129	7	77.2	19	46.9	234
25	17.9	320	.000252	33.0	0.970	5	.326	102	7	68.8	19	41.7	209
26	15.9	254	.000200	41.6	.769	6	.411	81	7	61.2	19	37.2	186
27	14.2	202	.000158	52.5	.610	7	.518	64.3	7	54.5	19	33.1	166
28	12.6	160	.000126	66.2	.484	8	.653	51	7	48.6	19	29.5	147
29	11.3	127	.0000995	83.5	.384	10	1.039	32	7	38.5	19	23.4	117
30	10.0	101.0	.0000789	105	.304	12	1.652	20	7	30.5	19	18.5	92
31	8.9	79.7	.0000626	133	.241	14	2.626	12.7	7	24.2	19	14.7	73
32	8.0	63.2	.0000496	167	.191	16	4.176	8	7	19.2	19	11.7	58
33	7.1	50.1	.0000394	211	.152	.....	.....	....	...	....	...	....	....
34	6.3	39.8	.0000312	266	.120	.....	.....	....	...	....	...	....	....
35	5.6	31.5	.0000248	336	.0954	.....	.....	....	...	....	...	....	....
36	5.0	25.0	.0000196	423	.0757	.....	.....	....	...	....	...	....	....
37	4.5	19.8	.0000156	533	.0600	.....	.....	....	...	....	...	....	....
38	4.0	15.7	.0000123	673	.0476	.....	.....	....	...	....	...	....	....
39	3.5	12.5	.0000098	848	.0377	.....	.....	....	...	....	...	....	....
40	3.1	9.9	.0000078	1070	.0299	.....	.....	....	...	....	...	....	....

* Values are only for annealed copper of standard resistivity. Hard drawn copper may be taken as about 2.7% higher resistivity than annealed copper.

† The values are 2% greater than for a solid rod of cross section equal to the total cross section of the wires of the cable.

Resistivity of pure copper at 20° C. = 0.15328 ohm per meter.

Table 5.—Sizes and Weights of Copper Wire and Cable

Size		Rubber Insulation								Weather Proof Insulation							
		Double Braid				Triple Braid				Double Braid				Triple Braid			
		Solid		Stranded		Solid		Stranded		Solid		Stranded		Solid		Stranded	
A.W.G.	Cir. mils	Diam., in.	Lb. per 1000 ft.	Diam., in.	Lb. per 1000 ft.	Diam., in.	Lb. per 1000 ft.	Diam., in.	Lb. per 1000 ft.	Diam., in.	Lb. per 1000 ft.	Diam., in.	Lb. per 1000 ft.	Diam., in.	Lb. per 1000 ft.	Diam., in.	Lb. per 1000 ft.
....	1,000,000	....	.....	1.46	3553	....	.....	1.54	3637	....	.....	1.37	3456	....	.....	1.45	3674
....	900,000	....	.....	1.40	3223	....	.....	1.48	3304	....	.....	1.31	3127	....	.....	1.39	3332
....	800,000	....	.....	1.33	2891	....	.....	1.42	2968	....	.....	1.24	2799	....	.....	1.33	2992
....	700,000	....	.....	1.27	2557	....	.....	1.35	2631	....	.....	1.18	2471	....	.....	1.27	2650
....	600,000	....	.....	1.19	2220	....	.....	1.28	2290	....	.....	1.11	2093	....	.....	1.19	2235
....	500,000	....	.....	1.09	1842	....	.....	1.17	1906	....	.....	1.03	1765	....	.....	1.11	1894
....	400,000	....	.....	1.00	1514	....	.....	1.09	1573	....	.....	0.94	1436	....	.....	1.02	1553
....	300,000	....	.....	0.90	1173	....	.....	0.99	1226	....	.....	.85	1083	....	.....	0.93	1174
0000	211,600	0.70	793	.77	833	0.78	835	.85	879	0.61	723	.71	745	0.66	767	.79	800
000	167,805	.65	646	.71	675	.73	685	.79	719	.56	587	.65	604	.60	629	.73	653
00	133,079	.61	528	.66	556	.69	564	.74	595	.52	467	.60	482	.55	502	.66	522
0	105,592	.57	439	.61	457	.65	474	.70	494	.47	377	.56	388	.51	407	.61	424
1	83,695	.53	363	.57	377	.61	395	.66	412	.41	294	.47	303	.45	316	.52	328
2	66,373	.45	276	.50	293	.51	297	.59	324	.37	239	.42	246	.40	260	.44	270
3	52,634	.42	228	.45	238	.48	247	.52	260	.35	185	.38	190	.37	208	.41	219
4	41,743	.39	190	.42	198	.46	208	.49	218	.32	151	.35	155	.35	164	.38	170
5	33,102	.36	154	.40	166	.41	167	.46	184	.30	122	.32	126	.32	130	.35	146
6	26,251	.34	130	.36	136	.39	142	.41	149	.28	100	.31	103	.30	112	.33	115
7	20,817	.30	105	.32	108	....	.....	....	.....	....	.....	....	.....	....	.....	....	.....
8	16,510	.27	82.1	.29	85.5	....	.....	....	.....	....	.....	....	.....	....	.....	....	.....
9	13,904	.26	68.4	.27	70.0	....	.....	....	.....	....	.....	....	.....	....	.....	....	.....
10	10,382	.25	58.1	.26	60.6	....	.....	....	.....	....	.....	....	.....	....	.....	....	.....
11	8,234	.24	50.0	.25	52.3	....	.....	....	.....	....	.....	....	.....	....	.....	....	.....
12	6,230	.23	43.1	.24	44.9	....	.....	....	.....	....	.....	....	.....	....	.....	....	.....
13	5,178	.22	38.5	.23	39.6	....	.....	....	.....	....	.....	....	.....	....	.....	....	.....
14	4,107	.21	33.0	.22	34.3	....	.....	....	.....	....	.....	....	.....	....	.....	....	.....

Table 6.—Allowable Current Carrying Capacities of Solid and Stranded Wires

(National Electrical Code)

Solid Wire					Stranded Wire					
		Carrying Capacities, amperes						Carrying Capacities, amperes		
A.W.G.	Area, cir. mils	Rubber Insulation	Varnished Cloth Insulation	Other Insulation and Bare	No. of Strands	A.W.G.	Area, cir. mils	Rubber Insulation	Varnished Cloth Insulation	Other Insulation
6	26,250	50	60	70	7	22	4,490	15	18	20
5	33,100	55	65	80	7	20	7,150	20	25	25
4	41,740	70	85	90	7	18	11,370	25	30	35
3	52,630	80	95	100	7	16	18,080	35	40	50
2	66,370	90	110	125	7	14	28,740	50	60	70
1	83,690	100	120	150	7	12	45,710	70	85	90
0	105,500	125	150	200	7	11	58,000	80	95	110
00	133,100	150	180	225	7	10	72,680	90	110	130
000	167,800	175	210	275	19	14	78,030	100	120	150
0000	211,600	225	270	325	19	13	98,380	125	150	175
	300,000	275	330	400	19	12	124,900	150	180	210
	400,000	325	390	500	19	11	157,300	175	210	250
	500,000	400	480	600	19	9	248,700	250	300	350
	600,000	450	540	680	37	11	360,400	275	330	400
	700,000	500	600	760	37	10	381,200	325	390	500
	800,000	550	660	840	61	10	633,300	475	565	700
	900,000	600	720	920	61	9	798,300	550	660	825
	1,000,000	650	780	1000	61	8	1,007,000	650	780	1000
	1,200,000	730	880	1150	91	9	1,191,000	725	870	1125
	1,400,000	810	970	1290	91	8	1,502,000	850	1020	1350
	1,600,000	890	1070	1430	127	9	1,660,000	900	1100	1460
	1,800,000	970	1160	1550	127	8	2,097,000	1100	1300	1700
	2,000,000	1050	1260	1670						

208/120-volt, 3-phase, 4-wire, A.C.; high-voltage with transformers at lighting centers. With the latter, small air-cooled lighting transformers would be used, supplying branch circuits from 440- or 220-volt primaries to 230/115-volt, single-phase, 3-wire secondaries. Usually this scheme is economical only for large areas.

Table 7.—Wire and Fuse Sizes for Motor Branch Circuits

	Min. size of copper wire, A.W.G. or cir. mils — Insulation			Running protection of motors	Max. allowable rating of branch-circuit fuses, amperes*		
					A	C	
1.25	3	4	5	80	200	150	100
2.50	2	4	4	85	225	150	110
3.75	2	3	4	90	225	150	110
5.0	1	3	3	95	250	175	125
6.25	1	2	3	100	250	175	125
7.50	0	2	2	105	250	175	150
10.0	0	2	2	110	300	200	150
12.50	0	1	2	115	300	200	150
15.0	0	1	2	120	300	200	150
17.5	0	0	2	125	300	200	150
20.0	00	0	1	137.5	350	225	175
22.5	00	0	1	150	400	250	200
25.0	000	00	0	162.5	400	300	200
30.0	000	00	0	175	450	300	225
35.0	200,000	000	0	187.5	450	300	225
40.0	200,000	000	0	200	500	350	250
45.0	0000	200,000	00	213	500	350	300
50.0	0000	200,000	00	225	600	400	300
55.0	250,000	200,000	000	238	600	400	300
60.0	250,000	0000	000	250	600	400	300
65.0	300,000	250,000	000	275	...	450	350
70.0	350,000	250,000	200,000	300	...	500	400
75.0							

* Motors indicated by column designations are as follows: A, Squirrel-cage, full voltage starting; single-phase repulsion or split phase. B, Squirrel-cage, reduced voltage starting; high-reactance squirrel-cage (up to 30 amp.). C, Squirrel-cage, reduced voltage starting; high-reactance squirrel-cage (above 30 amp.). D, Wound rotor, A.C. and D.C.

Table 8.—Dimensions and Current-carrying Capacity of Copper Tubing*

Pipe Size, in.	Approximate Diameter, in. Outside	Approximate Diameter, in. Inside	Exact Diameter, in. Outside	Exact Diameter, in. Inside	Lb. per ft.	Area, cir. mils	Carrying Capacity, amperes at cir. mils per amp. 1000	1200	1500	1600
					Standard					
1/2	13/16	5/8	0.840	0.625	0.95	317,471	317	265	212	198
3/4	1 1/16	53/64	1.050	.822	1.31	423,524	424	353	282	265
1	1 5/16	1 3/64	1.315	1.062	1.79	633,016	633	528	422	396
1 1/4	1 5/8	1 3/8	1.660	1.368	2.63	851,200	851	709	568	532
1 1/2	1 7/8	1 39/64	1.900	1.600	3.15	1,017,900	1018	848	679	636
2	2 3/8	2 1/16	2.375	2.062	4.20	1,368,136	1368	1140	912	855
2 1/2	2 7/8	2 15/32	2.875	2.500	6.04	2,179,536	2180	1816	1453	1362
3	3 1/2	3 1/16	3.500	3.062	8.72	2,849,644	2850	2375	1893	1781
3 1/2	4	3 1/2	4.000	3.500	11.45	3,411,696	3412	2843	2274	2132
4	4 1/2	4 1/32	4.500	4.000	13.33	4,041,324	4041	3368	2694	2526
					Extra Heavy					
1/2	......	......	0.840	0.542	1.330	411,834	412	343	275	257
3/4	......	......	1.050	.736	1.750	560,804	561	467	374	351
1	......	......	1.315	.951	2.478	824,824	825	687	450	516
1 1/4	......	......	1.660	1.272	3.465	1,137,616	1138	948	758	711
1 1/2	......	......	1.900	1.494	4.462	1,377,964	1378	1148	919	861
2	......	......	2.375	1.933	5.733	1,904,136	1904	1587	1269	1190
2 1/2	......	......	2.875	2.315	8.715	2,906,400	2906	2422	1938	1816
3	......	......	3.500	2.892	11.760	3,886,336	3886	3239	2591	2429
3 1/2	......	......	4.000	3.358	14.385	4,723,836	4724	3936	3149	2954
4	......	......	4.500	3.818	17.325	5,672,876	5673	4727	3784	3546

* Drawn to correspond to outside diameter of iron pipe sizes.

Table 9.—Sizes, Weights and Current-carrying Capacity of Rectangular Copper Bars

Bar Size, in.	tion, sq. in.	Wt. per ft., lb	Current Density, amperes per sq. in. 750	1000	Bar Size,	tion,		Current Density, amperes per sq. in. 750 | 1000 Current carrying capacity, amperes	Bar Size,	Cross-section, sq. in.		Current Density, amperes per sq. in. 750	1000
			Current carrying capacity, amperes									Current carrying capacity, amperes	
2 ×	0.250	0.962			2 × 1/4	0.500	1.925	500	× 3/8	0.750	2.89	563	750
2 1/4 × 1/8	.281	1.080			2 1/4 × 1/4	.562	2.165	562	2 1/4 × 3/8	.842	3.24	632	842
2 1/2 × 1/8	.313	1.205			2 1/2 × 1/4	.625	2.41	625	2 1/2 × 3/8	.938	3.61	703	938
2 3/4 × 1/8	.344	1.324			2 3/4 × 1/4	.688	2.65	688	2 3/4 × 3/8	1.030	3.97	772	1030
3 × 1/	.375	1.444			3 × 1/4	.750	2.89	750	3 × 3/8	1.125	4.33	844	1125
3 1/2 × 1/8	.437	1.680			3 1/2 × 1/4	.875	3.37	875	3 1/2 × 3/8	1.314	5.06	985	1314
4 × 1/	.500	1.9			4 × 1/	1.000	3.85	1000	4 × 3/8	1.500	5.77	1125	1498

Table 10.—Relation of Load, Distance, Loss, and Conductor Size of 2-wire Circuits

Wire Size, A.W.G. 110-volt circuit	220-volt circuit	Line Loss in Percentage of Rated Voltage. Power Loss in Percentage of Delivered Power	1.5				5			10
		Ampere-feet = (Amperes × length of one wire)								
	0000	21,550	32,325	43,100	64,650	86,200	107,750	129,300	172,400	215,500
	000	17,080	25,620	34,160	51,240	68,320	85,400	102,480	136,640	170,800
	00	13,550	20,325	27,100	40,650	54,200	67,750	81,300	108,400	135,500
0000	0	10,750	16,125	21,500	32,250	43,000	53,750	64,500	86,000	107,500
000	1	8,520	12,780	17,040	25,560	34,080	42,600	51,120	68,160	85,200
00	2	6,750	10,140	13,520	20,280	27,040	33,800	40,560	54,080	67,600
0	3	5,360	8,040	10,720	16,080	21,440	26,800	32,160	42,880	53,600
1	4	4,250	6,375	8,500	12,750	17,000	21,250	25,500	34,000	42,500
2	5	3,370	5,055	6,740	10,110	13,480	16,850	20,220	26,960	33,700
3	6	2,670	4,005	5,340	8,010	10,680	13,350	16,020	21,360	26,700
4		2,120	3,180	4,240	6,360	8,480	10,600	12,720	16,960	21,200
5		1,680	2,520	3,360	5,040	6,720	8,400	10,800	13,440	16,800
6		1,330	1,995	2,660	3,990	5,320	6,650	7,980	10,640	13,300
7		1,055	1,582	2,110	3,165	4,220	5,275	6,330	8,440	10,550
8		838	1,257	1,675	2,514	3,350	4,190	5,028	6,700	8,380
9		665	997	1,330	1,995	2,660	3,320	3,990	5,320	6,650
10		527	790	1,054	1,580	2,108	2,635	3,160	4,215	5,270
11		418	627	836	1,254	1,672	2,090	2,508	3,344	4,180
12		332	498	665	997	1,330	1,660	1,995	2,660	3,325
14		209	313	418	627	836	1,045	1,354	1,672	2,090

Table 11.—Size of Conduits for 2-wire and 3-wire Systems

National Electrical Code

Wire Size, A.W.G.	Number of Wires in One Conduit 1	2	3	4	5	6	7		
	Minimum Size of Conduit, in.								
14		1/2	1/2	1/2	3/4	3/4			1
12		1/2	1/2	3/4	3/4	1	1		1 1/4
10		3/4		3/4	1	1	1 1/4	1 1/4	1 1/4
8		3/4	1	1	1 1/4	1 1/4	1 1/4	1	1 1/2
6		1	1 1/4	1 1/4	1 1/2	1 1/2	2	2	2
5		1 1/4	1 1/4	1 1/4	1 1/2	2	2	2	2
4		1 1/4	1 1/4	1 1/2	2	2	2	2	2 1/2
3		1 1/4	1 1/4	1 1/2	2	2	2	2 1/2	2 1/2
2		1 1/4	1 1/2	1 1/2	2	2	2 1/2	2 1/2	2 1/2
1		1 1/2	1 1/2	2	2	2 1/2	2 1/2	3	3
0		1 1/2	2	2	2 1/2	2 1/2	3	3	3
00		2	2	2 1/	2 1/2	3	3	3	3 1/2
000	1	2	2	2 1/	3	3	3	3 1/	3 1/2
0000	1 1/4	2	2 1/	2 1/2	3	3	3 1/2	3 1/	4

Wire Size, cir. mils	Number of Wires in One Conduit 1	2		4		
	Minimum Size of Conduit, in.					
200,000	1 1/4		2 1/2	2 1/2	3	3
300,000	1 1/4	2 1/2	3	3	3 1/2	3 1/2
400,000	1 1/4	3	3	3 1/2	4	4
500,000	1 1/2	3	3	3 1/2	4	4 1/2
600,000	2	3	3 1/2	4		5
700,000	2	3 1/2		4 1/2		
800,000	2	3 1/2		4 1/2		
900,000	2	3 1/2		4 1/2		
1,000,000	2	4		5		
1,200,000	2 1/2	4 1/2	4 1/2	6		
1,400,000	2 1/2	4 1/2	5	6		
1,600,000	2 1/2	5	5	6		
1,800,000	3	5	6	6		
2,000,000	3	5				

Conduit size, in.	1/2	3/4	1	1 1/4	1 1/2	2	2 1/2	3	3 1/2	4	4 1/2
Weight per 100 ft., lb.	85.2	113.	168.4	228.1	273.1	367.8	581.9	761.6	920.2	1088.9	1264.2
Weight of 1 elbow, lb.		1.2	2	3	4.2	7	13	17	23	27	31
Weight of 1 coupling, lb.		0.2	0.3	0.5	0.7	1.2	1.7	2.4	4.2	4.7	5.5

WIRING SYSTEMS.—Common wiring systems are shown in Fig. 11. Three-wire D.C. or single-phase A.C. branch lighting circuits are taken off from either line and neutral and motors are operated from the two lines. These systems are used mainly for residences, small stores or other small domestic or commercial loads. The 3-phase, 3-wire and the 3-phase, 4-wire systems are used for large stores, offices, public buildings and factories. In 3-phase, 3-wire systems, the single-phase lighting load should be kept very nearly balanced on the three phases. For large lighting loads, 3-phase, 4-wire system is preferable where single-phase loads are taken off from line to neutral. The 2-phase, 4-wire and the 2-phase, 3-wire systems are not widely used. Almost all wiring is now single-phase, 2- or 3-wire, and 3-phase 3- or 4-wire systems. The 2-phase, 3- or 4-wire systems are important only in existing installations. The voltages indicated are those to be maintained at the loads, and, assuming the same load for each system, the current in the feeder lines of the different systems is indicated in terms of the current I of the 2-wire D.C. or single-phase A.C. system.

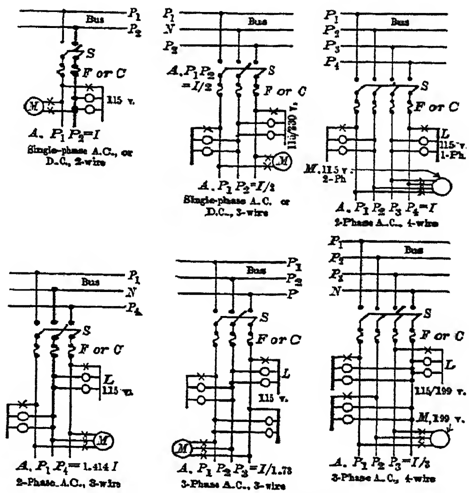


FIG. 11. Wiring Systems

Table 12.—Wire Sizes for Distance Voltage Drop

Capacity Amperes	Distance in Feet to Center of Distribution																	
	20	30	40	50	60	70	80	90	100	120	140	160	180	200	240	280	320	360
	Wire Size, A.W.G., for 2% Loss on 110 volts *																	
1	....	....	....	....	....	....	....	....	....	....	....	....	....	....	16	15	15	14
1.5	....	....	....	....	....	....	....	....	....	....	....	16	15	15	14	14	13	12
2	....	....	....	....	....	....	....	....	....	16	15	15	14	14	13	12	12	11
3	....	....	....	....	....	....	16	15	15	14	14	13	12	12	11	11	10	9
4	....	....	....	....	16	15	15	14	14	13	12	12	11	11	10	9	9	8
5	....	....	....	16	15	14	14	13	13	12	11	11	10	10	9	8	8	7
6	....	....	16	15	14	14	13	12	12	11	11	10	9	9	8	8	7	7
7	....	16	15	14	14	13	12	12	11	11	10	9	9	8	7	7	6	6
8	....	16	15	14	13	12	12	11	11	10	9	9	8	8	7	7	6	5
9	....	15	14	13	12	12	11	11	10	9	9	8	8	7	7	6	5	5
10	16	15	14	13	12	11	11	10	10	9	8	8	7	7	6	5	5	4
12	16	14	13	12	11	11	10	9	9	8	8	7	7	6	5	5	4	4
14	15	14	12	11	11	10	9	9	8	7	7	6	6	5	5	4	3	3
16	15	13	12	11	10	9	9	8	8	7	7	6	5	5	4	3	3	2
18	14	12	11	10	9	9	8	8	7	7	6	5	5	4	4	3	2	2
20	14	12	11	10	9	8	8	7	7	6	5	5	4	4	3	2	2	1
25	13	11	10	9	8	7	7	6	6	5	4	4	3	3	2	1	1	0
30	12	10	9	8	7	7	6	6	5	4	4	3	3	2	1	1	0	0
35	11	10	8	7	7	6	5	5	4	4	3	2	2	1	1	0	00	00
40	11	9	8	7	6	5	5	4	4	3	2	2	1	1	0	00	00	000
45	10	9	7	6	6	5	4	4	3	3	2	1	1	0	00	00	000	000
50	10	8	7	6	5	4	4	3	3	2	1	1	0	0	00	000	000	0000
60	9	7	6	5	4	4	3	3	2	1	1	0	0	00	000	000	0000	0000
70	8	7	5	4	4	3	2	2	1	1	0	00	00	000	000	0000	0000	....
80	8	6	5	4	3	2	2	1	1	0	00	00	000	000	0000	0000	....	....
90	7	6	4	3	3	2	1	1	0	00	00	000	000	0000	0000	....	....	....
100	7	5	4	3	2	1	1	0	0	00	000	000	0000	0000	....	....	....	....
120	6	4	3	2	1	1	0	0	00	00	000	0000	0000	....	....	....	....	....

* To determine wire size for a 2% drop at 220 volts divide the amperes at 220 volts by 2, and find the wire size for the corresponding distance at 110 volts. EXAMPLE.—Required the wire size for 70 amperes at 220 volts for a drop of 2% in a distance of 160 ft. 70/2 = 35. From the table the wire size required is No. 2.

For a given load and a given line loss the size of copper required will vary inversely as the square of the voltage. Distribution at 230-volts will require 1/4 the wire cross-section as for the same system operating at 115-volts. Table 12 gives wire sizes for a given voltage drop for various lengths of line.

5. CIRCUIT PROTECTION

GENERAL CONSIDERATIONS INVOLVED in circuit protection are: Safety to human life, insurance of circuit and connected apparatus against damage, fire hazard, and protection of service. Preventive measures minimize the possibility of circuit failures, and wiring, materials, and construction should be of high grade and approved standards. Danger to life by contact with live parts, and indirectly through fires and explosions, can be minimized by proper precautions. Circuits should be sufficiently protected by enclosures and insulation to prevent accidental contact. Grounding of circuits to limit voltage above ground will minimize hazard to life. Code requirements limit voltages above ground to 150 volts.

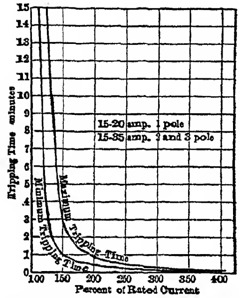


FIG. 12. Time-current Characteristics of Air Circuit Breaker

Over-current and short circuit protection, in the form of fuses and circuit breakers, prevents excessive damage to circuits and equipment under abnormal conditions. They protect the circuit by automatically disconnecting it from the source of power. Over-current protection prevents overheating of the conductors and destruction of insulation. Short circuits must be cleared instantly, to minimize damage to the circuit itself and the possibility of fire at the point of short circuit.

For process where power outages are costly, service protection is important. Duplicate feeders and automatic throwover to a standby power source effect continuity of service. The circuit protective device often sounds an alarm instead of automatically disconnecting the circuit.

Important factors involved in the selection of a circuit interrupting device are: Reliability, over-current and short circuit capacities, normal current and voltage rating, initial cost and maintenance. Fuses afford a high degree of reliability if properly applied. They are low in initial cost and maintenance consists chiefly of replacement. On a circuit subject to frequent overload, fuse replacements may be an expensive item. A manually-operated air circuit breaker of the type designed for use as an alternative to fuses on low voltage circuits may be the proper solution.

FUSES.—A fuse will open a circuit on excess current, with delayed action inversely as the magnitude of the current. Fuses are used for protection of low-capacity circuits of 600 volts and below. The maximum standard rating is 600 amperes, which limits the application of standard fuses to 360 kva. single-phase, 623 kva. 3-phase and 720 kva. 2-phase, 4-wire circuits. Tables 13 and 14 give data on fuses and wiring for various motor circuits.

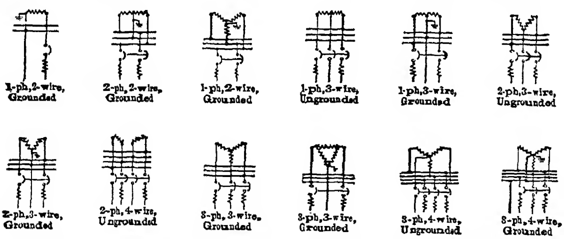


FIG. 13. Typical Circuit Breaker Arrangements for Different A.C. Circuits

AIR CIRCUIT BREAKERS are manually closed and are tripped on overload by a thermal device. They, with fuses, are the chief protective devices for low-voltage distribution circuits. Fig. 12 shows a typical time-current characteristic. Recommended ratings are given in Tables 15 and 16. These circuit breakers are replacing fuses and fused knife switches. Tripping elements are rated in accordance with the ratings of wires as established by the National Electric Code. These breakers are used with panelboards for branch circuits, at load centers, service entrances, and switchboards for the protection

of light and power circuits. They also will protect individual circuits and can be mounted in sheet metal enclosing cases for industrial applications. Their tripping time at all currents is longer than the clearing time of a fuse of the same rating. Arrangement of breakers for over-current protection of common circuits used are shown in Figs. 13 and 14. These will meet Underwriters' rules, except for incoming lines, where provision also must be made for disconnecting the grounded neutral. Table 17 specifies breakers required for direct-current machine circuits.

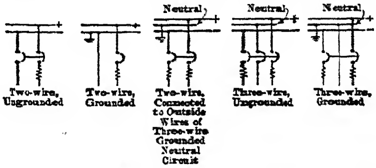


FIG. 14. Typical Circuit Breaker Arrangements for Different D.C. Circuits

Neutral Conductors.—Fuses or protective devices in the neutral conductor are not recommended. Opening the neutral would apply high voltage to the lightly-loaded side of a 3-wire system under unbalanced conditions. On 3-wire, single-phase circuits, each outer wire has over-current protection, and both outer wires must open to prevent overloading the neutral. The breaker should be double-pole with a current coil in each pole.

Table 13.—Wire, Fuse and Switch Data for A.C. Motors

Hp. of									apacity of Switch,							Minimu of W B.	
	Single-phase					Two-phase, 3-wire										
								110 Volts								
1/4	4	5	30	14	1/2	2	3	30	14	3/4	...	...	...	...	......	....
1/2	7.5	10	30	14	1/2	3.8	5	30	14	3/4	...	...	...	...	......	....
3/4	10	15	30	14	1/2	5.5	10	30	14	3/4	...	...	...	...	......	....
1	12.5	20	30	12	3/4	7	10	30	14	3/4	6	20	10	30	14	3/4
1 1/2	18	25	30	10	3/4	8.8	15	30	14	3/4	...	...	...	...	......	....
2	24	30	30	8	1	11	15	30	14	3/4	12	40	15	30	14	3/4
3	34	50	60	6	1 1/4	15.8	20	30	10	1	18	60	25	30	10	1
5	55	70	100	4	1 1/4	26.2	35	60	8	1	30	100	40	60	8	1
7 1/2	80	100	100	1	1 1/2	38.6	50	60	6	1 1/4	42	110	60	60	5	1 1/4
10	106	150	200	00	2	50.8	70	100	4	1 1/2	56	150	75	100	4	1 1/2
15	.....	..	..	..	...	72	100	100	2	1 1/2	84	250	125	200	0	2
20	.....	..	..	..	...	96	125	200	0	2	104	275	150	200	00	2 1/2
25	.....	..	..	..	...	120	150	200	00	2 1/2	...	...	...	...	......	....
30	.....	..	..	..	...	140	175	200	000	2 1/2	156	400	200	200	200,000*	2 1/2
35	.....	..	..	..	...	163	225	400	200,000*	2 1/2	...	...	...	...	......	....
40	.....	..	..	..	...	184	250	400	250,000*	3	...	...	...	...	......	...
50	.....	..	..	..	...	230	300	400	350,000*	3 1/2	...	...	...	...	......	...
60	.....	..	..	..	...	276	350	400	400,000*	3 1/2	...	...	...	...	......	...
75	....	..	..	..	...	345	450	600	600,000*	3 1/2	...	...	...	...	......	....
								220 Volts								
1/4	2	5	30	14	1/2	1	3	30	14	3/4	...	...	...	...	......	....
1/2	3.75	5	30	14	1/2	1.9	3	30	14	3/4	...	...	...	...	......	....
3/4	5	10	30	14	1/2	2.8	5	30	14	3/4	...	...	...	...	......	....
1	6.1	10	30	14	1/2	3.5	5	30	14	3/4	3	10	5	30	14	3/4
1 1/2	9	15	30	14	1/2	4.5	10	30	14	3/4	...	...	...	...	......	....
2	12	15	30	12	3/4	5.5	10	30	14	3/4	6	20	10	30	14	3/4
3	17	25	30	10	3/4	7.9	10	30	14	3/4	9	30	15	30	14	3/4
5	28	40	60	8	1	13.1	20	30	12	3/4	15	50	20	30	12	3/4
7 1/2	40	50	60	6	1 1/4	19.3	25	30	10	1	21	70	30	30	10	1
10	54	70	100	4	1 1/4	25.4	35	60	8	1	28	100	40	60	8	1
15	.....	..	..	..	...	36	50	60	6	1 1/4	42	110	60	100	5	1 1/4
20	.....	..	..	..	...	48	60	60	4	1 1/2	52	150	75	100	4	1 1/2
25	.....	..	..	..	...	60	75	100	3	1 1/2	...	...	...	...	......	....
30	.....	..	..	..	...	70	80	100	2	1 1/2	78	200	100	100	1	2
35	....	..	..	..	...	81	110	200	1	2	...	...	...	...	......	....
40	....	..	..	..	...	92	125	200	0	2	105	275	150	200	00	2 1/2
50	....	..	..	..	...	115	150	200	00	2 1/2	133	350	175	200	000	2 1/2
60	....	..	..	..	...	138	175	200	000	2 1/2	...	...	...	...	......	....
75	....	..	..	..	...	172	225	400	0000	2 1/2	184	500	250	400	250,000*	3
100	....	..	..	..	...	....	...	...	.....	.	245	600	350	400	400,000*	3 1/2

* Circular mils.

(*Table continued on following page*)

Grounding.—As shown in Figs. 13 and 14, the neutral wire of secondary distribution circuits are grounded. This limits the voltage to ground to a maximum of 150 volts, and minimizes danger to life. All parts of a grounding circuit must be continuous and of negligible resistance. In addition to limiting the voltage above ground on a secondary circuit, grounding also supplies protection against extraneous voltages which might appear on the circuit, as from a primary circuit accidentally coming in contact with the secondary circuit. Metal conduits and apparatus cases and frames also are grounded.

The actual connection to earth of a grounded system is of great importance. Code requirements suggest grounding to water pipes. Where no such ground connections exist, artificial grounds are

Table 13.—Wire, Fuse and Switch Data for A.C. Motors—*continued*

	Full load Ampere																...uit, in.
	Single-phase					Two-phase, 3 wire					Three-phase					
						440 Volts										
1/4	.....	..	..	..	...	0.5	3	30	14	3/4	...	...	...	...	......	....
1/2	.....	..	..	..	...	0.95	3	30	14	3/4	...	...	...	...	......	....
3/4	.....	..	..	..	...	1.35	3	30	14	3/4	...	...	...	...	......	....
1	.....	..	..	..	...	1.75	3	30	14	3/4	1.5	5	3	30	14	3/4
1 1/2	.....	..	..	..	...	2.25	3	30	14	3/4	...	...	...	...	......	....
2	.....	..	..	..	...	2.75	5	30	14	3/4	3	10	5	30	14	3/4
3	.....	..	..	..	...	4	5	30	14	3/4	4.5	15	10	30	14	3/4
5	.....	..	..	..	...	6.6	10	30	14	3/4	7.5	25	10	30	14	3/4
7 1/2	.....	..	..	..	...	9.7	15	30	14	3/4	10.5	35	15	30	14	3/4
10	.....	..	..	..	...	12.7	20	30	12	3/4	14	50	20	30	12	3/4
15	.....	..	..	..	...	18	25	30	10	1	21	70	30	30	10	1
20	.....	..	..	..	...	24	30	30	8	1	26	80	40	60	8	1
25	.....	..	..	..	...	30	40	60	6	1 1/4	...	...	...	...	......	....
30	.....	..	..	..	...	35	50	60	6	1 1/4	39	100	50	60	6	1 1/4
35	.....	..	..	..	...	40	50	60	6	1 1/4	...	...	...	...	......	....
40	.....	..	..	..	...	46	60	60	5	1 1/4	52.5	150	75	100	4	1 1/2
50	.....	..	..	..	...	57.5	75	100	3	1 1/2	66.5	175	100	100	2	1 1/2
60	.....	..	..	..	...	69	100	100	2	1 1/2	...	...	...	...	......	....
75	.....	..	..	..	...	86	110	200	0	2	92	250	125	200	0	2
100	.....	..	..	..	...	114	150	200	00	2 1/2	122.5	300	175	200	00	2 1/2
125	.....	..	..	..	...	139	175	200	000	2 1/2	...	...	...	...	......	....
150	.....	..	..	..	...	167	225	400	0000	2 1/2	184	450	250	400	250,000*	3
175	.....	..	..	..	...	195	250	400	250,000*	3	...	...	...	...	......	....
200	.....	..	..	..	...	224	300	400	300,000*	3	236	600	300	400	350,000*	3 1/2
						550 Volts										
1/4	.....	..	..	..	...	0.45	3	30	14	3/4	...	...	...	...	......	....
1/2	.....	..	..	..	...	.90	3	30	14	3/4	...	...	...	...	......	....
3/4	.....	..	..	..	...	1.1	3	30	14	3/4	...	...	...	...	......	....
1	.....	..	..	..	...	1.4	3	30	14	3/4	...	...	...	...	......	....
1 1/2	.....	..	..	..	...	1.8	3	30	14	3/4	...	...	...	...	......	....
2	.....	..	..	..	...	2.2	3	30	14	3/4	...	...	...	...	......	....
3	.....	..	..	..	...	3.2	5	30	14	3/4	...	...	...	...	......	....
5	.....	..	..	..	...	5.25	10	30	14	3/4	...	...	...	...	......	....
7 1/2	.....	..	..	..	...	7.75	10	30	14	3/4	...	...	...	...	......	....
10	.....	..	..	..	...	10.50	15	30	14	3/4	...	...	...	...	......	....
15	.....	..	..	..	...	14.50	20	30	12	3/4	...	...	...	...	......	....
20	.....	..	..	..	...	19.2	25	30	10	1	...	...	...	...	......	....
25	.....	..	..	..	...	24	30	30	8	1	...	...	...	...	......	....
30	.....	..	..	..	...	28	35	60	8	1	...	...	...	...	......	....
35	.....	..	..	..	...	32.50	50	60	6	1 1/4	...	...	...	...	......	....
40	.....	..	..	..	...	37	50	60	6	1 1/4	...	...	...	...	......	....
50	.....	..	..	..	...	46	60	60	4	1 1/2	...	...	...	...	......	....
60	.....	..	..	..	...	55	70	100	4	1 1/2	...	...	...	...	......	....
75	.....	..	..	..	...	69	100	100	2	1 1/2	...	...	...	...	......	....
100	.....	..	..	..	...	91	125	200	0	2	...	...	...	...	......	....
125	.....	..	..	..	...	111	150	200	00	2 1/2	...	...	...	...	......	....
150	.....	..	..	..	...	133	175	200	000	2 1/2	...	...	...	...	......	....
175	.....	..	..	..	...	156	200	200	200,000*	2 1/2	...	...	...	...	......	....
200	.....	..	..	..	...	179	225	400	0000	2 1/2	...	...	...	...	......	....

* Circular mils.

used, consisting of pipes or grounding rods driven in the earth, which often is treated with salt to reduce ground resistance. Ground connections should receive periodic inspection, and where artificial ground conductors are used resistance at the point where the system is actually earthed should be known at all times, and kept at a safe value. If grounding connections are neglected, what was originally installed as a protective measure may become a serious hazard.

CODES.—Numerous codes and standards have been set up, both national and local, covering apparatus for utilizing electricity. In laying out circuits for power distribution,

Table 14.—Direct-current Motor Fusing, Wiring and Full Load Current Data

Hp. of Motor	115 Volts: Approximate Full Load Current, amp.	Running Fuse	Ampere Capacity of Switch	Minimum Size of Wire, A.W.G.	Size of Conduit Underwriters	230 Volts: Approximate Full Load Current, amp.	Running Fuse	Ampere Capacity of Switch	Minimum Size of Wire, A.W.G.	Size of Conduit Underwriters
1	8	10	30	14	1/2	4	5	30	14	1/2
2	16	20	30	12	3/4	8	10	30	14	1/2
3	24	30	30	8	1	12	15	30	14	1/2
5	38	50	60	6	1 1/4	19	25	30	10	3/4
7 1/2	58	70	100	3	1 1/4	29	40	60	8	1
10	75	90	100	1	1 1/2	38	50	60	6	1 1/4
15	112	140	200	00	2	56	70	100	4	1 1/4
20	148	200	200	200,000*	2 1/2	74	100	100	1	1 1/2
30	221	275	400	300,000*	3	110	140	200	00	2
40	......	.....	.....	......	.....	145	190	200	200,000*	2 1/2
50	......	.....	.....	......	.....	175	225	400	0000	2 1/2
75	......	.....	.....	......	.....	260	350	400	400,000*	3
100	......	.....	.....	......	.....	351	450	600	600,000*	3 1/2

* Circular mils.

Table 15.—Recommended Ratings of Air Circuit Breakers for Standard Conductors

National Electric Code

Size of Conductors	Insulation: Rubber	Varnished Cambric	Other	Size of Conductors	Insulation: Rubber	Varnished Cambric	Other	Size of Conductors	Insulation: Rubber	Varnished Cambric	Other
A.W.G.	Ampere Rating			A.W.G.	Ampere Rating			C.M.	Ampere Rating		
14	15	15	20	2	90	100	125	350,000	300	325	450
12	20	25	25	1	100	100	150	400,000	325	325	500
10	25	25	35	0	125	150	200	500,000	400	450	600
8	35	35	50	00	150	175	225	600,000	450	500	600
6	50	50	70	000	175	200	275	700,000	500	600	...
5	50	50	70	0000	225	250	325	750,000	500	600	...
4	70	70	90	C.M.				800,000	550	...	...
3	70	90	100	250,000	250	300	325	900,000	600	...	...
				300,000	275	325	400				

Table 16.—Air Circuit Breaker Ratings for Use in A.C. Motor Branch Circuits

Breakers of the ratings given will trip at approximately 700% of motor current in not less than 10 sec.

Full-load Current Rating of Motor, amp.	Type of Motor* A	B	C	D	Full-load Current Rating of Motor, amp.	Type of Motor* A	B	C	D	Full-load Current Rating of Motor, amp.	Type of Motor* A	B	C	D
	Rating of Circuit Breaker, amp.					Rating of Circuit Breaker, amp.					Rating of Circuit Breaker, amp.			
0–6	15	15	15	15	61–66	100	100	100	100	183–195	...	300	300	300
7–9	20	20	20	15	67–90	125	125	125	125	196–202	...	325	325	325
10–12	25	25	25	20	91–110	150	150	150	150	203–225	...	350	350	350
13–18	35	35	35	25	111–129	.....	175	175	175	226–258	...	400	400	400
19–30	50	50	50	35	130–150	...	200	200	200	259–376	...	450	450	450
31–38	70	70	70	50	151–165	...	225	225	225	377–418	...	500	500	500
39–46	70	70	70	70	166–175	...	250	250	250	419–460	...	550	550	550
47–60	90	90	90	90	176–182	...	275	275	275	461–501	...	600	600	600

*Motors indicated by column designations are as follows: A, single-phase repulsion or split-phase; B, 3-phase squirrel-cage, full voltage starting; C, 3-phase squirrel-cage, reduced voltage starting; D, heavy-duty service, slip-ring.

the National Electrical Code should be consulted. Also local ordinances exist, which may vary in different localities. These should be known and observed. The accepted practice of A.I.E.E., N.E.M.A. and Edison Institute should be used as guides in electrical wiring and construction.

Table 17.—Air Circuit Breakers, 750 Volts and Less Required for Over-current Protection

Direct-current Machine Circuits

Circuit Controlled *	Breaker Required for Over-current Protection †
2-wire D.C. generator (engine-driven, not grounded). 2-wire D.C. generator (motor-driven). 2-wire synchronous converter. 2-wire battery (not grounded).	1 single-pole with 1 over-current coil. With generators, entire armature current flows through over-current coil. Generators used as exciters ordinarily require no protection, but power-directional relays with over-current relays connected in series, may be used. Generators charging large batteries usually require power-directional protection to prevent battery from pumping back into the generator. Usually only over-current protection is provided for small generators charging oil-circuit-breaker control batteries. No general rules can be laid down for battery work. For electrolytic generators below 25 volts and above 2000 amp., use automatic field switches, whose trip coils are actuated by main line current. Generators rated 25 volts and 2000 amp. or under, are not provided with automatic protection.
2-wire D.C. generator (engine-driven, negative grounded). 2-wire battery (negative grounded).	1 single-pole with 1 over-current coil if connected in negative (grounded) side. 2 single-pole with one over-current coil each for railway service. Positive side only is run to the panels. A breaker is connected in both positive and negative leads to take care of grounds. In both cases, entire armature or battery current must flow through over-current coil.
2-wire motor (not grounded).	1 single-pole with 1 over-current coil.‡ 1 double-pole with 1 over-current coil.‡
2-wire motor (negative grounded).	1 single-pole with 1 over-current coil connected in positive (ungrounded) lead.‡ If Underwriters' rules apply, 1 double-pole with 1 over-current coil in positive (ungrounded) lead.
3-wire generator. 3-wire synchronous converter with neutral derived from A.C. end. 3-wire battery.	1 double-pole with an over-current coil connected in each outside wire. In machine circuits, breaker must receive the entire armature current. Rated voltage of breaker must be equal at least to voltage between neutral and outside wires.
Balancer set for producing 3-wire system.	1 double-pole with over-current coil connected in each outside wire and carrying the entire line current. Rated voltage of the breaker must be equal at least to voltage between outside wires.

* Unless specifically stated otherwise, all circuits may be either grounded or ungrounded.

† The proper knife switches are to be used in all circuits, to disconnect them and take care of equalizing compound-wound machines.

‡ Where Underwriters' rules do not apply. See rules for modifications.

SECTION 16

POWER TEST CODES

TABLE OF CONTENTS

INSTRUMENTS AND APPARATUS

TEST CODES

POWER TEST CODES

Tests to determine the performance of power generating apparatus, or of equipment utilizing power, should be conducted under the Power Test Codes of the American Society of Mechanical Engineers. These codes prescribe the methods to be followed, the apparatus to be used, and the measurements and readings to be taken to completely determine the performance of power-generating or power-using apparatus.

Each type of apparatus is the subject of a separate code. In addition, there is published, in connection with the codes, supplementary information dealing with instruments and apparatus, whose use is common to all of the codes. The more important of these codes, and of the instruments and apparatus codes are abstracted below. Complete copies of any code can be obtained from the American Society of Mechanical Engineers, 29 West 39th St., New York.

In the tabulation of data and test results accompanying each of the codes, the following items, which are common to all codes have been omitted, and should be added to any report made in a test conducted under a code: 1. Date of test. 2. Location. 3. Owner. 4. Builder. 5. Test conducted by. 6. Object of test.

INSTRUMENTS AND APPARATUS

1. TEMPERATURE MEASUREMENTS

Approved, 1931

The methods of measuring temperature, the calibration of instruments, and the auxiliary apparatus to be used, together with precautions to be taken, are fully discussed in Section 3. See pp. 3–03 to 3–16. The data therein have been abstracted largely from the chapter on temperature measurement of the Power Test Codes.

2. SPEED MEASUREMENTS

Approved, 1930

Three general methods of obtaining angular speed of rotating machinery are: 1. By means of a revolution counter and time piece. 2. By use of a tachometer. 3. By use of a stroboscope. Linear speed is calculable from angular speed when the two are associated.

COUNTERS are instruments for determining the number of revolutions or strokes of rotating or reciprocating mechanisms. Continuous counters are used where a determination is sought over a considerable period of time. Hand counters are used for determinations over a short period. Continuous counters are of the following types:

A Direct-geared Counter comprises a series of gears arranged to rotate a set of dials or pointers that register the number of revolutions. It is positive, accurate, and is considered the most reliable type of counter.

A Cyclometer-type Counter is considered the next best type of counter. The set-back type counts one for each revolution of the shaft. A set-back device permits returning of all figures to zero. It will add when turned forward but it may not be depended upon to subtract when rotated backward. It is not recommended for determinations exceeding 500 counts per min. The locked-wheel type counts one for each revolution of the shaft and has no set-back device. It is especially recommended for high count determinations, and may be operated at twice the maximum speed of the set-back counter.

Set-back Rotary Ratchet Counters will count 10 for each complete forward rotation of the driving shaft, which is rotated by an oscillating motion of a lever on it. An oscillation of 40 to 60 deg. will count one on the number wheels. Backward rotation of the shaft will not disturb the number wheels. These counters may be operated at 150 counts per min., and for short periods at 200 or 300 counts per min. Without the set-back attachment, it may be run continuously at twice its normal speed if the driving shaft is fitted with ball-bearings.

Flat Ratchet Type Counters involve reciprocating motion and a ratchet drive. Unless

the travel is properly controlled, overtravel may occur especially at high speeds. This type is limited to slow count determinations, and is unsuited to high-accuracy determinations.

Continuous counters usually are fastened rigidly to a part of the machine. The primary drive is operated by gear, chain or flexible shaft in determining revolutions, or a system of continuous or intermittent linkages in determining number of strokes.

Hand Counters may be of either the dial or cyclometer type. If carefully handled and well made they are fairly reliable over a range of 200 to 2000 r.p.m. The possibilities of errors in these instruments are slippage between the shaft and the point of the instrument where contact is made, and also inaccuracy in determining time between making and breaking the contact. The counter should be allowed to operate at shaft speed for a few seconds, and, as the dial passes the zero mark, the stop watch should be started. After running for about two minutes, the stop watch should be stopped as the dial again passes the zero mark. This will give the time for an integral number of revolutions and eliminate errors due to slippage and sudden starting or stopping. The longer the period, the less will be the error. For a $^1/_5$-second stop watch, an error of one scale division on the watch will cause the following percentage of errors in observed r.p.m.

Period of contact.........	15 sec.	30 sec.	1 min.	2 min.	3 min.
Error in r.p.m., %........	±1.33	±0.67	±0.33	±0.17	±0.11

THE TACHOSCOPE is an instrument in which revolution counter and stop watch are mounted in one frame and arranged to operate simultaneously. The range varies from 0 to 5000 r.p.m. Its accuracy depends on the stop watch incorporated in it.

The Speed Indicator is an instrument operating on the same principle as the tachoscope, and averages the speed over a short period of time, indicating directly the speed in r.p.m. It may be used to measure speeds up to 30,000 r.p.m. The errors usually do not exceed 0.3 to 1.0% for readings taken in the upper part of the scale.

TACHOMETERS.—A tachometer indicates speed directly and continuously. Tachometers are made in several different types. They have a substantially constant r.p.m. error over the upper 70 to 80% of the working range, which usually is from 0 to 2500 or 0 to 1200 r.p.m. In most types the readings should be in the upper 75% of the range and preferably in the upper 50%. The driving gear ratio can be changed to meet any actual speed range.

Tachometers may be driven by any one of the following methods, the first three being preferred: 1. Gear or chain. 2. Flexible shaft or cable enclosed in flexible housing. 3. Speed pin threaded into the end of the revolving shaft under observation, with a metal sleeve, slotted in both ends, for a driving link between shaft and tachometer. These three all are positive drives. 4. Adapter on tachometer shaft pressed into contact with a so-called center on the rotating shaft of apparatus under test. 5. Belt and friction drive. This latter method should be avoided as unreliable.

Centrifugal Tachometers of the fly-ball and tilting-ring types operate through the centrifugal force produced by one balance ring or two or more revolving weights acting against springs. The compression of the springs moves the indicator or recorder pen. Good makes of this type of tachometer may have an error of 15 to 30 r.p.m. between the range of 600 and 2500 r.p.m. when new, and an error of 50 to 100 r.p.m., with a 25 to 50 or more r.p.m. difference between up and down readings, after being in service for some time. The temperature effect may be approximately 1 r.p.m. for each 10 deg. F.

Liquid Pump Tachometers comprise a centrifugal pump, a reservoir for liquid and an indicating tube into which the pump forces the liquid. Speed is determined by noting the position of the meniscus on the scale graduated in r.p.m. This instrument should be used only over the upper 40% of its range. The best instruments of this type, in good condition, can indicate speeds within 10 to 15 r.p.m. at 6000 r.p.m.

Air Tachometers are tachometers of the centrifugal type in which air is the working fluid as well as the transmitting medium. With it, indicating or recording mechanism may be located at a distance. An air tachometer should be used only in the upper 75% of its range. With a rotor making a maximum of 1500 r.p.m., it has an accuracy of 10 r.p.m. when provided with a sensitive mechanism for indicating the air pressure produced and when operating under standard atmospheric conditions.

The Autographic Hydraulic Pump Tachometers use an autographic speed recording mechanism which can portray speed changes incidental to changes of load, as in governor testing. It comprises three main elements, *i.e.*, the tachometer pump, the indicating mechanism and the auxiliary or return pump.

Force-drag Tachometers depend on the drag produced by a rotating element which transmits a force from it to another movable part, to which the indicator needle is attached. The motion of the needle is limited by a spring. These tachometers are used largely on

automobiles, but very little in power plant work. They are not recommended for use in connection with the Power Test Codes.

Chronometric Tachometers comprise a repeating type of combination clock and revolution counter. They depend on direct revolution counting during an increment of time. They cannot be depended on closer than 6 to 7 r.p.m. over the entire range of a scale of 2500 r.p.m. maximum reading, and may have errors as great as 20 r.p.m. when new, and an increased error of 5 or more r.p.m. after a reasonable amount of continuous service, with possibility of complete failure.

Electric Tachometers in commercial use are: 1. The electro-magnetic type, consisting of a direct-current generator with indicating volt meter. 2. An ordinary frequency meter calibrated to read speed instead of frequency. The electro-magnetic type should be used with direct drive at not over 2500 r.p.m. Its accuracy is not closer than 20 r.p.m. and it may have a probable error of 0.5% for each 20 deg. F. temperature change.

The frequency meter tachometer may be of the vibrating reed type, the induction type, or the moving coil type. The reed type indicates only certain fixed speeds, corresponding to the individual reed, and these are rarely closer together than 0.5%. The range is from 800 to 12,500 r.p.m. with an accuracy within 50 r.p.m. for individual reeds when new. It may be accurate only within 100 r.p.m. for individual reeds after use.

Vibration Tachometers are similar in construction to vibrating reed frequency meter type tachometers, excepting that they depend only on the mechanical vibration produced by the rotating member. Their range and limitations are the same as those of the electrical instrument.

STROBOSCOPES utilize the persistence of vision when an object is viewed intermittently. The end of the rotating shaft under observation, or a disc attached to it, may be marked with several dots located equidistantly around the circumference of a circle whose center coincides with the center of the shaft. Vision is interrupted by using either a tuning fork arrangement or a rotating perforated disc driven by a separate machine. When sighting through a perforated disc, separately driven, the speed of this disc is varied until the marks or figures on the rotating shaft or disc appear to be stationary. Then the speed of the shaft under observation will be the same as that of the separately-driven shaft, or some multiple of it, and equals (Indicated r.p.m. × number of holes in disc) ÷ (number of images). The number of images is the number of times a single mark on the indicating shaft is seen through all the holes in the disc of the adjusted-to-speed stroboscope. There is practically no limit to the range over which the stroboscope can be used.

3. PRESSURE

Approved, 1929

BAROMETERS are used to measure pressure of the atmosphere. Readings, usually, are in inches of mercury. Barometric pressure plus gage pressure is the *absolute pressure*. Where measured pressure is less than atmospheric pressure, the pressure indicated by the reading of a U-tube or vacuum gage is called the *vacuum*. The absolute pressure then is the difference between atmospheric pressure and the vacuum.

Mercurial barometer and mercury column readings always should be corrected to the value which would obtain if the mercury column were at 32° F. A calibration correction also should be applied. See below for methods of making these corrections.

The Mercurial Barometer is used generally and recommended. The diameter of bore of the glass tube of a barometer used under the test codes shall be not less than 0.25 in. The range usually is 25 to 32 in. of mercury. Special barometers for mountainous regions begin at 15 or 20 in. A vernier attachment permits readings to 0.001 in., although for many engineering purposes readings to 0.01 in. are sufficiently accurate.

The Aneroid Barometer consists of an exhausted chamber whose ends are corrugated diaphragms. Atmospheric pressure on the diaphragms, balanced by a stiff spring, deflects them. The deflection is transmitted to a pointer which gives the reading, equivalent to that which would be given by a mercury barometer in inches or millimeters of mercury. Aneroid barometers are not as reliable as mercurial barometers, the temperature correction being uncertain. Their use is permissible only where accuracy in determination of barometric pressure is not of primary importance. The range of accuracy is from 0.01 to 0.1 in.

The Recording Barometer or Barograph is an aneroid barometer actuating a pen moving over a recording drum rotated by clockwork. It is subject to all of the inaccuracies of the aneroid barometer, plus those due to the pen and its mechanism.

Installation of Barometers.—A barometer must be so installed as to be free from vibration, air currents, and violent temperature changes. It must be in good light, not directly exposed to the sun and not heated by nearby electric lamps. A mercurial barometer

must hang free to insure a vertical position of the column. It must have a small vent to

N.—Barometers should

for

Station (provided it is not over 20 miles distant) as shown on the daily weather maps issued by the bureau. Weather Bureau barometer readings always are taken at 8 A.M., Eastern Standard Time. If the station is at a different elevation than the barometer being calibrated, a correction should be made to compensate for this difference. Calibration also can be made by means of the weather maps. The location of the barometer is indicated by a dot on the map, and the barometric pressure is estimated by interpolation between the values shown on the two adjacent isobars. The isobars are for each 0.1 in., so that interpolation will give pressures to the nearest 0.01 in. The following is an example of the calibration of a mercurial barometer at West Lynn, Mass.

Data	
Uncorrected barometric pressure	29.800 in. Hg
Check reading	29.801 in. Hg
Temperature	75 deg. F.
Height of cistern above floor	3 ft.
Floor elevation above mean sea level	16.82 ft.
Latitude	42.8 deg. N.
Calibration	
Mean uncorrected reading	29.80 in. Hg
Temperature correction, Table 1	− 0.13 in. Hg
Elevation correction, Table 2	0.02 in. Hg
Gravity correction, Table 3	− 0.01 in. Hg
Corrected reading, reduced to sea level	29.68 in. Hg
Weather map reading, reported by Boston station of U. S. Weather Bureau reduced to sea level	29.67 in. Hg
Calibration correction	− 0.01 in. Hg

Observed readings of barometer shall be corrected by the data in Tables 1 to 3, or by Tables in U. S. Weather Bureau Circular F, or by the Smithsonian Tables. The following is an example of a mercurial barometer reading, correction and reduction at West Lynn, Mass.

Uncorrected barometric pressure (actual reading)	29.80 in. Hg
Barometer temperature	75 deg. F.
Temperature correction, Table 1	− 0.13 in. Hg
Gravity correction, Table 3	− 0.01 in. Hg
Calibration correction (*always* must be made)	− 0.01 in. Hg
Barometric pressure (at barometer elevation)	29.65 in. Hg
Level of barometer cistern *below* turbine center line	16 ft.
Elevation correction, Table 2	− 0.02 in. Hg
Barometric pressure at elevation of turbine center line	29.63 in. Hg

Reduced reading, $29.63 \times 0.4912 = 14.55$ lb. per sq. in. = absolute barometric pressure at elevation of turbine center line.

Temperature Corrections.—Table 1 is used to correct, for temperature, the readings of a mercurial barometer, mercury gage or U-tube. The correction reduces the reading to the value that would obtain were the mercury at 32° F. The table also includes a slight correction for temperature expansion of a brass scale which is correct at 62° F.

Elevation Corrections.—For exact work, gage readings must be corrected to account for weight of column of air between the level of the center line of apparatus under test, or any other reference plane, and the level where atmospheric pressure acts on the gage. See Table 2.

The correction is applied as follows: Let C = correction to be applied; c = correction value from Table 2; d = difference in elevation, ft. Then $C = (c \times d)/100$. If the gage reading is below atmosphere and the gage is set below the reference plane, or if the gage reading is above atmospheric pressure, and the gage is above the reference plane, C is *subtracted* from the gage reading. If the gage reading is below atmospheric pressure and the gage is above the reference plane, or if the gage reading is above atmospheric pressure and the gage is set below the reference plane, C is *added* to the gage reading.

Gravity Corrections need not be made for most engineering work, but Weather Bureau barometric readings always include a gravity correction. Hence the corrections in Table 3 are necessary for accurate calibration by comparison with a Weather Bureau reading.

Meniscus Corrections are not necessary with barometers, the correction being allowed for in setting of the scale. For mercury columns comprising a reservoir and a single leg, or for mercury U-tubes with legs of different diameters, the corrections of Table 4 should be added to the reading taken at the top of the meniscus.

Table 1.—Temperature Corrections for Barometers and Mercury Columns

Temp. of column, deg. F.	Observed reading of column in inches of mercury								
	16	18	20	22	24	26	28	30	32
	Add								
−20	0.07	0.08	0.09	0.10	0.11	0.11	0.12	0.13	0.14
−10	0.06	0.06	0.07	0.08	0.08	0.09	0.10	0.11	0.11
0	0.04	0.05	0.05	0.06	0.06	0.07	0.07	0.08	0.08
10	0.03	0.03	0.03	0.04	0.04	0.04	0.05	0.05	0.05
20	0.01	0.01	0.02	0.02	0.02	0.02	0.02	0.02	0.02
30	0.00	0.00	0.00	0.00	0.00	0.00	0.00	0.00	0.00
	Subtract								
35	0.01	0.01	0.01	0.01	0.01	0.01	0.02	0.02	0.02
40	0.02	0.02	0.02	0.02	0.02	0.03	0.03	0.03	0.03
45	0.02	0.03	0.03	0.03	0.04	0.04	0.04	0.04	0.05
50	0.03	0.03	0.04	0.04	0.05	0.05	0.05	0.06	0.06
55	0.04	0.04	0.05	0.05	0.06	0.06	0.07	0.07	0.08
60	0.05	0.05	0.06	0.06	0.07	0.07	0.08	0.08	0.09
65	0.05	0.06	0.07	0.07	0.08	0.09	0.09	0.10	0.10
70	0.06	0.07	0.07	0.08	0.09	0.10	0.10	0.11	0.12
75	0.07	0.07	0.08	0.09	0.10	0.11	0.12	0.13	0.13
80	0.07	0.08	0.09	0.10	0.11	0.12	0.13	0.14	0.15
85	0.08	0.09	0.10	0.11	0.12	0.13	0.14	0.15	0.16
90	0.09	0.10	0.11	0.12	0.13	0.14	0.15	0.17	0.18
95	0.10	0.11	0.12	0.13	0.14	0.16	0.17	0.18	0.19
100	0.10	0.12	0.13	0.14	0.15	0.17	0.18	0.19	0.20

Table 2.—Elevation Corrections for Barometers and Pressure Gages

In. of mercury decrease of atmospheric pressure per 100 ft. increase in elevation

Mean Altitude, ft.	Mean atmospheric temperature, deg. F.						
	−20	0	20	40	60	80	100
0	0.13	0.12	0.12	0.11	0.11	0.10	0.10
1000	0.12	0.12	0.11	0.11	0.10	0.10	0.10
2000	0.12	0.11	0.11	0.10	0.10	0.10	0.09
3000	0.11	0.11	0.10	0.10	0.10	0.09	0.09
4000	0.11	0.10	0.10	0.10	0.09	0.08	0.08
5000	0.10	0.10	0.10	0.09	0.09	0.08	0.08
6000	0.10	0.10	0.09	0.09	0.08	0.08	0.08
7000	0.10	0.09	0.09	0.09	0.08	0.08	0.08

Table 3.—Correction of Barometers and Mercury Columns to Standard Gravity

North latitude, degrees	Elevation, feet											
	0	0	2000	2000	4000	4000	6000	6000	8000	8000	10,000	10,000
	Height of column, in. of mercury											
	30	28	28	26	26	24	24	22	22	20	20	18
25	−0.05	−0.05	−0.05	−0.05	−0.05	−0.05	−0.06	−0.05	−0.06	−0.05	−0.05	−0.05
30	−0.04	−0.04	−0.04	−0.04	−0.05	−0.04	−0.05	−0.04	−0.05	−0.04	−0.05	−0.04
35	−0.03	−0.03	−0.03	−0.03	−0.03	−0.03	−0.04	−0.03	−0.04	−0.03	−0.04	−0.03
40	−0.02	−0.01	−0.02	−0.02	−0.02	−0.02	−0.03	−0.02	−0.03	−0.03	−0.03	−0.03
45	−0.00	−0.00	−0.01	−0.01	−0.01	−0.01	−0.01	−0.01	−0.02	−0.02	−0.02	−0.02
50	+0.01	+0.01	+0.01	+0.01	−0.00	−0.00	−0.00	−0.00	−0.01	−0.01	−0.01	−0.01

Table 4.—Mercury-meniscus or Capillarity Corrections

Internal diameter of tube, in.	Height of meniscus, in.							
	0.01	0.02	0.03	0.04	0.05	0.06	0.07	0.08
	Corrections to be added, in.							
0.15	0.024	0.047	0.069	0.092	0.116	.....	.....	.....
0.20	0.011	0.022	0.033	0.045	0.059	0.078	.....	.....
0.25	0.006	0.012	0.019	0.028	0.037	0.047	0.059	.....
0.30	0.004	0.008	0.013	0.018	0.023	0.029	0.035	0.042
0.35	.....	0.005	0.008	0.012	0.015	0.018	0.022	0.026
0.40	.....	0.004	0.006	0.008	0.010	0.012	0.014	0.016
0.45	.....	.....	0.003	0.005	0.007	0.008	0.010	0.012
0.50	.....	.....	0.002	0.004	0.005	0.006	0.006	0.007
0.55	.....	.....	0.001	0.002	0.003	0.004	0.005	0.005

Head and Water Column Corrections.—For precise work, the mean of temperatures at inlet and outlet of apparatus under test must be used. Values of density of water at various temperatures together with multipliers for reducing readings in inches of water to lb. per sq. in. or inches of mercury are given in Table 5. For ordinary engineering work variation of density with temperature need not be considered, and density is taken as corresponding to usual room temperatures. Multipliers for conversion of barometer and U-tube readings are given in Table 6.

Table 5.—Density of Water and Multipliers for Water Columns

Temp., deg. F	Density of water, lb. per cu. ft.	Multipliers to reduce in. of water column to Lb. per sq. in.	Multipliers to reduce in. of water column to In. mercury at 32 deg. F.	Temp., deg. F.	Density of water, lb. per cu. ft.	of water column to Lb. per sq. in.	of water column to In. mercury at 32 deg. F.
32	62.42	0.03614	0.07354	70	62.31	0.03606	0.07341
35	62.42	0.03614	0.07354	75	62.28	0.03604	0.07337
40	62.42	0.03614	0.07354	80	62.23	0.03601	0.07331
45	62.42	0.03614	0.07354	85	62.18	0.03598	0.07325
50	62.41	0.03613	0.07352	90	62.13	0.03595	0.07320
55	62.39	0.03611	0.07350	95	62.08	0.03592	0.07314
60	62.37	0.03609	0.07348	100	62.02	0.03589	0.07306
	62.34	0.03607	0.07344				

Table 6.—Multipliers for Barometers and Mercury and Water U-tubes and Gages*

Instrument	Reading	Remarks	Multipl Lb. per sq. in.	:n. of mercury	Feet of water
Barometer Mercury gage Mercury U-tube	In. Hg corrected to 32° F.	Correction made by Table 1	0.4912	1.00	70.74/d exactly .134 for water †
Mercury gage Mercury U-tube	In. Hg at t deg. F.	Use actual reading at actual temp. t	$0.4912 \times$ $1.0 - \frac{(t - 3}{10{,}000}$	$.0 - \frac{(t-32)}{10{,}000}$ ‡	$(70.74/d) \times$ $1.0 - \frac{(t-32)}{10{,}000}$
Mercury gage Mercury U-tube Barometer (in exceptional cases)	In. Hg †	Use uncorrected reading. Results are accurate at 56° F.	0.49		1.132 for water and mercury †
Mercury U-tube reading hydraulic pressure above atmosphere	In. Hg between levels of the two legs †	Water on top of lower column only, completely filling pressure pipe. Result is pressure at base of a column of water extending to zero level of mercury tube	$0.49 - \frac{0.036}{2}$ 0.472	1.00 − 0.037 †	1.090 for water and mercury †
Differential mercury U-tube reading difference between two hydraulic pressures, both above atmosphere	In. Hg between levels in the two	Water on top of both columns completely filling both pressure pipes Pressure is given between two points at same level	0.49 − 0.036 = 0.454	1.00 − 0.074	1.048 for water and mercury †
Water U-tube or water gag	In. water at t deg. F.	Requires precision gage and precision computations. See Table 5	$\frac{d}{1728}$	0.001178d	1/12 for same temperature of gage and column
Water U-tube or water gage	In. water	For usual engineering work	0.036	0.074 †	1/12
Bourdon pressure gage	Lb. per sq. in.	Reading to be corrected per gage calibration	1.00	2.036 ‡ 2.04 †	144/d exactly 2.31 for water †

* Readings are to be multiplied by the multipliers indicated. d is density of water, lb. per cu. ft., at actual temperature (see Table 5). † At usual room temperature. ‡ For in. mercury at 32° F.

TEST CODES

4. TEST CODE FOR SOLID FUELS

Approved, 1931

Solid fuels include the lignites, bituminous coals, anthracite and coke.

OBJECTS of testing solid fuels may be any one or more of the following:

Determination of: *a.* Composition of fuel. *b.* Heating value of fuel. *c.* Size of fuel. *d.* Character of combustion by analysis of products of combustion. *e.* Certain ash constituents. *f.* Fusing point of ash. Another object may be classification of fuel. (See p. 4–13.)

COLLECTING SAMPLE.—For a standard test, fuel shall be collected and prepared in accordance with the method described below.* If less accurate results are permissible, a smaller total weight of sample than specified in the code may be collected, with the understanding that the probable error in sampling varies inversely with the weight of the sample.

Every sample must be collected and prepared carefully. If sampling is improperly done, the sample will be in error, and it may be impossible to take another. Gross samples of the quantities designated must be taken, irrespective of the size of the lot to be sampled, because the larger the sample, the smaller will be the effect of accidental errors in sampling. Thus, the accidental inclusion of a 10-lb. piece of slate in a 100-lb. sample would cause an error of 10% in the ash content; in a 1000-lb. sample, the error would be but 1%.

Time of Sampling.—Fuel shall be sampled when it is being loaded into or unloaded from railroad cars, vessels, or wagons, or when discharged from supply bins, industrial railway cars, grab buckets or coal conveying equipment. If fuel is crushed as received, samples usually can be taken after fuel has passed crusher. Samples from the surface of fuel in piles, bins, cars, vessels, etc., usually are unreliable.

Size of Increments may be as small as 5 to 10 lb. for slack or small sizes of anthracite; for lump or run-of-mine coal, increments should be from 10 to 30 lb. All increments shall be of equal size.

Collection of Gross Sample.—Increments shall be regularly and systematically collected, so that the entire quantity sampled will be represented proportionately in the gross sample. Increments shall be collected with such frequency that a gross sample of the required amount will be obtained. Standard gross sample shall be not less than 1000 lb., except that for slack coal and small sizes of anthracite in which impurities are not in abnormal quantities, or in sizes larger than 3/4 in., gross sample may be 500 lb. If impurities are present in abnormal quantities, or are of large size, gross sample should be 1500 lb. or more. Gross sample should contain the same proportions of lump, fines and impurities as the fuel sampled. Preservation of the integrity of the sample is important.

Quantity Represented.—A gross sample shall be taken for each 500 tons or less, or with larger tonnages in agreed quantities.

Crushing.—The gross sample shall be systematically crushed, mixed and reduced in quantity to convenient size, either by hand or by mechanical means, for transmittal to laboratory. Conditions must be such as to prevent loss or accidental admixture of foreign matter. The largest size of fuel and impurities allowable as determined by visual inspection shall be as follows:

Weight of sample before division, lb.	1000	500	250	125	60	30
Largest size of fuel or impurity, in.	1	3/4	1/2	3/8	1/4	3/16*

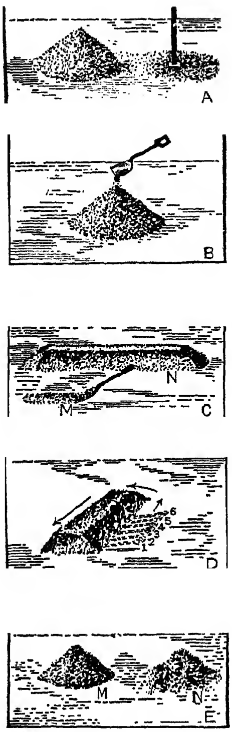
* To pass a No. 16 sieve.

The progressive reduction in size shall be done as follows, the various steps being shown in Figs. 1 to 3.

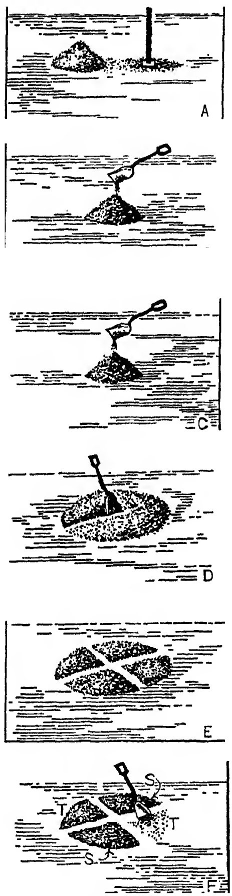
The gross sample shall be formed in a conical pile by depositing each shovelful on top of the preceding one (see Fig. 1B). It then is formed into a long pile by spreading a shovelful in a straight line, 5 to 10 ft. long, and of the width of the shovel (Fig. 1C), followed by a second shovelful, spread on top of the first, but in the opposite direction. The process is repeated, the pile occasionally being flattened, until all the coal has been formed in one long pile as at *N* in Fig. 1C. Half of the pile then is discarded by removing a shovelful from one corner of the pile and setting it aside, advancing along one side of the pile a distance equal to the width of the shovel and removing a second shovelful (Fig. 1D) and discarding it. The process is repeated, travelling around the pile, always in the same direction, until the pile has been divided into two equal parts. The two foregoing steps are repeated until sample has been reduced to 250 lb.

The 250 lb. sample then is reduced by quartering, as shown in Fig. 2, to obtain a 125 lb. sample. Samples of 125 to 250 lb. shall be thoroughly mixed by coning and reconing (Figs. 2B, 2C). Samples less than 125 lb. shall be mixed by rolling on a cloth (Fig. 3B) measuring about 6 × 8 ft., and then formed in a conical pile by gathering together the four corners of the cloth.

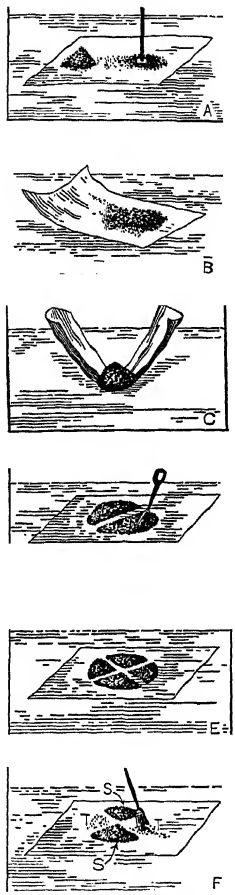
* This method is a modification of that worked out jointly by the Am. Chemical Society and the A.S.T.M. and described in A.S.T.M. specification D21-16.



1st and 2nd Stages



3rd Stage



4th, 5th, 6th Stages

FIG. 1A Crush sample
1B Cone sample
1C Mix in long pile
M—Spreading first shovelful
N—Pile completed
1D Halving by alternate shovelfuls
1E Reserved, M, shovelfuls 1, 3, 5
Rejected, N, shovelfuls 2, 4, 6

FIG. 2A Crush sample
2B Cone sample
2C Mix by forming new cone
2D Quarter after flattening cone
2E Sample quartered
2F Retain quarters SS
Reject quarters TT

FIG. 3A Crush sample
3B Mix by rolling on blanket
3C Form cone
3D Quarter after flattening cone
3E Sample quartered
3F Retain quarters SS
Reject quarters TT

Before each stage, sample is to be crushed to degree of fineness designated under Crushing.

Quartering is accomplished by flattening the cone, by pressing vertically downward with a board or shovel, and then marking it into quarters by two lines at right angles that intersect at a point corresponding to the apex of the original cone (Figs. 2 D, 2E). Diagonally opposite quarters are discarded. The quartering is repeated until a sample of the desired size is obtained. Six stages of obtaining a laboratory sample are shown in Figs. 1 to 3.

The laboratory sample immediately shall be divided into two parts, placed in suitable containers and so sealed as to preclude tampering. One container is sent to laboratory for analysis. The other is retained until the laboratory analysis is complete.

In collecting and reducing coke samples, all instruments for crushing shall be of iron or steel to permit removal by magnetic means of particles abraded from them.

DETERMINATION OF MOISTURE.—See p. 4–14.

PROXIMATE ANALYSIS.—See p. 4–14.

ULTIMATE ANALYSIS should be performed by a recognized laboratory. See A.S.M.E. Test Code for Solid Fuels and A.S.T.M. Specification D271–30 for procedure.

FUEL SIZING.—Gross samples shall be collected as above described, and passed over and through standard screens (see pp. 4–24, 4–35), and percentage by weight remaining

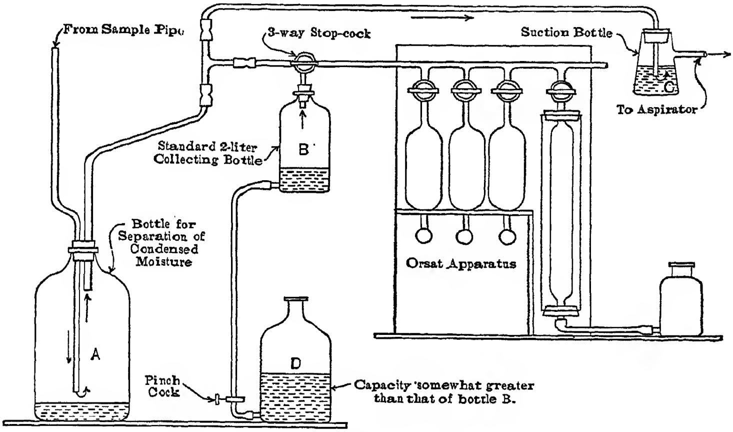


Fig. 4. Continuous Sampling Orsat Apparatus

over each screen determined. First screen in series shall be large enough to retain not over 5% by weight of total sample. Last screen will be of such size that not over 20% by weight of total sample will pass through.

In designating sizes of anthracite used for tests, size of perforations in screens through and over which the coal must pass should be definitely stated. Report should show percentage by weight passing over and retained by each size of screen.

FLUE GAS ANALYSIS shall be made by means of an Orsat apparatus, and shall determine CO_2, CO and O_2. If more than 0.5% CO is present, it shall be removed with two cuprous chloride pipes in the Orsat apparatus. If hydrogen is present, suitable apparatus, as the Orsat or Hempel should be used.

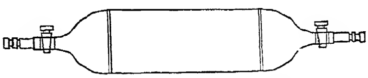


Fig. 5. Gas Sample Collecting Bottle

Sampling.—The standard sampler consists of single open-end iron, copper or glass exploring tubes (1/4 to 3/4 in. diam.). Ends of tubes are so placed that composition of sample is representative of gas being sampled. Fig. 4 shows apparatus, which draws a continuous sample from the main stream of gases. A small portion of this sample is drawn off continuously for 15 to 30 min. and then analyzed in the Orsat. If water is used in bottles *B* and *D*, it should be saturated with gas being analyzed, or concentrated brine may be used. If it is not feasible to draw sample directly into the Orsat, it should be collected at the aspirator in collecting bottles *B*, Fig. 4, or in tin or glass tubes, Fig. 5.

Gas samples may be collected in quantities proportioned to the rate of burning fuel, or in equal quantities at equal intervals. If the latter, average analysis must be determined by weighting the separate determinations proportionally to the weights of the fuel burned during the intervals of collecting samples from which analyses are made.

ASH.—Immediately on removal from ashpit, quench with water to stop further combustion. Method of collecting gross samples depends on whether or not total ash can be crushed.

Collection of Sample.—*Standard Method.* If total ash can be crushed to a maximum size of $^3/_4$ in., gross sample should be collected in increments of about 50 lb. per ton of ash, unless total gross sample is less than 1000 lb. In such event, increments shall be so increased as to give gross sample of 1000 lb. If total ash is less than 1000 lb., total ash is taken as gross sample. Sample shall be collected in galvanized iron cans with tight-fitting covers.

Alternate Method.—If crushing of ash in one operation is not feasible, separate clinker and fines by passing through a 2-in. mesh wire screen. Unburned combustible is to be separated by hand picking and added to fine ash. Weigh both clinker and fines, designating them as *weight of wet clinker from ashpit* and *weight of wet ash plus combustible from ashpit.* Crush and thoroughly mix separated clinker. Thoroughly mix wet ash. Mix crushed clinker and ash plus combustible in proportion, by weight, to the ratio of weight of wet clinker from ashpit and weight of wet ash plus combustible from ashpit. Weight increments in mixing process for gross sample shall be the same as in the standard method.

Moisture.—Sample to be taken from collected gross sample before crushing. A grab sample of 10 to 20 lb. will serve as moisture sample. Before taking it, mix gross sample by turning over several times with shovel.

To determine moisture, crush sample in jaw crusher to pass through a 4-mesh sieve, and reduce to a 5 lb. sample. Spread over galvanized iron pans (A.S.T.M. Specification D271–33) and place in an oven. Air is passed over samples at a maximum temperature of 200° F. until weight loss is not over 0.1% per hr.

Combustible in refuse shall be computed by means of difference in weight of total coal, computed weight of total dry ash from ultimate analysis, and weight of boiler refuse. Heating value of combustible is to be that of pure carbon.

To determine combustible in fuel sifting through grate, treat sifting sample according to method described in A.S.T.M. specification D271–33 for determination of ash in coal. Heating value of siftings shall be determined by the calorimetric method.

POWDERED COAL.—Ash. Gross samples from furnace shall be collected and weighed at convenient intervals, care being taken that none of the fine ash deposit is lost. Large slag formations should be broken down and weighed separately.

Gross ash samples from slack are best collected from a cinder catcher. No standard method is in use where a cinder catcher is not available. Methods that have been used successfully are those reported by the U. S. Bureau of Mines (Bull. 223), and that of the Chicago Dept. of Smoke Inspection (see 1915 Report of Chicago Assoc. of Commerce Committee on Smoke Abatement).

Laboratory samples of from 2 to 5 lb. from both gross ash slack and furnace samples shall be prepared by standard process of reducing, quartering and mixing. Slag to be analyzed shall be crushed to pass through a $^3/_{16}$-in. screen. Analysis of laboratory samples for combustible shall be by the standard method.

5. TEST CODE FOR LIQUID FUELS

Approved, 1930

The Test Code for Liquid Fuels is intended primarily to specify standard methods of determining those ascertainable chemical and physical properties which are extensively used in the generation of heat and power. The code consists essentially of the standards adopted by other societies, pertaining to the particular characteristic under investigation. Following is a list of the codes so adopted:

Methods of Collecting Samples: D270–27T, A.S.T.M.
Calorific Value: D240–27, A.S.T.M.
Gravity: Circular No. 154, National Standard Petroleum Oil Tables, Bureau of Standards; Bureau of Mines, Bull. No. 207.
Sulphur Content: D90–29T, A.S.T.M.; D129–27, A.S.T.M.
Water and Sediment: D95–28, A.S.T.M.; D96–28, A.S.T.M.
Viscosity: D88–26, A.S.T.M.
Distillation Range: D86–27, A.S.T.M.; D158–28, A.S.T.M.
Flash and Fire Points: D56–21, A.S.T.M.; D93–22, A.S.T.M.; D92–24, A.S.T.M.
Cloud and Pour Points: D97–28 A.S.T.M.
Color: D156–23T A.S.T.M.
Corrosion: D130–27T A.S.T.M.
Acidity: Method 510.0, Bureau of Mines, Technical Paper No. 323B.
Burning Quality of Wick Burners: D187–27, A.S.T.M.
Carbon Residue: D189–28, A.S.T.M.
Quantity Measurements: Circular No. 154, Bureau of Standards, National Standard Petroleum Oil Tables.

6. TEST CODE FOR STATIONARY STEAM-GENERATING UNITS

Approved, 1930

DEFINITION.—A steam-generating unit is a combination of apparatus for producing, furnishing, or recovering heat, together with apparatus for transferring the heat so made available to the fluid being heated and vaporized. Such a unit will include boiler, water walls, water floor, water screen, superheater, reheater, economizer, air heater, furnace and fuel burning equipment. The economizer and air heater are not included when the heat absorbed by them is not returned to the unit. This code applies to all of the component parts as above defined but not to the apparatus required for their operation.

Economizers may be classified as: *a*. Separate economizers, with their own housing and connected to the boiler by flues; *b*. integral economizers contained within the boiler setting and through which a part of the boiler circulating water may or may not pass.

When testing steam boilers equipped with class *a* economizers, observations may be separated either to include or exclude the economizer in the test results. With class *b* economizers, it is practically impossible to separate the economizer from the boiler, and tests of units so equipped should be conducted and reported as if the economizer was a part of the boiler.

PRECISION OF RESULTS.—The absolute accuracy of the results of a steam-generating unit test is doubtful. There is as yet no possible basis upon which to determine what the probable limit of errors might be. The more important of several sources of indeterminate error are discussed below. The limits of accuracy of a test reasonably may be taken to be within $\pm 3\%$.

SOURCES OF ERROR.—One source of probable error is the sampling of coal. Another source of error is the moisture contained in the coal. Despite the greatest care taken to obtain a representative sample, there may be an indeterminate error in the heating value of the coal, even though the laboratory determination is substantially correct as regards sample tested.

Similarly, it is problematical whether or not samples collected for determination of moisture in steam, and for gas analyses, are representative of the bulk.

While heat balances often are reported to the nearest B.t.u. and to the nearest 1/100 of 1%, the present state of the art does not provide means for attaining such accuracy. In general, results should be reported only to the nearest significant figure.

MEASUREMENTS listed below are the principal ones which must be made in a performance test of a stationary steam-generating unit: *a*. Area of heating surfaces; *b*. grate surface; *c*. furnace volume and dimensions; *d*. analysis and heating value of fuel; *e*. steam pressure; *f*. quality of steam; *g*. analysis of gases; *h*. draft pressures; *i*. steam temperatures; *j*. superheat; *k*. air temperatures; *l*. gas temperatures; *m*. feed-water temperatures; *n*. weight of fuel; *o*. weight of refuse; *p*. analysis of refuse; *q*. weight of water.

INSTRUMENTS AND APPARATUS required for boiler tests are: *a*. Scales for weighing coal, oil or other fuel, ashes, furnace refuse, etc.; *b*. graduated scales for water level measurements; *c*. tanks and scales for volumetric or weight measurement of water; *d*. meters or other apparatus for measuring gaseous fuels; *e*. pressure and draft gages; *f*. thermometers and pyrometers; *g*. calorimeter for determining quality of steam; *h*. gas sampling and analyzing apparatus. As no two plants are alike, the location of the various instruments only can be suggested.

Fuel Weighing Apparatus should be near point where fuel is to be used, and under direct observation of person in charge of test. Refuse may be weighed at any convenient point.

Water-level Scales should be so attached that breaking of water-glass will not disturb scales.

Water Weighing Apparatus, or apparatus for volumetric measurement, should be easily accessible and under direct observation of person in charge of test.

Gas Meters or other apparatus for measuring gaseous fuels may be located as conditions dictate.

Gages.—Pressure gages should be away from disturbing influences, such as extreme heat, and in a position to be read easily. Draft gages should be at not less than two points: (1) After the flue gas has passed beyond contact with any part of the boiler, and before it has passed the damper, or entered the outlet duct if there is no damper; (2) in the furnace immediately before the flue gas comes in contact with the heating surface of the boiler proper. Draft gages may be located at such other points as may be necessary to ascertain draft in various parts of the boiler.

Temperatures.—Feedwater temperatures should be measured as close to boiler inlet as possible. The pipe between boiler and thermometer should be protected with heat insulation. Saturated steam temperatures may be measured at any point in steam pipe where pressure is same as that at the point where the temperature is desired. Condensate must not be allowed to cool the thermometer well. Temperature and pressure of superheated steam should be measured as close to outlet of superheater as possible. Pyrometers must have the part on which heat impinges so located that temperature which it is desired to measure actually is obtained. Thermometers for determining atmospheric temperatures should be so placed as to give average indication, away from cold or hot air currents, surfaces, etc. For design of thermometer wells, methods of calibrating thermometers and pyrometers, see pp. 3–05 to 3–09.

Steam Calorimeter sampling tubes should be as close as possible to point at which quality of steam is desired. Usually, quality of steam leaving boiler or entering superheater is determined.

Flue Gas Analysis usually is made at point of exit of the gases. Frequently, analyses are desired at other points. Sample tubes must be so located that only the gas to be analyzed enters. The apparatus should be readily accessible and provided with good light.

STARTING AND STOPPING.—Combustion, fuel, draft and temperature conditions, water level, rate of feeding water, rate of steaming, and steam pressure should be as nearly as possible the same at the end as at the beginning of the test. If an economizer is included, the average temperature of the water in it should be the same at the start and end of the test.

Hand Firing.—To obtain equality of conditions with hand fired units, the following method should be used:

Heat the furnace by a preliminary run at the same combustion rate that will prevail during the test, and sufficiently long to thoroughly heat the setting. Burn the fire low and thoroughly clean it, leaving from 2 to 4 in. of live coal spread evenly over the grate. Estimate thickness of coal bed, note water level, steam pressure and time. Record the latter as starting time of test. Fresh coal then should be fired from that weighed for the test, the ashpit thoroughly cleaned and the regular test observations begun.

To end test, burn fire low and so clean it as to leave on the grate the same amount of live coal as at the start. When this condition is reached, note water level, steam pressure and time. Record the latter as stopping time. If water level is lower or higher than at beginning, make correction by computation and not by feeding additional water. Remove ashes and refuse from ashpit.

If several boilers are under test and it is impracticable to clean them simultaneously, fires should be cleaned one after another as rapidly as possible, and each one, after cleaning, charged with enough coal to maintain a thin fire in good working condition. After last fire is cleaned, burn all fires to 4 to 6 in. thick. Note thickness of each, the water levels, steam pressure and time. The latter is the starting time. To stop test, repeat procedure and make final observations.

Mechanical Stokers require a modification of the above procedure as follows:

Stokers should be in operation at approximately the same rate as will prevail during test for at least 12 hr. before starting test. At start and finish of test, level coal in stoker hopper. Make starting and stopping observations as in hand-fired tests.

With continuous dumping stokers, desired operating conditions should be maintained as nearly constant as possible for at least 1 hr., and preferably for 2 hr., before starting and stopping test. With intermittent dumping stokers, proceed as above, except that stokers should be cleaned about 1 hr. before starting and before stopping test.

Pulverized, Liquid or Gaseous Fuel.—Boiler to be tested should be operated for not less than 3 hr. before start of test, under the same fuel, furnace and combustion conditions that are to be maintained throughout. Fuel temperature, fuel pressure and draft conditions to be kept as nearly constant as possible during this period and throughout the test. The same observations as are made for starting and stopping hand-fired tests are made for pulverized, liquid and gaseous fuel tests.

DURATION of test to determine efficiency of a steam-generating unit burning coal, either hand- or stoker-fired, preferably should be 24 hr. Where operating conditions do not permit, the length of test may be reduced to not less than 10 hr. When rate of combustion is less than 25 lb. of coal per sq. ft. of grate surface per hr., the test should continue until 250 lb. per sq. ft. of grate surface has been burned, except that where a type of stoker is used that does not permit either or both quantity of fuel and the condition of the fuel bed to be accurately estimated, duration of test should not be reduced below that required to minimize the error.

Duration of tests of units using pulverized fuel should be not less than 6 hr.; with liquid or gaseous fuels, not less than 4 hr.

Duration of test of waste heat boilers or steam-generating units in connection with an industrial furnace, whose operation is continuous with constant furnace conditions, should be not less than 6 hr. If the industrial furnace operates in cycles, duration of the test should cover at least one cycle of furnace operation, with start and end of test at same point in cycle.

Duration of tests conducted under plant operation conditions where the service requires 24 hr. operation, with frequent shifts of firemen, should be at least 24 hr. Duration should be not less than 24 hr. when the unit operates regularly a certain number of hours and during the remainder of the day is banked.

Duration of test to determine maximum evaporative capacity, when efficiency is not determined, should be not less than 2 hr.

RECORDS.—Readings of instruments at 15 min. intervals usually are sufficient. With sudden and wide fluctuations, readings should be taken at 10 min. intervals, or intervals short enough to determine a true average.

Fuel.—Approximate quantity of fuel needed each hour should be determined and, if possible, delivered on firing floor at beginning of the hour. If the whole amount cannot be delivered at

beginning of the hour, convenient quantities should be weighed out at appropriate intervals. Quantity of fuel left on floor at end of the hour should be estimated.

When hopper scales are used, receiving coal from bunkers and discharging directly into furnace hoppers, hourly quantities may be roughly determined by estimating furnace conditions.

Hourly quantities should be properly noted on the log sheet, but only totals are to be used in final calculations.

Steam Rate.—Records should be so kept as to ascertain the approximate consumption of feed-water each hour and thereby determine the degree of uniformity of evaporation.

Sampling of Fuel should be done regularly throughout the test for purposes of analysis. See Test Code for Solid Fuels, p. 16–09.

Ashes and Refuse withdrawn from the furnace and ashpit during and at the end of the test should be weighed, if possible, in dry condition. If wet, the amount of moisture should be ascertained and allowed for, a sample being taken and dried for this purpose.

Ash sampling at best is subject to large errors and precautions should be taken to obtain as representative a sample as possible. If ash is sufficiently hot to allow combustion, it should be quenched with water immediately after dumping. Ash should not be weighed until heat of the refuse has driven off most of this moisture. If possible, ash should be crushed, mixed, and reduced to a laboratory size sample by successive quartering. If crushing is impracticable, clinkers and fines should be separated and each pile weighed and sampled separately. The two samples are then combined in proportion to the relative weights of the respective piles.

Where loss in cinders, soot and fly ash is important, amount and heat value of such materials should be determined separately from ashpit refuse.

Flue Gas Analyses should be made by the Orsat apparatus or some modification thereof. Gas samples preferably should be taken continuously. If momentary samples are obtained, analyses should be made as frequently as possible, noting furnace and firing conditions at time samples are drawn. Where firing is intermittent, samples should be taken at such intervals that the complete firing cycle will be covered by the average of individual readings.

Smoke Observations, when required, should be made regularly throughout the test at intervals of 5 min. or, if necessary, of 1 min.

DATA AND RESULTS should be reported in accordance with the following form. Items of data not provided for may be added, or if certain items are not required, they may be omitted. Unless otherwise indicated, quantities recorded should be the average of the observations.

DATA AND RESULTS OF TESTS OF STATIONARY STEAM-GENERATING UNITS.—SOLID, LIQUID AND GASEOUS FUELS

Item No.

Liquid Fuels			
		Date of test	
		Maker and type of boiler	
		Maker and type of water walls	
		Maker and type of superheater	
		Maker and type of economizer	
		Maker and type of reheater	
		Maker and type of air heater	
		Maker and type of fuel burning equipment	
		Test conducted by	
		Object of test	
		DESCRIPTION, DIMENSIONS, ETC.	
		Boiler heating surface	sq. ft.
		Water wall surface (water walls, screen or floor, and roof)	sq. ft.
		Superheater surface	sq. ft.
		Economizer surface	sq. ft.
		Reheater surface	sq. ft.
		Total steam-generating unit surface	sq. ft.
		Air heater surface	sq. ft.
		Grate surface	sq. ft.
21	21	Number of burners	
		Fuel burning equipment	
		Method of producing draft	
		Fuel	
		Volume of combustion space	cu. ft.
26	26	Area of furnace floor ....wide..........deep	sq. ft.
		Furnace, center of grate to nearest heating surface	ft.
28	28	Height, furnace floor to nearest heating surface	ft.
29	29	Furnace volume per sq. ft. of boiler heating surface	cu. ft.

(Continued on following page)

TEST OF STEAM-GENERATING UNITS (*Continued*)

Item No.

FUEL AND GAS ANALYSES AND DATA

Volatile matter .... percent

Fixed carbon .... percent

Ash .... percent

Moisture (as fired) .... percent

Heating value per lb. (as fired) .... B.t.u.

Heating value per lb. (dry) .... B.t.u.

Fusion temperature of ash .... deg. F.

Size of coal as fired ....

Flash point .... deg. F.

Ultimate Analyses

39 Carbon .... percent

40 Hydrogen .... percent

41 Oxygen .... percent

42 Nitrogen .... percent

.. Baumé gravity .... deg.

44 Sulphur .... percent

45 Ash .... percent

46 46 Moisture .... volume* ...... percent .... weight* ....

Carbon monoxide, CO ..... volume .... percent .... weight ....

Hydrogen, H_2 .... volume .... percent .... weight ....

Methane, CH_4 .... volume .... percent .... weight ....

Acetylene, C_2H_2 .... volume .... percent .... weight ....

Ethylene, C_2H_4 .... volume .... percent .... weight ....

Ethane, C_2H_6 .... volume .... percent .... weight ....

Hydrogen sulphide, H_2S ... volume .... percent .... weight ....

Oxygen, O_2 .... volume .... percent .... weight ....

Nitrogen, N_2 .... volume .... percent .... weight ....

Carbon dioxide, CO_2 ...... volume .... percent .... weight ....

Heating value per cu. ft. (standard conditions) .... B.t.u.

Heating value per lb. .... B.t.u.

Weight per cu. ft. (standard conditions) .... lb.

Weight of total hydrogen per lb. of fuel .... lb.

Gases of Combustion

Gas analyis, furnace, percent, CO_2 ...., O_2 ...., CO ...., N_2 ...., SO_2 ....

Gas analysis, boiler outlet, percent, CO_2, ..., O_2 ...., CO ...., N_2 ...., SO_2 ....

Gas analysis, economizer outlet, percent, CO_2 ...., O_2 ...., CO ...., N_2 ...., SO_2 ....

Gas analysis, air heater outlet, percent, CO_2 ...., O_2 ...., CO ...., N_2 ...., SO_2 ....

Dry gas per lb. fuel, furnace (as fired) (dry) .... lb.

Dry gas per lb. fuel, boiler outlet (as fired) (dry) .... lb.

Dry gas per lb. fuel, economizer outlet (as fired) (dry) .... lb.

Dry gas per lb. fuel, air heater outlet (as fired) (dry) .... lb.

Dry gas per lb. fuel, theoretical (as fired) (dry) .... lb.

Air supplied per lb. fuel, furnace (as fired) (dry) .... lb.

PRESSURES AND DRAFTS

Moisture in air .... lb. per lb. air

Steam pressure by gage, boiler .... lb. per sq. in.

Steam pressure by gage, superheater outlet .... lb. per sq. in.

Steam pressure by gage, reheater inlet .... lb. per sq. in

Steam pressure by gage, reheater outlet .... lb. per sq. in.

Air pressure in ashpit zone, at burners .... in. of water

Pressure of fuel at burners .... lb. per sq. in.†

Pressure of air for combustion at burners .... in. of water

Draft in furnace .... in. of water

Draft at boiler outlet .... in. of water

Draft at economizer outlet .... in. of water

Draft at air heater outlet .... in. of water

* For gaseous fuels only.

† For gaseous fuel, in. of water.

TEST OF STEAM-GENERATING UNITS (*Continued*)

Item No.

Solid Fuels				
			TEMPERATURES	
83	83	83	Steam temperature at superheater outlet	deg. F.
84	84	84	Moisture in steam	percent
85	85	85	Superheat	deg. F.
86	86	86	Moisture or superheat in steam entering reheater	percent or deg. F.
87	87	87	Superheat in steam leaving reheater	deg. F.
88	88	88	Temperature of air surrounding boilers (t_1)	deg. F.
89	89	89	Temperature of air entering air heater (t_2)	deg. F.
90	90	90	Temperature of air leaving air heater (t_3)	deg. F.
91	91	91	Temperature of air for combustion (t_4)	deg. F.
92	92	92	Temperature of furnace (t_5)	deg. F.
93	93	93	Temperature of gases leaving boiler (t_6)	deg. F.
94	94	94	Temperature of gases leaving economizer (t_7)	deg. F.
95	95	95	Temperature of gases leaving air heater (t_8)	deg. F.
96	96	96	Temperature of feedwater entering boiler (t_9)	deg. F.
97	97	97	Temperature of feedwater entering economizer (t_{10})	deg. F.
98	98	98	Temperature of water in boiler at point where gases leave boiler (t_{11})	deg. F.
99	99*	99*	Temperature of fuel (t_{12})	deg. F.
			HOURLY QUANTITIES	
100	100	100	Duration of test	hr.
101	101	101	Fuel per hr., as fired	lb.
102	102	...	Fuel per hr., dry	lb.
		103	Fuel gas per hr. (standard conditions)	cu. ft.
104			Fuel as fired per sq. ft. of grate per hr.	lb.
		105	Fuel gas per burner per hr.	lb.
	106	106	Fuel per cu. ft. furnace volume per hr. (as fired)	lb.
107	107		Fuel as fired per retort or per burner per hr.	lb.
108			Dry fuel per sq. ft. of grate per hr.	lb.
109	109		Dry fuel per retort or per burner per hr.	lb.
110			Combustion space per lb. of coal per hr. (as fired) (dry)	cu. ft.
...	111		Dry fuel per cu. ft. of furnace volume per hr.	lb.
112	...		Refuse per hr.	lb.
113	113	113	Actual water per hr.	lb.
114	114	114	Steam through reheater per hr.	lb.
			UNIT QUANTITIES	
115	115	115	Heat absorbed by water in economizer	B.t.u. per lb.
116	116	116	Heat absorbed by water and steam in boiler	B.t.u. per lb.
117	117	117	Heat absorbed by steam in superheater	B.t.u. per lb.
118	118	118	Heat absorbed by steam in reheater	B.t.u. per lb.
			HOURLY QUANTITIES	
119	119	119	Rate of heat absorption in economizer	*k*B. per hr.†
120	120	120	Rate of heat absorption in boiler	*k*B. per hr.
121	121	121	Rate of heat absorption in superheater	*k*B. per hr.
122	122	122	Rate of heat absorption in reheater	*k*B. per hr.
123	123	123	Total rate of heat absorption by steam-generating unit	*k*B. per hr.
			REFUSE	
124			Refuse, percent of fuel (as fired) (dry)	percent
125			Percentage of combustible in refuse	percent
126			Carbon burned per lb. of fuel (as fired) (dry)	percent
			EVAPORATION	
127	127	127	Rate of heat absorption per lb. fuel (as fired)	*k*B.
128	128		Rate of heat absorption per lb. fuel (dry)	*k*B.
		129	Rate of heat absorption per cu. ft. fuel (standard conditions)	*k*B.
130	130	130	Rate of heat absorption per sq. ft. of steam generating unit surface per hr.	*k*B.
			EFFICIENCY	
131	131	131	Efficiency of steam-generating unit	percent
132	132	132	Comparative efficiency of steam-generating unit (when not equipped with air heater)	percent

* At burner.
† 1 *k*B. = 1000 B.t.u.

Computations for Test of Stationary Steam-generating Unit

Test may be on an "as fired" or "dry" basis. If on an "as fired" basis the word "dry" should be eliminated wherever it appears in the report form and *vice versa*. The basis chosen must be followed throughout the computations.

Notation.—CO_2, CO, O_2, N_2 = respectively percentage, by volume, in gases of combustion of carbon dioxide, carbon monoxide, oxygen and nitrogen. C = carbon content per lb. of fuel gas = ($^3/_7$ item 47 + $^3/_4$ item 49 + $^{12}/_{13}$ item 50 + $^6/_7$ item 51 + $^4/_5$ item 52 + $^3/_{11}$ item 56) (by weight); H = total heat of saturated steam at boiler outlet pressure; H_1 = total heat of superheated steam at superheater outlet pressure; H_2, H_3 = total heat of steam at reheater inlet and outlet pressures, respectively; h = total heat in feedwater at boiler inlet; h_1 = total heat in feedwater at economizer inlet; L = latent heat in steam at pressure in steam main. All heats are in B.t.u. per lb. t_s = temperature of steam after expansion in calorimeter; t_1 to t_{12} = (see items 88 to 99); kB. = 1000 B.t.u. Italic figures below refer to item numbers in the Data and Results form p. 16–15.

Item No.

30–4 Solid Fuels = dry-analysis items × {1 − (item *33*/100)}

33–44 Liquid Fuels = dry analysis items × {1 − (item *33*/100)}

46–60 Liquid Fuels. Should be reported, and B.t.u. value calculated, on dry basis. Calorimeter determination should be corrected for moisture. If $t_{12} > t_1$, add to calculated B.t.u. value, mean specific heat × $(t_{12} - t_1)$. Standard conditions are 29.92 in. Hg and 68° F. Use data items *61–64*

58 = Item *57*/item *59*

60 = Item *48* + $^1/_4$ item *49* + $^1/_{13}$ item *50* + $^1/_7$ item *51* + $^1/_5$ item *52* + $^1/_{17}$ item *53*

65–68 Solid Fuels = $[\{11CO_2 + 8O_2 + 7(CO + N_2)\}/\{3(CO_2 + CO)\}]$ × item *126*

Liquid Fuels = $[\{11CO_2 + 8O_2 + 7(CO + N_2)\}/\{3(CO_2 + CO)\}]$ × (item *39*/100)

Gaseous Fuels = $[\{11CO_2 + 8O_2 + 7(CO + N_2)\}/\{3(CO_2 + CO)\}] \times C$

69 Solid Fuels = 12.52 × item *126* + {(26.56 × item *40* + 5.325 × item *44* + it

Liquid Fuels = (12.52 × item *39* + 26.56 × item *40* + item *42* + 5.325 × item

Gaseous Fuels = {(3.46 × item *47* + 26.56 × item *48* + 16.03 × item *49*)/100}
+ {(13.60 × item *50* + 14.52 × item *51* + 15.33 × item *52*)/100}
+ {(6.57 × item *53* + item *55* + item *56*)/100} (by weight)

70 Solid Fuels = Item *65* + {(9 × item *40*)/100} − {(100 − item *124*)/100}

Liquid Fuels = Item *65* + {(9 × item *40*)/100} − 1

Gaseous Fuels = Item *65* + (9 × item *60*) − 1

84 Determined by charts or throttling calorimeter. Formula used with latter is
$100 \times \{H - 1150.4 - 0.47 \times (t_s - 212)\}/L$

112 = (combined weight of ashes and cinders collected from all sources during test) ÷ item *78*, or,
= {Item *32*/(100 − item *125*)} × item *101* (or *102*)

115 = $(h - h_1)$

116 = $(H - h)$

117 = $(H_1 - h)$

118 = $(H_3 - H_2)$

119 = (Item *113* × item *115*)/1000

120 = (Item *113* × item *116*)/1000

121 = (Item *113* × item *117*)/1000

122 = (Item *114* × item *118*)/1000

123 = (Item *119* + item *120* + item *121* + item *122*)

124 = Item *112*/item *101* (or *102*)

126 = (Item *39*/100) − (item *124* × item *125*/10,000)

127

128 = Item *123*/item *102*

129 = Item *123*/item *103*

130 = Item *123*/item *18*

131 Solid and Liquid Fuels = {Item *127* (or *128*) × 1000/item *34* (or *35*)} × 100

Gaseous Fuels = {Item *127* (or *128*) × 1000/item *58* (or *57*)} × 100

132 Solid and Liquid Fuels = (Item *127*×1000×100)/{item *34*+(t_4-70) × (item *70* × 0.24)}, or
= (Item *128*×1000×1000)/{item *35*+(t_4-70)×(item *70*×0.24)}

Gaseous Fuels = (Item *127* × 1000 × 100)/{item *58* + $(t_4 - 70)$ × (item *70* × 0.24)}, or
= (Item *128* × 1000 × 100)/{item *57* + $(t_4 - 70)$ × (item *70* × 0.24)}

HEAT BALANCE—SHORT FORM

NOTE. The code gives additional more elaborate heat balance forms for units comprising boiler, superheater and reheater, with or without integral economizer; for units comprising boiler, superheater, reheater and economizer; for units comprising boiler, superheater, reheater, economizer and air heater. These are omitted here, and the reader is referred to the code for them. For all except the most elaborate tests, the short form is sufficiently comprehensive.

Item No.				B.t.u. per lb. of fuel		Per-cent
		eous		As fired	Dry	
133	133	133	Heating value of fuel	.....	.....	.....
134	134	134	Heat absorbed by water in economizer	.....	.....	.....
135	135	135	Heat absorbed by water and steam in boiler	.....	.....	.....
136	136	136	Heat absorbed by steam in superheater	.....	.....	.....
137	137	137	Heat absorbed by steam in reheater	.....	.....	.....
138	138	138	Heat absorbed by steam generating unit	.....	.....	.....
139	139	...	Heat loss due to moisture in fuel	.....	.....	.....
...	...	140	Heat loss due to moisture in fuel	.....	.....	.....
141	141	...	Heat loss due to water from combustion of hydrogen	.....	.....	.....
...	...	142	Heat loss due to water from combustion of hydrogen	.....	.....	.....
143	143	143	Heat loss due to moisture in air	.....	.....	.....
144	144	144	Heat loss due to dry chimney gases	.....	.....	.....
145	...		Heat loss due to incomplete combustion of carbon	.....	.....	.....
	146		Heat loss due to incomplete combustion of carbon	.....	.....	.....
		147	Heat loss due to incomplete combustion of carbon	.....	.....	.....
148			Heat loss due to unconsumed combustible in refuse	.....	.....	.....
149			Heat loss due to unconsumed hydrogen and hydrocarbons and unaccounted for	.....	.....	.....
	150		Heat loss due to unconsumed hydrogen and hydrocarbons and unaccounted for	.....	.....	.....
		151	Heat loss due to unconsumed hydrogen and hydrocarbons and unaccounted for	.....	.....	.....

Heat Balance Computations

Item No. Solid Fuels	Item No. Liquid Fuels	Item No. Gaseous Fuels	
134	134	134	(Item *119* $\times$ 1000)/item *101* (or *102*)
135	135	135	(Item *120* $\times$ 1000)/item *101* (or *102*)
136	136	136	(Item *121* $\times$ 1000)/item *101* (or *102*)
137	137	137	(Item *122* $\times$ 1000)/item *101* (or *102*)
138	138	138	(Item *134* + item *135* + item *136* + item *137*)
139	139	...	(Item *38*/100) $\times$ $(1090.7 + 0.455\,t_6 - t_{12})$
...	...	140	(Item *46*/100) $\times$ $(1090.7 + 0.455\,t_6 - t_{12})$
141	141	...	(Item *40*/100) $\times$ 9 $\times$ $(1090.7 + 0.455\,t_6 - t_{12})$. See note *a*.
...	...	142	(Item *60* $\times$ 9/100) $\times$ $(1090.7 + 0.455\,t_6 - t_{12})$. See note *a*.
143	143	143	(Item *70* $\times$ item *71* $\times$ $0.47\,(t_6 - t_1)$. See note *b*.
144	144	144	Item *66* (*67* or *68*) $\times$ $0.24\,(t_6 - t_1)$. See note *c*.
145	...	...	$\{CO/(CO_2 + CO)\}$ $\times$ item *126* $\times$ 10,160. See note *d*.
...	146	...	$\{CO/(CO_2 + CO)\}$ $\times$ (item *39*/100) $\times$ 10,160. See note *d*.
...	...	147	$\{CO/(CO_2 + CO)\}$ $\times$ C $\times$ 10,160. See note *d*.
148	...	...	{(Item *124* − item *32*)/100} $\times$ 14,600
149	...	...	Item *34* (or *35*) − {(items *134* + *135* + *136* + *137* + *138* + *139* + *141* + *143* + *144*)}
...	150	...	Item *34* (or *35*) − {(items *134* + *135* + *136* + *137* + *138* + *139* + *141* + *143* + *144*)}
...	.	151	Item *58* − {items (*134* + *135* + *136* + *137* + *138* + *140* + *142* + *143* + *144*)}

NOTES: *a*. If economizer is installed without air heater, substitute t_7 for t_6; if air heater is installed, substitute t_8 for t_6 in items *139*, *141*, *143*, *144* (for items *140*, *142*, *143*, *144* with gaseous fuels). *b*. This loss is small and often is included in item *144*. *c*. If boiler and superheater, use item *66*; if economizer, use item *67*; if air heater, use item *68*. *d*. If boiler alone, use analysis from item *62*; if economizer, use item *63*; if air heater, use item *64*.

7. TEST CODE FOR FEEDWATER HEATERS

Approved, 1925

This code applies to open and closed boiler feedwater heaters and, with slight modifications, it may apply to heaters for heating water for any purpose when the heating element is either live or exhaust steam.

OBJECT usually is to determine: *a.* Whether heater meets design conditions. *b.* The variation, with capacity, of temperature rise of water and friction drop in water in closed heaters. *c.* The closeness with which outlet water temperature approaches steam temperature corresponding to pressure in the open heater.

INSTRUMENTS AND APPARATUS required for a heater test comprise:

a. Barometer, preferably mercurial. *b.* Mercury columns for measuring vacua and low pressures, having scale variations not greater than 0.01 in. with vernier attachments. *c.* Bourdon gages for measuring pressures too high for mercury columns. *d.* Thermometers graduated by 1/2 deg., with scale readings from 32° to 350° F. for determining temperatures of feedwater, condensate and vapors, and graduated by 1 deg., with scale readings from 32° to 350° F. for determining steam temperatures. *e.* Tanks and platform scales for measuring water, or water meters cali-

DATA AND RESULTS OF TEST OF OPEN FEEDWATER HEATERS

Item		
	DESCRIPTIONS AND DIMENSIONS, ETC.	
7	Type of heater	
8	External dimensions of heater	
9	Gross volume of heater	cu. ft.
10	Weight of heater, empty	lb.
11	Weight of heater when operating (including water)	lb.
12	Volume of steam space	cu. ft.
13	Shape and dimensions of steam inlet opening into heater	in.
14	Size of water inlet	in.
15	Size of water outlet	in.
16	Material of shell	
17	Volume of water in heater at operating water level	cu. ft.
18	Volume of water between overflow level and level at which make-up valve will open	cu ft.
19	Rated capacity of heater, water per hour	lb.
20	Location of thermometers	
21	Number and arrangements of baffles or trays	
22	Description of filtering or purifying arrangement	
23	Size and arrangement of venting connections	
24	Description of metering apparatus installed in heater	
25	Description of automatic steam and water control	
26	Description of water distributing boxes	
27	Nature and amount of insulation on heater	
	OBSERVED DATA	
28	Duration of test	
29	Barometer	in. Hg
30	Room temperature	deg. F.
31	Quantity of water admitted to heater	lb. per hr.
32	Inlet-water temperature	deg. F.
33	Outlet-water temperature	deg. F.
34	Steam pressure in heater, gage............lb. per sq. in., or	in. of Hg
35	Steam temperature in heater, by thermometer	deg. F.
36	Steam temperature at inlet, by thermometer	deg. F.
37	Steam used per hour	lb.
38	Total water discharged from heater	lb. per hr.
39	Volume of water in mixing compartment of heater	cu ft.
40	Volume of water in storage compartment of heater	cu ft.
41	Pressure drop from steam end of heater to vent end	
42	Time lag between occurrence of steam deficiency and change of outlet temperature	
43	Analysis of water entering heater	
44	Analysis of water leaving heater	
45	Oxygen content of water entering heater	
46	Oxygen content of water leaving heater	
	COMPUTED AND DEDUCED DATA	
47	Steam temperature corresponding to absolute pressure	deg. F.
48	Quality of steam supplied to heater	percent moisture or deg. superheat
49	Temperature difference between steam temperature corresponding to heater pressure and outlet water temperature	deg.F.
50	Lb. steam theoretically required per lb. water	
51	Lb. steam used per lb. water, actual	
52	Time required to empty heater when operating at rated capacity	
53	Steam lost up stack	lb. per hr.

brated in place under conditions of use. *f.* Steam calorimeter, throttling or separating. *g.* Apparatus for testing oxygen content of water.

It is desirable that wherever possible rate of water flow, pressures and temperatures be obtained by continuous recording instruments, in addition to observations made by instruments for instantaneous measurement.

OPEN HEATERS may be designed to heat water within a few degrees of the steam temperature corresponding to pressure in heater. Since in most open heaters water is filtered or treated as well as heated, arrangements should be made to obtain analyses of water entering and leaving heater. When these heaters are used for partial deaeration of water, means should be provided to sample water and determine its oxygen content.

DATA AND RESULT OF TEST OF CLOSED FEEDWATER HEATERS

Item		
	DESCRIPTIONS AND DIMENSIONS, ETC.	
7	Type of heater.	
8	Position of heater, horizontal or vertical	
9	Condition of heating surface	
10	Number of tubes	
11	Number of passes	
12	Length of single tube	ft., in.
13	Distance of travel of water through heater	ft., in.
14	Special type of tube, description	
15	Outside diameter of tube	in.
16	Thickness of tube	in.
17	Heating surface, of tubes, outside of tube	sq. ft.
18	Diameter of heater over shell	ft., in.
19	Length of heater over shell	ft., in.
20	Thickness of shell	in.
21	Material of tubes	
22	Material of shell	
23	Weight of heater, empty	lb.
24	Weight of water in heater	lb.
25	Gross volume of heater	cu. ft.
26	Shape and dimensions of steam inlet opening into heater	in.
27	Arrangement of steam supply pipes into heater	
28	Size of water inlet and outlet	in.
29	Size of drain	in.
30	Type and size of drain trap	
31	Location and type of air vents	
32	Arrangement of baffles	
33	Nature and amount of insulation on heater	
34	Location of thermometers	
	OBSERVED DATA	
35	Duration of test	
36	Barometer reading	in. Hg
37	Room temperature	deg. F.
38	Quantity of water through heater	lb. per hr.
39	Steam pressure in heater, gage	lb. per sq. in. or in. Hg
40	Steam temperature in heater by thermometer	deg. F.
41	Drain temperature	deg. F.
42	Vent temperature by thermometer	deg. F.
43	Steam temperature at inlet thermometer	deg. F.
44	Inlet water temperature	deg. F.
45	Outlet water temperature	deg. F.
46	Weight of steam used	lb. per hr.
47	Inlet water pressure, gage	lb. per sq. in.
48	Outlet water pressure, gage	lb. per sq. in.
49	Water-pressure drop by differential mercury column	
	COMPUTED AND DEDUCED RESULTS	
50	Velocity of water in tubes	ft. per sec.
51	Steam temperature in heater corresponding to absolute steam pressure	deg. F.
52	Quality of steam supplied to heater	percent moisture or deg. superheat
53	Temperature rise	deg. F.
54	Logarithmic heat-transfer coefficient, B.t.u. per hour per sq. ft. of surface, per degree of logarithmic mean temperature difference.	
55	Weight of steam theoretically required per lb. of water, computed from heat balance	lb
56	Weight of steam used per lb. of water	lb.
57	Water-pressure drop in heater	lb. per sq. in.
58	Steam lost up stack	lb. per hr

It is important to determine time required to empty heater under normal operating conditions in event of failure of water supply to heater, in order to establish rated capacity and size of heater. Equally important in establishing capacity is time lag between occurrence of insufficient steam for a desired outlet temperature and appearance of improper temperature at water outlet. The test, if possible, should determine these items in terms of definite rates and temperatures.

CLOSED HEATERS have a definite relation between capacity and outlet water temperature, determined by ability of the surface to transfer heat as measured by the heat transfer coefficient. The principal items affecting value of this coefficient are tube type, length and arrangement, water velocity, condition of tubes as regards presence of a film of scale, oil or dirt, and presence of accumulated air in steam space of heater. During tests, the shell should be thoroughly drained of condensate and vented air. Accessories necessary for good operation are pressure gages, thermometers, safety valves, vacuum breaker, water gage glass and trap. The thermometer in the steam space must not be installed where there is liable to be an air pocket or near a cold water manifold. If near cold tubes, it must be shielded for radiation.

CALCULATION OF RESULTS.—Logarithmic heat transfer coefficient is computed by

$$K = (w/S)\log_e\{(T - T_i)/(T_s - T_o)\}$$

where K = heat transfer coefficient, B.t.u. per hr. per sq. ft. of surface per deg. F. of logarithmic mean temperature difference; w = water per hr., lb.; S = heating surface measured on outside of tubes, sq. ft.; T_s = steam temperature in heater, deg. F. (if superheated steam is supplied, use temperature of saturated steam at pressure in heater); T_i, T_o = inlet and outlet water temperatures, respectively, deg. F.

Heat transfer coefficient when plotted against velocity of water in the tube gives a straight line on logarithmic paper, from which data may be obtained for determining constants in the equation $K = av^n$, where v = velocity of water, ft. per sec.

Friction drop when plotted against velocity usually gives a straight line on logarithmic paper, which serves to determine constants in friction drop formula, $H = bLv^m$, where H = total friction drop, lb. per sq. in., L = length of tube, ft. (if multi-pass, length of total path of travel of water in heater); v = velocity of water ft. per sec. Constants obtained in these formulas may be used to compare performance of heater under test with that of other heaters.

Record.—Observations and computed and deduced data should be recorded in the accompanying forms, for open and closed feedwater heaters, respectively.

8. TESTS FOR STEAM TURBINES AND TURBO-GENERATORS

Tentative,

The object of the tests may be to verify guarantees as to output, steam or heat consumption, or emergency governor operation. The Code provides rules for the tests of complete expansion and of other types of turbines.

Before tests are started, the two parties should enter into a written agreement regarding the object of the test, the methods to be employed in conducting the test, particularly where the Code permits the use of alternate methods, the magnitude of the corrections to be applied for variations from test conditions, and the method of applying these corrections; and upon the party who shall direct and carry out the test. Acceptance tests should be made within two months after the turbine enters commercial service. Preliminary tests may be run to determine whether the turbine is in suitable condition for the test, to check instruments, and to train personnel.

Conditions shall be caused to become as nearly constant as possible before the test begins and shall be so maintained during the test within limitations specified. No tests shall be made if the steam is not superheated where the initial pressure is measured.

Only accurate and reliable instruments shall be used, and these shall be calibrated both before and after the tests. Duplicate instruments shall be provided for those liable to breakage or failure. Only such observations are needed that apply and are necessary for the object of the test.

Condenser leakage during a test shall be determined when the condensate is measured, by silver nitrate, electrolytic conductance, or direct weight methods. The limit of permissible leakage with turbines over 1000 kw. is 0.3%.

Among the apparatus and instruments required for a performance test of a steam turbine or turbo-generator are: 1. A dynamometer (for a turbine alone); 2. Electrical instruments to measure the output of a turbo-generator; 3. Weighing or volumetric tanks for a condenser; 4. Water weighing and measuring tanks to an isolated boiler for a noncondensing turbine; 5. Bourdon gages; 6. Mercury manometers; 7. Thermometers or other temperature measuring instruments; 8. Speed indicators; 9. Clock or synchronized

watches; 10. Steam tables and charts (preferably Keenan's); 11. Nozzles or orifices for tests on certain types of turbines.

In general the rules of the A.I.E.E. governing the measurement of output of generators shall be followed in turbo-generator tests.

The duration of tests shall be: 1. Six hours in the case of measurement of feedwater to a boiler; 2. One hour for tests with weighed or measured condensate; 3. Seven consecutive sets of simultaneous readings for nozzle or orifice methods.

In tests involving the measurement of condensate, any leakage around pump glands or elsewhere must be carefully checked. The Code provides detailed instructions for methods of measuring electrical output, weight of condensate, leakage, and measurements of high and low pressures, temperatures, speed, and time.

HEAT CONSUMPTION of a turbine in B.t.u. per kw-hr. is expressed:

For a complete expansion, heat rate $= W_1(h_1 - h_4)/\mathrm{kw}_g$

For reheating without feedwater heating, heat rate $= \{W_1(h_1 - h_4) + W_3(h_3 - h_2)\}/\mathrm{kw}_g$.

For a regenerative turbine, heat rate

$$+ \{(h_8 - h_7) - \frac{144V(p_8 - ?}{778.6} \quad \frac{W_7}{2}$$

For a regenerative-reheating turbine, heat rate

778.6

where W_1 = weight of steam entering turbine; W_3 = weight of steam from reheater to turbine; W_6 = weight of feedwater discharged from final heater; W_5 = weight of condensate discharged from condenser; W_7 = weight of feedwater through any pump intermediate the condenser and final heater; h_1 = total heat of steam supplied; h_4 = heat of liquid at temperature corresponding to exhaust pressure; h_3 = total heat of steam returned from reheater; h_2 = total heat of steam to reheater; h_6 = heat in feedwater discharged from final heater; h_5 = heat of water of condensate; h_7 = heat of feedwater at inlet of pump; h_8 = heat of feedwater at outlet of pump; V = specific volume of 1 lb. of feedwater at mean temperature at pump, cu. ft. per lb.; p_7 = pressure at the inlet to the pump; p_8 = pressure at the discharge to the pump; kw_g = output measured at the generator terminals, kw.

All weights are in lb. per hr.; all heats in B.t.u. per lb. and all pressures in lb. per sq. in., abs.

Heat rates for bleeder and mixed pressure turbines are based on the energy from adiabatic heat drops of the various sections. See Code.

COMPUTATION OF RESULTS.—a. Complete Expansion Turbines may have their performances expressed alternatively as a *steam* or a *heat rate*. In the steam rate test, the turbine is charged with the net steam quantity supplied. In the heat rate test, the turbine is charged with the total heat supplied, and credited with the total heat of the water at the boiling temperature corresponding to the exhaust pressure.

b. Condensing Turbines Combined with Reheating have their performances expressed only as a heat rate, the turbine being charged with the initial total heat of the steam supplied plus the heat added by the reheater, and credited with the total heat of the water at the boiling temperature corresponding to the exhaust pressure.

c. Condensing Regenerative Turbines with Extraction for Heating Feedwater have their performances expressed: 1. As a heat rate, the turbine being charged with the total heat of the steam supplied, and credited with the total heat of the feedwater leaving the final heater, with an allowance should the temperature of the condensate be below that corresponding to the exhaust pressure, and also allowance for pump energy, or (2) as a steam rate in the same manner as for type *a*, the test being carried out with the heaters blanked off.

d. Condensing Turbines with Both Reheating and Extraction for Heating Feedwater have their performances expressed only as a heat rate, the turbine being charged with the initial total heat of the steam supplied plus the heat added by the reheater, and credited with the total heat of the feedwater leaving the final heater, with an allowance should the temperature of the condensate be below that corresponding to exhaust pressure, and also allowance for pump energy.

e. Non-condensing and Back-pressure Turbines have their performances expressed as a steam rate, the turbine being charged with the net steam quantity supplied.

f. Bleeder and Mixed-pressure Turbines have their performances expressed only as a heat rate, the turbine being charged with the sum of the calculated adiabatic heat drops of the quantities of steam at the respective flows.

Total steam chargeable to the turbine shall be only that which passes through the turbine throttle.

The Code gives complete directions regarding the corrections that may be applied to tests to reduce them to standard conditions and on how these shall be applied.

The following tabulation presents in abstract certain of the data and results that are shown in the complete Report of Tests in the Code, to which reference should be made for the complete form.

DATA AND RESULTS OF CONDENSING TURBINE TESTS

Item

Date of test..

Location..

Owner..

Builder..

Test conducted by..

Object of test..

Statement describing installation as regards type of turbine, condensing or non-condensing, bleeder or mixed-pressure (impulse or reaction), steam flow, number and arrangements of cylinders, reheating, regenerative feedwater heating (number and arrangement of feed heating stages), etc.

Specified operating conditions and guarantees for the particular turbine under test, together with a complete statement of the agreed corrections and whether each applies to steam or heat consumption.

Statement of conditions prevailing during the test including remarks as to test operating conditions, load at which test is made and its percent of maximum continuous rating, character of load, number of governor-controlled valves and extent of opening, disposition of any manually-operated valves and extent of opening, description of method of sealing shaft glands, stating origin of sealing steam, and disposition of leak-off steam.

10 Description of methods of measurement employed for the determination of steam quantities, output, and exhaust pressure, of methods of measurement differing from the rules prescribed in the Code, which methods have been agreed to by the parties to the test, alternative method employed where alternative methods are permissible, identification of calibration certificates and by whom made, and steam tables employed in the calculation of test results.

MEAN OBSERVATIONS DERIVED FROM LOG SHEETS

(All Calibrations Having Been Applied)

11 Duration of test.. hr.

12 Initial steam pressure.. lb. per sq. in., abs.

13 Initial steam temperature.. deg. F.

14 Barometric pressure as observed.. in. Hg

15 Vacuum at exhaust flange as observed by mercury columns.. in. Hg

16 Temperature of mercury columns.. deg. F.

17 Vacuum at exhaust flange, corrected.. in. Hg

18 Absolute pressure at exhaust flange.. lb. per sq. in., abs.

19 Temperature corresponding to the absolute exhaust pressure at exhaust flange (from steam tables).. deg. F.

20 Temperature of condensate leaving condenser.. deg. F.

21 Temperature of feedwater leaving final heater.. deg. F.

22 Pressure of feedwater leaving last heater.. lb. per sq. in., abs.

23 Speed.. r.p.m.

24 Reheater observations at test load including conditions at inlet and outlet, of absolute pressure, temperature, B.t.u. per lb., quantity in lb., and pressure drop through reheaters.

MEASUREMENT OF STEAM

25 Quantity of steam abstracted between initial admission and reheater (if any).. lb. per hr.

26 Total condensate or feedwater quantity as measured.. lb. per hr.

27 Additions to or deductions from the measured quantity, such as condenser leakage, condensate pump leakage, steam from ejectors, make-up water, steam from auxiliaries, sealing water, etc.. lb. per hr.

28 Total net condensate or steam conumption (Item 26 ± algebraic sum of Item 27).. lb. per hr.

MEASUREMENT OF OUTPUT

29 Mean output at electrical terminals..........kw......, ...kva. at.... power factor

30 Mean output at turbine coupling.. kw.

DATA AND RESULTS OF CONDENSING TURBINE TESTS (*Continued*)

Item

CORRECTIONS

31 Corrections to reduce the measured steam (or heat) consumption to that conditions, including corrections for: *a*, Initial steam pressure; *b*, initial steam temperature; *c*, exhaust pressure; *d*, superheat of steam entering turbine from reheater; *e*, pressure drop between turbine outlet and inlet from reheater; *f*, feedwater discharge temperature; *g*, speed; *h*, power factor; *j*, load correction.
Aggregate correction divisor ....

STEAM CONSUMPTION

33 Steam consumption under test operating conditions:
a, at electrical terminals .... lb. per kw-hr.
b, at turbine coupling .... lb. per kw-hr.
34 Steam consumption corrected to specified operating conditions except for load correction:
a, at electrical terminals .... lb. per kw-hr.
b, at turbine coupling .... lb. per kw-hr.
35 Steam consumption with load correction applied:
a, at electrical terminals .... lb. per kw-hr.
b, at turbine coupling .... lb. per kw-hr.

HEAT

36 Total heat of initial steam supplied .... B.t.u. per lb.
37 Total heat of water at temperature corresponding to exhaust pressure .... B.t.u. per lb.
38 Total heat of condensate at measured temperature .... B.t.u. per lb.
39 Total heat of feedwater at discharge from final heater at the measured temperature and pressure .... B.t.u. per lb.
40 Total net heat supplied under the operating conditions of the test .... B.t.u. per hr.
41 Heat consumption under the operating conditions of the test:
a, at electrical terminals .... B.t.u. per kw-hr.
b, at turbine coupling .... B.t.u. per kw-hr.
42 Heat consumption corrected to the specified operating conditions except for load correction:
a, at electrical terminals .... B.t.u. per kw-hr.
b, at turbine coupling .... B.t.u. per kw-hr.
43 Heat consumption with load correction applied:
a, electrical terminals .... B.t.u. per kw-hr.
b, at turbine coupling .... B.t.u. per kw-hr.

TESTS OF EMERGENCY GOVERNOR OPERATION

44 Turbine speed at which emergency governor operates .... r.p.m.
45 Emergency speed of turbine. The extreme speed of the turbine when accelerated from below normal speed with internal pressure corresponding to that of the maximum continuous rating, not exceeding specified exhaust pressure, no load, and without main governor control .... percent
46 Percentage overspeed of the turbine at which the emergency governor operates .. percent
47 Percentage emergency overspeed reached by the turbine when accelerated with specified initial and exhaust pressure, no load, and without main governor control .... percent

The Code also gives tables and methods for reporting tests of bleeder turbines and mixed-pressure turbines, which are not abstracted here. Useful appendices to the Code give numerical examples of test data and corrections, and show in detail how the final results are computed. These include: 1. A simple condensing turbine; 2. A turbine with one stage of reheating and three stages of feedwater heaters; 3. A condensing bleeder turbine with steam extracted for industrial or other purposes at two pressures; 4. A mixed-pressure turbine with two admission points for lower pressure steam than at the throttle.

9. TEST CODE FOR RECIPROCATING STEAM ENGINES

Approved, 1935

OBJECT AND SCOPE of test should be recorded as the verification of manufacturer's guarantees, or determination of: *a.* thermal-economy characteristics of the engine (heat rate, steam rate, or both); *b.* capacity in brake or indicated horsepower or kilowatt output; *c.* verification of special guarantees or other data, which are subjects of written agreement by both parties to the test.

DEFINITIONS.—**Throttle Pressure** is average pressure on boiler side of, and directly preceding, throttle valve during steam admission period, *i.e.*, from start of admission to cut-off.

Exhaust Pressure is average pressure at outlet side of, and adjacent to, exhaust nozzle during exhaust period, *i.e.*, from dead-center point immediately after release to beginning of compression.

Engine Output is power developed, expressed in indicated or brake or shaft horsepower. Output at coupling may be expressed in kilowatts.

Electrical Output is net output of generator in kilowatts at generator terminals. For a direct-current generator this is (corrected terminal volts × amperes) ÷ 1000. For an alternating current generator, it is {(electrical power output of generator, kw.) − (excitation power separately supplied, kw.) − (power for ventilation separately supplied, kw.)}.

PREPARATIONS.—The engine should be put in proper operating condition by the manufacturer; all unused connections should be blanked off or connections so broken that outlets are under constant supervision.

Cylinder Dimensions should be taken when engine is cold, and suitable corrections applied to conform with mean working temperature, which is the mean of saturation temperatures corresponding to throttle and exhaust pressures. Average diameter of much-worn cylinders should be found. Clearance volumes need not be found unless they are guaranteed or test conditions require them.

Leakage of pistons and valves is included in total steam consumption and need not be found. Valve stem and piston rod leakage should be measured and included in steam consumption. Condenser leakage must be determined if steam is measured by weighing or measuring condensate (see p. 16–31). Maximum permissible condenser leakage, in percentage of flow at rated load as stated in steam rate guarantee, is:

Engine capacity, Hp......	Under 700 (500 kw.)	700–1400 (500–1000 kw.)	Over 1400 (1000 kw.)
Maximum leakage, percent	0.5	0.4	0.3

Preliminary Tests may be run to determine whether engine is in suitable condition for test, to check all instruments, and to train personnel.

DURATION OF TEST, when surface condenser method is used, shall cover not less than four successive 15-minute records, during which the uncorrected steam rate for each period shall check within 3.0%. When steam consumption is measured by feedwater supplied to a boiler, duration of test shall be 8 hr. Each test should be preceded by a period adequate to establish constant conditions. In capacity tests, operating conditions including output, initial and exhaust pressures, quality or superheat, vacuum and speed shall be held sensibly constant for not less than 15 min.

Constant Conditions shall be maintained as nearly as possible throughout preliminary and test periods. Steam pressure may be adjusted by proper boiler control or by throttling (with correction for changed superheat). Superheat may be adjusted by boiler control. Reduction of superheat by spraying water into steam line is not permissible. Exhaust pressure can be raised by admitting air to condenser or suction line of air pump. Back pressure in non-condensing engines may be raised by throttling, or decreased by using a test condenser or by providing free atmospheric exhaust. Constant load may be obtained by limiting governor or cut-off. The load on engine-generator sets can be adjusted by changing electric circuit conditions, terminal voltage being held within ±5% of specified value.

INSTRUMENTS AND MEASUREMENTS.—The following instruments are required for a heat-rate acceptance or performance test:

For an engine running alone, a dynamometer; for an engine-generator set, instruments for measurement of net kilowatt output, including that for excitation and ventilation; for an engine exhausting to a surface condenser, weighing tanks and scales, or volumetric measuring tanks, and suitable instruments to determine condenser leakage; for an engine exhausting to a jet condenser, to a heating line, or simply non-condensing, water weighing or measuring tanks and boiler isolated for a feedwater test, orifices or nozzles to measure steam quantity as water in condenser tests, upon agreement in writing by the parties to the test.

For all classes of engines, the following are necessary: Steam engine indicators to measure power developed in cylinder, throttle and exhaust pressures (by agreement in writing, a Bourdon or deadweight pressure gage may be used to determine throttle pressure, and mercury manometers to determine vacuum or exhaust pressure); calipers and scales to determine dimensions; temperature measuring instrument to determine steam temperature on boiler side of throttle valve; if steam is wet, a throttling calorimeter to determine steam quality; thermometers to determine exhaust temperature, temperature of mercury columns, manometers, barometers, exposed stems of glass high-temperature thermometers, temperature of water in volumetric measuring tanks, and tem-

perature of condensate; clock, or synchronized watches or signaling system from a central point; steam tables and charts; planimeters to measure indicator cards; graduated scales for gage glasses of boilers and hotwells. All instruments shall be calibrated before and after test.

Indicators to measure power developed in the cylinder need be used only when performance guarantees are based on I.Hp., or when it is desired to measure points of cut-off, etc.

Observations of instruments shall be made and indicator cards taken from each end of each cylinder at least every 15 min. when conditions are uniform, and oftener if possible. Each indicator card shall be marked with number, date, time, scale of spring and end of cylinder.

Indicated Horsepower (I.Hp.) shall be calculated for each working end of cylinder by

$$\text{I.Hp.} = (pLAn)/33{,}000,$$

where p = mean effective pressure, lb. per sq. in.; L = length of stroke, ft.; A = area of piston, minus area of piston- or tail-rod, sq. in.; n = rev. per min. Total horsepower of a cylinder is the sum of horsepower developed in the two ends.

Brake Horsepower (B.Hp.) shall be measured by dynamometer or other loading device. With absorption dynamometers, such as water brakes, the operating fluid shall enter and leave in substantially a radial direction. In electric absorption dynamometers, admission and emission of cooling air shall be sensibly radial in direction. Dynamometers shall be examined before and after test and deadweight determined. In absorption dynamometers, effective-radius arm shall be measured with an accuracy within 1 part in 2000; weighing means shall be effective within 2 parts in 1000. Dynamometer readings shall be taken with such frequency that average of all observations will not differ from average of all alternate observations by more than 1%. A test shall be deemed unsatisfactory if a condition exists that gives variations in dynamometer readings exceeding 2%. Brake horsepower is calculated by

$$\text{B.Hp.} = 2\pi LFn/(550 \times 60) = 6.2832\, LFn/33{,}000$$

where F = net force on brake arm, lb. (gross weight − tare weight on scales); L = horizontal distance between center of shaft and bearing point at end of brake arm, ft.; n = rev. per min. of brake shaft.

Output at coupling may be expressed in kilowatts (1 kw = 736.5 ft.-lb. per sec.).

ELECTRICAL OUTPUT.—For a 3-phase generator with grounded neutral, power output of main unit shall be measured by 3-wattmeter method; with isolated neutral, power output shall be measured by either 2-wattmeter or 3-wattmeter method. For single-phase generators, power output shall be measured by the 1-wattmeter method. Power output of direct-current generators shall be measured by the direct-current voltmeter-ammeter method. Single-phase wattmeters or watt-hour meters are preferable for measuring polyphase loads at high voltage. If generator leads are connected solidly to power transformers, and it is necessary to connect instruments on secondary side of transformers, the calculated losses in transformers and leads shall be added to transformer output to give generator output.

Calculations.—Average kilowatt output of any run is the sum of the corrected average reading of wattmeters.

$$\text{Coupling or shaft kw.} = \{(\text{kw. at generator terminals})/(\text{generator efficiency})\}$$

$$\text{B.Hp or shaft Hp.} = 1.34 \times (\text{coupling kilowatts}).$$

Generator efficiency shall be as defined in the current A.I.E.E. rules.

STEAM QUANTITY MEASUREMENT always shall be in terms of liquid water, either after it is condensed or before it has evaporated. Measurements shall be made as follows:

a. Measurements of condensate from a surface condenser, including steam discharged by drips and glands of engine proper. *b.* For engine with jet condenser, or a non-condensing or back-pressure engine, measurement of feedwater delivered to the boiler, whose only outlet is to the engine under test, after making any necessary corrections for leakage, water level, etc.; accuracy in quantity of steam so determined cannot be expected within 3%. *c.* If engine is steam jacketed, all steam condensed in jackets and not passing directly through the cylinder, but discharged to traps or drains, shall be recovered and weighed; this weight is to be added to weight under (*a*) to determine total quantity of steam chargeable to engine.

The steam shall be measured by one of the following methods: *a.* Actual weighing by means of tanks and scales. *b.* Calibrated volumetric measuring tanks. *c.* By agreement in writing, and with understanding that accuracy cannot be expected within 2%, a Venturi tube, nozzle or orifice, with means for accurate measurement of pressure differences, may be used for measurement of condensate or feedwater, if supplied at constant rate.

Water level in condenser hotwell or in boiler should be the same at beginning and end of test. If any variation exists, the necessary corrections shall be made.

Condenser Leakage shall be determined by one of the following methods: *a.* Silver nitrate method, suitable only for circulating water containing an appreciable amount of salt; preferred

(*Continued on page 16–30*)

REPORT OF TEST OF RECIPROCATING STEAM ENGINES

Item

DESCRIPTION, DIMENSIONS, ETC.

Type of engine (simple or multiple-expansion, Corliss, uniflow, etc.):
(a) Number of cylinders............; (b) Condensing or non-condensing............
(c) With or without jackets............
Class of service (mill, electric, marine, etc.)............

Cylinders:	First	Second	Third
(a) Diameter, in............	......	......	......
(b) Cylinder ratio (based on net piston displacement) .. 1 to	......	......	......
10 Pistons:			
(a) Diameter, in............	......	......	......
(b) Diameter of piston-rods, in............	......	......	......
(c) Diameter of tail-rods, in............	......	......	......
(d) Stroke, in............	......	......	......

Clearance volume in percent of piston displacement as specified by manufacturer:
(a) Head end............percent; (b) Crank end............percent
Horsepower constant (stroke × net piston area ÷ 33,000):
(a) Head end............ (b) Crank end............
Auxiliaries:
(a) Valves, type............ (b) Governor, type............
(c) Condensing equipment:
Type............ Make............ Rated capacity............
(d) Pumps (direct or independently driven):
Oil, type............ Jacket, type............ Reheater, type.......size

14 Driven unit:
(a) Type............ (b) Size............ (c) Rating............
(d) Capacity of generator or apparatus consuming power of engine............Hp. or kw.

SPECIFIED OPERATING CONDITIONS AND GUARANTEE

15 Manufacturer's rating:	I.Hp.	Kw. of generator	Power factor	Kw. rating at engine shaft
(a) Rated load............	....	....	....	....
(b) Most economical rating............	....	....	....	....

Rated speed at rated load............r.p.m.
Ratio of no-load speed to rated speed............
Throttle pressure............lb. per sq. in., abs.
Throttle steam temperature............deg. F.; or quality............percent
Steam consumption guaranteed by manufacturer:

Output	Absolute exhaust pressure, lb. per sq. in.	Steam consumption, per I.Hp.-hr. or lb. per B.Hp.-hr. or lb. per kw.-hr.
....	....	....

20 Heat consumption guaranteed by manufacturer:

Output	Absolute exhaust pressure, lb. per sq. in.	Heat consumption, B.t.u. per I.Hp.-hr. or B.t.u. per B.Hp.-hr. or B.t.u. per kw.-hr.
....	....	....

21 Weighted steam (or heat) consumption guaranteed:
(Total column 4/Total column 3)

.per I.Hp.-hr.
.per B.Hp.-hr.
...per kw.-hr.

			4
Output, percent of rated load	Steam (or heat) consumption per unit output, lb. or B.t.u. per I.Hp.-hr., or per B.Hp.-hr., or per kw.-hr.	Factor	Product of column (2) and (3)
	Total......................	....	....

AGREED CORRECTIONS

22 Agreed corrections to steam flow (as tabulated below or from curves identified below) as applied to (1) steam rate or (2) heat rate:

Variable	Curve No.	Value, percent per unit	Unit
Output percent of rated load			
(a) Throttle pressure............	....	....	1 lb. per sq. in.
(b) Throttle steam temperature or quality	....	....	10° F. or 1% moisture
(c) Exhaust pressure............	....	....	1 in. Hg
(d) Back-pressure correction............	....	....	1 lb. per sq. in.
(e) Power factor............	....	....	0.05 variation
(f) Load correction............Determined from plot of test results			

REPORT OF TEST OF RECIPROCATING STEAM ENGINES (*Continued*)

Item

METHODS OF TEST

23 Description of methods other than those prescribed by the Code. . . . .

Mean Observations Derived from Log Sheets, All Calibrations Having Been Applied

24 Run number. . . . .
25 Duration of test. . . . . .hr.
26 Speed. . . . . .r.p.m.; (*a*) Piston speed. . . . . .f.p.m.
27 Test made at. . . . . I.Hp., B.Hp., Kw.;. . . . . .percent of rating
28 Throttle pressure. . . . . .lb. per sq. in. (abs.)
29 Throttle steam temperature. . . . . .deg. F., or quality at throttle. . . . . .percent
30 Barometer pressure as observed. . . . . .in. Hg at 32° F.
31 Absolute exhaust pressure:
(*a*) From steam pipe diagram. . . . . .lb. per sq. in. (abs.)
(*b*) As measured by mercury columns . . . . .in. Hg
(*c*) As shown by Bourdon gage. . . . . .lb. per sq. in. (abs.)
32 Temperature at exhaust flange (if observed) . . . . .deg. F.
33 Absolute pressure in jackets. . . . . .lb. per sq. in. (abs.)
34 Temperature of condensate. . . . . .deg. F.

Item		1st Cyl.	2d Cyl.	3d Cyl.
35	Indicator diagrams:			
	(*a*) Nominal cut-off, percent.	....	....	....
	(*b*) Nominal release, percent.	....	....	....
	(*c*) Nominal compression, percent.	....	....	....
	(*d*) Mean effective pressure, lb. per sq. in.	....	....	....
36	Indicated horsepower:			
	(*a*) Head end, I.Hp.	....	....	....
	(*b*) Crank end, I.Hp.	....	....	....
	(*c*) Total, I.Hp.	....	....	....
	(*d*) Total for engine.	....	....	....

37 Brake horsepower output of engine. . . . . .B.Hp.
38 Mechanical efficiency . . . . .percent
39 Net electrical output (for engine-generator set). . . . . .kw.
40 Thermal data:
(*a*) Total heat of steam at throttle (h_1). . . . . .B.t.u. per lb.
(*b*) Entropy of steam at throttle. . . . . .B.t.u. per deg. F. per lb.
(*c*) Total heat of liquid at exhaust pressure (h_f). . . . . .B.t.u. per lb.
(*d*) Heat supplied per lb. of steam ($h_1 - h_f$) . . . . .B.t.u. per lb.
(*e*) Total heat per lb. after isentropic expansion to exhaust pressure (h_3). . .B.t.u. per lb.
(*f*) Heat available for work ($h_1 - h_3$) . . . . .B.t.u. per lb.
41 Total condensate or feedwater quantity, as measured, and without allowances for leakages . . . . .lb. per hr.

Item	42 Additions to and deductions from measured quantity in Item 41:	Lb. per hr. Plus	Lb. per hr. Minus
	(*a*) Condenser leakage.	......	......
	(*b*) Condensate pump leakage.	......	......
	(*c*) Make-up water.	......	......
	(*d*) Steam from auxiliaries.	......	......
	(*e*) Piston rod and valve stem leakage.	......	......
	(*f*) Condensate from steam jackets.	......	......
	(*g*) Any other leakages.	......	......
	Total.	......	......

Total net steam consumption (Item 41 ± algebraic sum of Item 42). . . . . .lb. per hr.
Steam consumption under test operation conditions:
(*a*) Based on I.Hp. (Item 43 ÷ Item 36*d*). . . . . .lb. per I.Hp.-hr.
(*b*) Based on B.Hp. (Item 43 ÷ Item 37) . . . . .lb. per B.Hp.-hr.
(*c*) Based on net kw. (Item 43 ÷ Item 39) . . . . .lb. per kw.-hr.
45 Heat consumption under test operating conditions:
(*a*) Based on I.Hp. (Item 43 × Item 40*d*/Item 36*d*) . . . . .B.t.u. per I.Hp.-hr.
(*b*) Based on B.Hp. (Item 43 × Item 40*d*/Item 37). . . . . .B.t.u. per B.Hp.-hr.
(*c*) Based on net kw. output (Item 43 × Item 40*a*/Item 39) . . . . . .B.t.u. per B.Hp.-hr.

Item	46 Corrections to reduce the measured steam flow or heat consumption to that under specified conditions:	Percent Plus	Percent Minus	Divisor
	(*a*) For throttle pressure.			
	(*b*) For throttle steam temperature or quality.			
	(*c*) For exhaust pressure			
	(*d*) For power factor.			
	(*e*) For load correction.			
	Total.			

(*Continued on following page*)

REPORT OF TEST OF RECIPROCATING STEAM ENGINES (*Continued*)

Item

METHODS OF TEST.- 'ontinued

47 Aggregate correction divisor (product of individual divisors)
48 Steam consumption corrected to specified operating conditions:
- Item 47)........................lb. per
ıo ÷ Item 47)......................lb. per B.Hp.-hr.
ıet kw. output (Item 44c ÷ Item 47)..................lb. per kw.-hr.
49 ›nsumption corrected to specified operating conditions:
Based on I.Hp. (Item 48a × Item 40d)....................B.t.u. per I.Hp.-hr.
Based on B.Hp. (Item 48b ×
Based on net kw. output (Item 48c
50 ency under specified operating conditions:
(a) Based on I.Hp. (2545 ÷ Item 49a)....................................percent
(b) Based on B.Hp. (2545 ÷ Item 49b)....................................percent
(c) Based on net kw. output (3412 ÷ Item 49c) ....percent
Engine efficiency under specified operating conditio
(a) Based on I.Hp. {2545 ÷ (Item 40f × Iten .percent
(b) Based on B.Hp. {2545 ÷
(c) Based on net kw. output

URE
sures above 25 lb. per s
gages or their equivalent m
shall be of such size
they may not have

by steam engine indicator for pres-
calibrated Bourdon or dead weight
shall not be used. The Bourdon
of the total scale easily can be
% of the absolute pressure.

Throttle Pressure shall be measured
diameter of throttle valve. Average thro
ordinate of a pressure curve plotted against
from end of stroke to point of cut-off. Thi
indicator operated from reducing motion o

indicator within, if possible, one
all be obtained by determining
ment during steam admission period
obtained from a card drawn by an
pipe on boiler side of

lemiscus or capillarity; c. Gravity to reduce reading to
ıtion of instrument had International Standard value
c.; d. Difference in elevation between the barometer and the mercury
barometer reading shall be corrected for this difference, having first been
capillarity and gravity.

Exhaust ... e by means of a steam engine indicator attached to exhaust pipes at specified point and operated from reducing motion of engine. Exhaust pressure shall be obtained by determining mean ordinate of a pressure curve plotted against piston position during steam-exhaust stroke, *i.e.*, from a piston displacement diagram. Exhaust period shall extend from dead center point to instant of closure of exhaust valve or port, as determined with exhaust pipe diagrams.

Temperature Measurements shall be made by mercury-in-glass thermometers with etched stems. Thermometer wells (see p. 3–08) may contain fused solder, sodium nitrate or mercury for higher temperature ranges, or oil if temperature is so low as not to evaporate the oil.

For high temperatures, measurements may be made, under written agreement, by an electric resistance thermometer or an iron-constantan or other suitable thermocouple with nul-reading potentiometer. Important temperature readings should be the mean of readings taken at two points close together. Discrepancies in excess of 1% above the ice point should be investigated.

QUALITY OF STEAM shall be determined by throttling calorimeter, when the engine operates without superheat; calorimeter sampling tube to be inserted in steam pipe on boiler side of, and adjacent to, throttle valve.

SPEED MEASUREMENTS may be made by: *a*. Hand counter; *b*. Hand tachometer of mechanical weight type; *c*. Belted or positively-driven tachometer of mechanical weight type; *d*. Frequency meter connected to generator circuit; *e*. Electric tachometer; *f*. Mechanical integrating counter; all except *a* to be calibrated prior to test.

For steam or heat consumption tests involving measurement of brake or indicated horsepower speed shall be determined by a mechanical integrating revolution counter, positively geared or con-

nected to some portion of engine or driven machine. Average speed per unit of time shall be determined from total number of revolutions during entire test. Time of counter observation at beginning and end of test shall be accurate within 1 sec.

MAXIMUM VARIATION IN OPERATING CONDITIONS shall be as follows: *a.* Throttle pressure, ±8% of absolute pressure. *b.* Initial steam superheat ±25° F. *c.* Throttle quality, 2% from guarantee. *d.* Exhaust pressure not exceeding 2 lb. per sq. in., abs. Actual absolute pressure not more than 150%, nor less than 75%, of absolute pressure corresponding to specified exhaust pressure. From 2.01 to 5 lb. per sq. in., abs., ±10% of absolute pressure. Above 5 lb. per sq. in., abs., ±5% of absolute pressure. *e.* Speed ±5%. During any one test, not more than ±2% of average speed of test. *f.* Power factor, from unity P.F. to 5% below specified value. *g.* Load fluctuation ±5%. *h.* Output (mean) ±5% from specified value.

DATA AND RESULTS should be reported on a suitable form. The form on p. 16–28 is suggested in the code, but can be modified to suit conditions.

10. TEST CODE FOR CONDENSING APPARATUS

Approved, 1926

This code covers surface-, low-level jet-, barometric-, and ejector-condensing apparatus.

OBJECT of test usually is: *a.* To determine fulfilment of manufacturer's guarantee; *b.* To determine performance of the apparatus in part or as a whole.

INSTRUMENTS AND APPARATUS required comprise: *a.* Mercurial barometer. *b.* Mercurial columns for measuring vacua, the smallest scale graduations being not greater than 0.1 in., with vernier attachment reading to 0.01 in. *c.* Bourdon gages for measuring pressures too great for mercury manometer tubes. *d.* Thermometers graduated to 1/2 deg. F., with maximum scale readings from 32 to 150 deg. F. for determining temperatures of circulating water, condensate and vapors; thermometers graduated by degrees, with maximum scale readings from 32 to 250 deg. F. for determining steam temperature. *e.* Tanks and platform scales for measuring water, or water meters calibrated in place under conditions of use. *f.* Appliances for tests of auxiliary pumps, such as steam engine indicators, electrical instruments, speed indicators, etc. *g.* Air measuring devices, gasometer or calibrated orifices.

Amount of Circulating Water may be estimated by the heat balance method, using the equation

$$F_w = h_c w_s/(t_2 - t_1)$$

where F_w = circulating water, lb. per hr.; h_c = B.t.u. per lb. of steam absorbed by condenser; w_s = steam discharged to condenser, lb. per hr.; t_1, t_2 = respectively, inlet and outlet temperatures of circulating water.

Location of Thermometers within the steam space should be such that they are shielded from direct impact of steam currents and from radiation from cold tubes. It usually is best to insert bare thermometers through rubber stoppers. Thermometers measuring temperature of discharge water should be located as far as practicable, but not more than 100 ft., from the condenser. At least three thermometers uniformly spaced should be used to determine the temperature of discharge circulating water.

Inlet Pressure shall be measured by mercury columns accurate to 0.01 in. mercury, corrected for all observable causes of error. Each column shall be mounted as near as practicable to the pressure hole in the conduit or casing. Extreme care shall be exercised to make piping and joints between the column and pressure hole air-tight. Each column must be so mounted as to be as free as possible from vibration.

If conduit connecting turbine exhaust nozzle and condenser inlet nozzle is approximately straight, inlet pressure shall be measured as far as possible from turbine exhaust flange (toward the condenser) and at a distance from the condenser inlet flange (toward the turbine) not to exceed three diameters of a circle of area equal to that of the inlet conduit. Each column shall be connected to a static pressure hole normal to the inner wall of the conduit and not smaller than 1/2-in. standard iron pipe size. There shall be approximately one such hole and column for each 16 sq. ft. of inlet conduit area, but not less than 2 nor more than 6 columns. Pressure holes shall be distributed uniformly around the periphery of the conduit.

If there is no straight conduit between turbine and condenser, pressure holes shall be located within 12 in. of the plane of the face of the condenser inlet flange at that plane where steam flow is most likely to be smooth.

Readings of all Mercury Columns, corrected to inches of mercury at 32 deg. F., shall be averaged and subtracted from a similarly corrected reading of a mercury barometer to give absolute exhaust pressure. Correction must be made for any difference in elevation of barometer and mercury columns.

PREPARATIONS.—Dimensions and physical condition of condenser and auxiliaries should be ascertained. All foul tube surfaces, both inner and outer, should be thoroughly cleaned. Quantity of leakage from water to steam spaces of surface condensers should

be determined by measuring water discharged from condensate pump with all steam shut off from entering condenser, the normal quantity of circulating water being in use and all other conditions normal. Leakage so revealed should be rectified before proceeding with test, unless precluded by the object in view.

Leakage test should continue for at least one hour after normal conditions are established. Leakage so determined should be deducted from total condensate measured on main test. If leakage exceeds 1% of the normal capacity of condenser, test should be stopped until it is reduced to the stated figure. The average of a leakage test made before and after the main test may be considered as condenser leakage.

By filling a surface condenser shell and its connections with water at a temperature approximately that of surrounding air, and under slight pressure, air leaks will be revealed. The quantity of air removed from the condenser shall be measured by gasometer or other approved apparatus. The tightness of steam valves and pistons of steam-driven auxiliaries may be ascertained by standing leakage tests.

OPERATING CONDITIONS, such as speed of auxiliaries, degree of vacuum to be carried, quantity of circulation or injection water needed, should be determined prior to beginning of main test; these conditions should prevail throughout the test. Since condenser performance depends on amount of air leakage, which the condenser manufacturer does not fully control, a definite air handling capacity of the air-removing apparatus must be assumed. Separate or shop tests of such apparatus often are desirable.

STARTING, STOPPING AND DURATION of a condenser test should follow the same rules that govern a steam engine or turbine test.

Surface Condenser Tests should determine the quantity of steam condensed by collecting and weighing the condensate and correcting the total as measured for leakage.

Jet Condenser Tests, including those of the barometric type, may be made by measuring feedwater consumption and correcting for drips, leakage of boilers and piping, and for steam which is not supplied to the condenser.

Such steam preferably should be taken from isolated boilers. It may be determined approximately by steam meter measurement if the meter is calibrated in place and under conditions of use, or by calculation of the heat given to the injection water if determined.

Circulating water may be measured by discharge into an open tank with a V-notch or rectangular weir, or by an orifice inserted in a discharge pipe. These indirect methods must be considered as only approximate.

Quantity of steam consumed by auxiliaries may be found by connecting them to a surface condenser and measuring the condensate, or by water glass tests made at a time when they can be operated independently and preferably taking steam from a single boiler.

CALCULATION OF RESULTS.—**Heat Supplied to Condenser** per hour depends on heat content of steam as delivered to condenser. Heat supplied to condenser is heat supplied to prime-mover exhausting into the condenser, less that converted into work and that lost by generator losses, friction, radiation and other wastes not readily determined. It is expresed by the following formulas (where condenser is bolted direct to turbine exhaust opening):

For a simple non-bleeding, non-reheating turbine

$$h_x = h_t - (3413/f_s\, e_{me}) \quad . \; . \; . \; . \; . \; . \; . \; . \; . \; . \quad [1]$$

For a reheat, non-bleeder turbine

$$h_r = h_t + (h_2{}^1 - h_1{}^1) - (3413/f_s\, e_{me}) \quad . \; . \; . \; . \; . \; . \; . \; . \quad [2]$$

in which the expression ($h_2{}^1$ — is repeated for each stage of reheating.

For a bleeder non-reheat turbine

$$h_x = (h_t\, F_d/X) - (3413\ K \qquad {}_1 h_1 + \qquad + \qquad {}_i h_n)/X\} \qquad [3]$$

For a bleeder and reheat turbine

$$h_x = [\{(h_t\, F_d/X) + (F_d - w)(h_2{}^1 - h_1{}^1)\}/X] - (3413\ K/e_{me}\, X)$$
$$\{(w_1\, h_1 + w_2\, h_2 + \; . \; . \; . \; w_n\, h_n)/X\} \quad . \; . \; . \; . \; . \; . \quad [4]$$

where h_x = B.t.u. per lb. of steam at prime-mover exhaust; h_t = B.t.u. per lb. of steam at prime-mover throttle; F_s = steam supplied to throttle of prime-mover, lb. per kw.-hr.; e_{me} = mechanical and electrical efficiency of prime mover complete including all losses; $h_2{}^1$ = B.t.u. per lb. of steam at point of re-entrance to turbine after reheating; $h_1{}^1$ = B.t.u. per lb. of steam after partial expansion in turbine, at point of extraction for reheat. F_d = total steam supplied to throttle of prime mover, lb. per hr.; $w_1, w_2 \; . \; . \; . \; w_n$ = respectively, steam bled from first, second and nth bleeder point of turbine, lb. per hr.; w = total steam bled from turbine at all bleeding points between throttle and point of reheating, lb. per hr., in a bleeder and reheat turbine; $h_1, h_2 \; . \; . \; . \; h_n$ = B.t.u. per lb. of steam at first, second and nth bleeder point of the turbine; K = load on prime-mover, kw.; $X = \{F_d - (w_1 + w_2 \; . \; . \; . + w_n)\}$.

DATA AND RESULTS OF CONDENSING-APPARATUS TESTS
(Abridged)

Item

DESCRIPTIONS AND DIMENSIONS, ETC.

Unit number...

How long in service?...

When were tubes last cleaned (surface type)?...

Condensing steam from........ No........ size....... make..... type (horizontal or vertical) turbine or engine...

Type condenser (horizontal or vertical—jet or surface):

(a) Material, shape and overall dimensions of shell...

(b) Weight of condenser, empty........ (c) including circulating water........ lb.

(d) Vacuum breaker—type and description...

10 Distance from prime-mover exhaust flange to condenser exhaust opening; size of connecting pipe; fittings and valves in same...

Exhaust relief valves: Size, make and location...

Source of supply and analysis of circulating or injection water...

Surface Condenser

(1) Standard type, water inside tubes; (2) Water-works type, water outside tubes.

Number of passes (water, if type 1) (steam, if type 2)...

Amount of effective cooling surface (including air cooler but excluding tube head area and internal heater) measured (type 1) on outside of tubes; (type 2) on inside.. sq. ft.

Amount of effective heating surface in internal heater.................... sq. ft.

Amount of effective cooling surface in air cooler.................... sq. ft.

Diameter, gage, material and length of tubes between heads...

Number of tubes in passes..........; 1st......; 2nd......; 3rd......; 4th......

Dimensions, shape and material of tube head, and total area, sq. ft., of each tube head in steam space (before drilling for tubes)...

Number, material and total thickness of support plates...

Tube-head area in contact with steam.................... percent

Condition of tube surface (as to cleanliness)...

Low-head Jet, Barometric or Ejector Condenser, 3/4 Parallel-flow or Counter-current

Approximate volume of space available for condensation.................... cu. ft.

Type of water distributor (jet or gravity)...

Number and type of jets...

Circulating Pump of Surface Condenser, or Removal (or Tail) Pump of Low-head Jet Condenser, or Injection Pump of Barometric or Ejector Condenser

Type, horizontal or vertical, centrifugal or reciprocating...

Number of stages if centrifugal—single or duplex—single or double suction...

Type and r.p.m. of drive (engine, turbine or motor)...

Rated horsepower, dimensions of cylinders and stroke of reciprocating apparatus.......

Size of all pipe openings...

Air-vapor Removal Apparatus

Reciprocating Dry Vacuum Pump

Type (horizontal or vertical) number of stages...

Type of drive (steam cylinder or motor)...

Size: (a) Diam. steam cylinder and stroke.......in. (b) Diam. air cylinder, 1st stage ....in. (c) Diam. air cylinder, 2nd stage.......in. (d) Diam. of piston rods, air cylinders........in. (e) Diam. of piston rods, steam cylinders........in.

34 Actual air piston displacement per revolution.................... cu. ft.

35 Clearance, in percent of air piston displacement...

Hydraulic Type

36 Type and trade name...

37 Size, including supply pump if separate...

38 Type of drive (engine, turbine or motor).................... I.Hp. or B.Hp. and speed

39 Size of all pipe openings...

Steam-jet or Ejector Type

40 Type, trade name, number of stages...

41 Number and sizes of precooler, intercooler and aftercooler (jet or surface); amount of surface in each, sq. ft...

42 Size of steam and water connections...

43 Are single or multi-nozzles used in ejectors?...

Condensate Pump

44 Type—horizontal or vertical—number of stages...

45 Size of suction and discharge openings...

46 Type of drive (engine, turbine or motor).................... I.Hp. or B.Hp. and speed

(Continued on following page)

CONDENSING APPARATUS TESTS (*Continued*)

Item

TEST DATA AND RESULTS

Duration of test . . . . . . hr.

Steam Supply

48 Load on prime-mover or other apparatus exhausting to condenser . . . . . . I.Hp. or kw.
49 Heat equivalent of work per hour . . . . . . B.t.u.
50 Steam pressure at main throttle . . . . . . lb.
51 Quality of steam at main throttle . . . . . . percent moisture or deg. superheat
52 Quantity of steam entering condenser . . . . . . lb.
53 Quantity of steam at throttle of each auxiliary, if required . . . . . . lb.

AVERAGE PRESSURES AND TEMPERATURES

54 Barometric reading as observed . . . . . . in. Hg . . . . . . deg. F.
(*a*) Temperature of barometer . . . . . . deg. F.
55 Barometric pressure as corrected . . . . . . in. Hg at 32° F.
56 Temperature of atmospheric air . . . . . . deg. F.
(*a*) Relative humidity . . . . . . percent
57 Absolute pressure at exhaust flange of steam leaving prime-mover . . . . . . in. Hg at 32° F.
58 Absolute pressure at inlet flange of steam entering condenser . . . . . . in. Hg at 32° F.
59 Saturated-steam temperature corresponding to this pressure . . . . . . deg. F.
60 Measured temperature of same . . . . . . deg. F.
61 Absolute pressure at air-removal flange . . . . . . in. Hg at 32° F.
62 Absolute pressure at air-removal suction nozzle . . . . . . in. Hg at 32° F.

Surface Condenser

Total steam delivered to prime-mover or other apparatus exhausting to condenser, as measured . . . . . . lb.; (*a*) Rate per hr . . . . . . lb.
Total steam bled from 1st bleeder connection . . . . . . lb.; (*a*) Rate per hr . . . . . . lb.
Total steam bled from 2nd bleeder connection . . . . . . lb.; (*a*) Rate per hr . . . . . . lb.
Total steam bled from *n*th bleeder connection . . . . . . lb.; (*a*) Rate per hr . . . . . . lb.
Total water delivered by hotwell pump . . . . . . lb.; (*a*) Rate per hr . . . . . . lb.
Water leakage including steam from seals per hr . . . . . . lb.
(*a*) Immediately prior to test . . . . . . lb.; (*b*) Immediately after test . . . . . . lb.
(*c*) Average leakage—corrected to rate per hr . . . . . . lb.
Prime-mover steam condensed by condenser . . . . . . lb.; (*a*) Rate per hr . . . . . . lb.
Temperature of circulating water:
(*a*) Entering condenser . . . . . . deg. F.
(*b*) Leaving 1st pass . . . . . . ; 2nd pass . . . . . . ; 3rd pass . . . . . . deg. F.
(*c*) Leaving condenser . . . . . . deg. F.
Temperature of condensate in hotwell . . . . . . deg. F.
Temperature of condensate leaving internal heater—if any . . . . . . deg. F.
Quantity, temperature, etc., of make-up water introduced to condenser or hotwell . . . . . .

Circulating Pump

Circulating water (measured if possible) . . . . . . gal. per min.
Circulating-water pressure above or below atmospheric:
(*a*) At condenser inlet nozzle . . . . . . ; (*b*) At outlet nozzle . . . . . . lb. per sq. in.
(*c*) Difference in level between circulating nozzles center lines . . . . . . ft.
(*d*) Net condensing-circulating friction . . . . . . lb. per sq. in.
Piston speed (or r.p.m. if centrifugal) . . . . . . ft. per min.
Pressure: (*a*) at pump intake nozzle . . . . . . ; (*b*) at discharge . . . . . . lb. per sq. in.
(*c*) Difference in level of gages . . . . . . ft.
(*d*) Total corrected pumping head . . . . . . lb. per sq. in.
Absolute pressure: (*a*) at throttle . . . . . . ; (*b*) at exhaust nozzle . . . . . . lb. per sq. in.
(*c*) Quality of steam at throttle, percent moisture or deg. superheat . . . . . .
Total I.Hp. or B.Hp. (or electrical Hp. if motor driven) . . . . . .
Weight of steam consumed per hour . . . . . . lb.

Jet, Barometric or Ejector Condensers

Amount of water handled by injection or removal pump . . . . . . gal. per min.
Temperature of injection water, deg. F.:
(*a*) entering condenser . . . . . . ; (*b*) leaving condenser . . . . . .
83 Absolute pressure of injection water in distributing box at inlet flange . . . . . . lb. per sq. in.
(*a*) Height of same above center line of pump . . . . . . ft.
84 Pump pressure (by gage): (*a*) discharge . . . . . . ; (*b*) intake . . . . . . lb. per sq. in.
(*c*) Difference in level between center of gages . . . . . . ft.
(*d*) Water level in condenser or hotwell above center line of pump . . . . . . ft.
85 Correct total pumping head . . . . . . lb. per sq. in.
86 Speed of pump . . . . . . r.p.m.

APPARATUS TESTS (*Continued*)

Item

AIR-VAPOR REMOVAL APPARATUS. ALL TYPES OF RECIPROCATING DRY VACUUM PUMPS, STEAM OR MOTOR DRIVEN. ALL TYPES OF HYDRAULIC PUMPS, STEAM OR MOTOR DRIVEN

87 Steam conditions:
Absolute pressure at:
(*a*) Throttle............; (*b*) steam exhaust nozzle.............. lb. per sq. in.
(*c*) Quality of steam at throttle, percent moisture or deg. superheat ....
(*d*) Weight of steam consumed.................................... lb. per hr.
88 Air-vapor pressures and temperatures:
(*a*) Absolute pressure at pump suction nozzle.................... in. Hg at 32° F.
(*b*) Temperature of air-vapor at suction nozzle.......................... deg. F.
(*c*) Lowest absolute pressure with closed suction. (Determine before or after main test, under test conditions.)...................... in. Hg at 32° F.
89 Horsepowers of drivers:
(*a*) Reciprocating steam cylinders.......I.Hp.; (*b*) Turbine................ B.Hp.
(*c*) Power input if motor driven.. kw.
(1) Speed of motor......r.p.m. (2) Efficiency of motor............. percent
90 Reciprocating dry vacuum pumps:
(*a*) Horsepower of vacuum cylinders.................................. I.Hp.
(*b*) Speed...................................r.p.m. or strokes per min.
(*c*) Displacement of piston................................. cu. ft. per min.
(*d*) Mechanical efficiency of pump.................................. percent
Temperature of jacket cooling water:
(*e*) entering...................; (*f*) leaving.......................... deg. F.
(*g*) Amount of jacket cooling water............................... lb. per hr.
(*h*) Temperature of air-vapor at pump discharge nozzle.................... deg. F.
Hydraulic type vacuum pump:
(*a*) Speed of pump... r.p.m.
Temperature of hurling water: (*b*) entering.......; (*c*) leaving............. deg. F.
(*d*) Amount of hurling water circulated......................... gal. per min.
Absolute pressure of hurling water:
(*e*) at pump suction nozzle........; (*f*) at discharge nozzle.......... lb. per sq. in.
Hurling water make-up:
(*g*) amount...........gal. per min.; (*h*) temperature.................... deg. F.
92 Steam air ejector: (*a*) Absolute steam pressure at ejector throttle........ lb. per sq. in.
(*b*) Quality of steam at ejector throttle......deg. F. superheat or percent moisture
Steam used in nozzles: (*c*) 1st stage.......; (*d*) 2nd stage.......... lb. per hr.
Absolute air-vapor pressure at suction nozzles:
(*e*) main..................; (*f*) 2nd stage................ in. Hg at 32° F.
Temperature of air-vapor at suction nozzles:
(*g*) main................; (*h*) 2nd stage....................... deg. F.
Temperature of intercooler water: (*i*) ....; (*j*) leaving.............. deg. F.
(*k*) Amount of water used in intercooler......................... gal. per min.
(*l*) Amount of condensate leaving intercooler (if surface).................. deg. F.
(*m*) Temperature of condensate leaving intercooler....................... deg. F.
(*n*) Absolute pressure at exhaust nozzle.......................... lb. per sq. in.
(*o*) Temperature at exhaust nozzle................................... deg. F.
93 Actual amount of air discharged (measured by gasometer or orifice) .... cu. ft. per min. at 60° F. and 14.7 lb.

Condensate Pump

Speed of pump... r.p.m.
Heat contained in condensate above 32° F.................................. B.t.u.
Total amount of water handled per hour.....lb.; (*a*) Rate per min............. gal.
Water pressures (corrected to center line of pump):
(*a*) Pump discharge..............; (*b*) Pump suction........................ ft.
(*c*) Total head on pump.. ft.
(*d*) Water level in condenser or hotwell above center line of pump................ ft.
98 Water horsepower from above.. Hp.
99 Power required to drive, if motor-driven.................................. B.Hp.
100 Absolute pressure: (*a*) at throttle.........; (*b*) at exhaust............ lb. per sq. in.
101 Quality of steam.......................percent moisture or deg. F. superheat
102 Steam required to drive.. lb. per hr.

GENERAL RESULTS OF TESTS

103 Steam condensed per hour:
(*a*) Guarantee..............; (*b*) Test (Item 69*a*)........................ lb.
104 Absolute pressure at condenser exhaust nozzle:
(*a*) Guarantee.............; (*b*) Test (Item 58)............. in. Hg at 32° F

(*Continued on following page*)

CONDENSING APPARATUS TESTS *(Continued)*

Item		
	GENERAL RESULTS OF TESTS—*Continued*	
105	Temperature corresponding to test pressure (Item 59)...........	deg. F.
106	Temperature of condensate in hotwell:	
	(*a*) Guarantee...............; (*b*) Test (Item 71).....................	deg. F.
107	Temperature of entering condensing water:	
	(*a*) Guarantee...............; (*b*) Test (Item 70*a*).....................	deg. F.
108	Quantity of condensing water: (*a*) Guarantee........; (*b*) Test........	gal. per min.
109	Amount of free air handled by air pump:	
	(*a*) Guarantee......; (*b*) Test (Item 93).....cu. ft. per min. at 60° F. and 14.7 lb.	
110	Total heat delivery to prime mover: (From Items 48 to 52)..............	B.t.u. per hr.
111	Total heat:	
	(*a*) Carried to condenser by exhaust steam (Formulas 1 to 4).........	B.t.u. per hr.
	(*b*) Carried from condenser by condensate (Items 67 to 71)...........	B.t.u. per hr.
	(*c*) Absorbed by condensing water (Items 70 to 74)..................	B.t.u. per hr.
112	Steam supplied to auxiliaries:	
	(*a*) Absolute pressure at throttle................................	lb. per sq. in.
	(*b*) Quality of steam....................percent moisture or deg. F.	superheat
113	Steam consumed by auxiliaries: (*a*) Guarantee........; (*b*) Test..........	lb. per hr.
	(*c*) Percent of steam delivered to condenser..............................	percent
114	I.Hp. or B.Hp. of auxiliaries: (*a*) Guarantee........; (*b*) Test..................	Hp.
115	Mechanical efficiency of circulating or tail pump (Items 74 to 85)..............	percent
116	Mechanical efficiency of condensate pump (Items 98 to 102)..................	percent
117	Mechanical efficiency of air removal pump..................................	percent
118	Volumetric efficiency of air removal pump..................................	percent
119	Average velocity of circulating water through tubes......................	ft. per sec.
120	Heat transfer per deg. difference of temperature per sq. ft. per hr. (based on total surface):	
	(*a*) Using arithmetical mean.....; (*b*) Using logarithmic mean...............	B.t.u.
121	Steam condensed per sq. ft. of total surface per hr. (Item 52 ÷ Item 14)..............	lb.
122	Circulating water per lb. of steam condensed (Item 74 ÷ Item 69*a*)..................	lb.
123	Circulating water per sq. ft. of total condenser surface (Item 74 ÷ Item 14)..	lb.

In using the above formulas to determine heat absorbed by the condenser, the heat carried away by the condensate should be subtracted from the heat absorbed by the condenser.

$$h_c = h_x - (t_c - 32) \quad . \quad . \quad . \quad . \quad . \quad . \quad . \quad . \quad . \quad . \quad [5]$$

where h_c = B.t.u. per lb. of exhaust steam absorbed by condenser and carried away by circulating water; t_c = temperature of condensate.

If prime-mover steam rate is given in lb. per Hp.-hr., the constant 2545 must be substituted for constant 3413. Value of e varies with type and size of prime mover and should be obtained from the turbine manufacturer.

HEAT TRANSFER IN SURFACE CONDENSER is the measure of performance and is expressed in B.t.u. transmitted per sq. ft. of tube surface per hr. per degree mean difference in temperature between the two sides of the tube.

Arithmetical mean difference $= t_s - \{(t_2 + t_1)/2\}$

Logarithmic mean difference $= (t_2 - t_1)/\log_e \{(t_s - t_1)/(t_s - t_2)\}$

where t_s = temperature in steam space, t_1, t_2 = respectively, temperature of incoming and outgoing circulating water. Both arithmetical and logarithmic mean temperature differences should be calculated in order to make comparison possible with all other tests.

Data and Results should be reported in accordance with the form on p. 16–33.

11. TEST CODE FOR RECIPROCATING STEAM-DRIVEN DISPLACEMENT PUMPS

Approved, 1925

This code applies to tests to determine performance of pump and engine, including reheaters, heaters and jackets, if any, and jacket pumps, circulating pumps, condensate pumps, and vacuum pumps which are concerned in their operation.

MEASUREMENTS.—*a*. Amount of water pumped, lb. *b*. Average total head, ft. *c*. Amount of steam supplied, lb., or amount of heat supplied, B.t.u.

Standard units for volume, head and power are the same as those used for centrifugal pumps. See p. 16–41.

INSTRUMENTS AND APPARATUS required for performance tests include:

a. Tanks and platform scales for weighing water. *b*. Graduated scales attached to water glasses of boilers if feedwater is measured. *c*. Pressure gages, vacuum gages, thermometers with suitable wells. *d*. Steam calorimeter. *e*. Barometer, *f*. Revolution counter or other speed measuring device. *g*. Indicators. *h*. Planimeter. *i*. Deadweight gage tester.

Gage on discharge main should be attached near pump discharge nozzle; that on suction main near suction nozzle. Gages should be at an angle of 90° to direction of flow, and should be provided with valves and pet cock air vents. When a suction or discharge surface condenser forms part of the unit under test, water-head readings should be made outside of and beyond condenser.

LEAKAGE TEST to determine tightness of suction and discharge valves and plunger packing should be made by applying specified discharge pressure and observing leakage through a handhole or manhole on suction side. If leakage is sufficient to affect capacity, conditions should be corrected before proceeding with test.

QUANTITY OF WATER PUMPED should be determined by computing actual plunger displacement in gallons or cubic feet. If possible, total pumpage should be checked by Venturi tubes, pitometer, weir, calibrated orifice, etc. In direct-acting pumps, actual stroke of each plunger must be determined by graduated stroke scales.

DURATION OF TEST should be not less than 8 hr. when surface condenser measurement is used, and 10 hr. if steam consumption is determined by measuring feedwater to the boiler.

RECORDS.—Instruments should be read at least every 15 min., and indicator cards taken from each end of each cylinder once each hour with uniform conditions, and oftener if conditions vary.

Throttle Pressure in steam pipe is that shown by a calibrated gage on the steam pipe, $1^1/_2$–2 diameters from the throttle.

CALCULATION OF RESULTS.—Steam Rate is reported as the actual steam rate as measured. *Estimated steam* also is reported as corrected for pressure, quality or vacuum in accordance with correction factors based on data obtained from engine under test.

Heat Supplied is the total heat content of steam entering the engine and its auxiliaries, less total heat returnable from engine and auxiliaries to the boiler as follows: *a.* In condensate from air pump. *b.* In any feedwater heater using exhaust steam or steam from working charge of the engine. *c.* In high-temperature jacket and other drain water.

Total Head is the sum of the discharge head and suction head when suction main is under vacuum, and the difference between discharge head and suction head when suction main is under pressure.

Discharge Head is the sum of the head in ft. shown by the gage on the discharge main and the vertical distance between the center of the gage (Bourdon spring type) or lower surface of the mercury (mercury type) above center line of pump cylinder. If gage be located below the center line, the distance is to be subtracted instead of added.

Suction head is the difference between pressure, in feet, shown by suction gage and vertical distance, in feet, between center of gage on suction main (Bourdon type), or lower surface of the mercury (mercury type), and the datum line.

Suction lift is the sum of the reading, in feet, shown by gage on suction main and the vertical distance, in feet, between datum line and point of connection of gage to suction pipe. Should point of connection be above datum line, distance should be subtracted instead of added.

CAPACITY by displacement is found by

$$Q = A \times L \times n \times 1440 = \text{cu. ft. per 24 hr.}; \quad Q_1 = 7.48\,Q = \text{U. S. gal. per 24 hr.}$$

where A = net or effective area of all plungers, sq. ft.; L = length of stroke, ft.; n = number of discharge strokes per min.

WATER HORSEPOWER is found by

$$\text{Hp.}_w = (w \times H)/1{,}980{,}000$$

where w = weight of water pumped, lb. per hr., H = average total dynamic head, ft.

ENGINE INDICATED HORSEPOWER for entire test is found by dividing average water horsepower by average hydraulic and mechanical efficiency of the entire unit.

DUTY FOR 1000 LB. OF STEAM in foot-pounds of work done is

$$W_s = \{(W \times H)/w_s\} \times 1000$$

where W_s = duty per 1000 lb. of steam, ft.-lb.; W = weight of water pumped during test, lb.; H = average total dynamic head, ft.; w_s = steam supplied during test, lb.

DUTY PER 1,000,000 B.T.U. in foot-pounds of work done is

$$W_h = (W \times H)/H_s \times 1{,}000{,}000$$

where W_h = duty per million B.t.u.; H_s = total number of B.t.u. supplied. Other notation as before.

FRICTION AND MECHANICAL EFFICIENCIES.—Average combined hydraulic and mechanical efficiency of the entire unit is 100 times the average of the ratios of instantaneous water horsepower to engine indicated horsepower shown by all normal indicator cards taken during test.

DATA AND RESULTS should be reported in accordance with the form on p. 16–38.

DATA AND RESULTS OF RECIPROCATING STEAM-DRIVEN DISPLACEMENT PUMP TEST

Item

DESCRIPTION AND GENERAL DIMENSIONS

8 Type of unit..
9 Number of steam cylinders..
10 Diameter of steam cylinders: h.p.........in.; i.p.........in.;.........l.p......... in.
11 Diameter of steam piston rods.. in.
12 Stroke of steam pistons.. in.
13 Plungers, number..
14 Single or double-acting..
15 Inside or outside packed..
16 Diameter of plungers.. in.
17 Stroke of plungers.. in.
18 Diameter of plunger rods (if any).. in.
19 Net area of each plunger.. sq. ft.
20 Type of condenser..
21 Location of condenser..
22 Cooling surface in condenser.. sq. ft.
23 Type and size of condenser pumps..
24 Type and size of any exhaust or receiver, feedwater heaters, steam reheaters and jacket pumps or other auxiliaries forming part of the unit..

TEST DATA AND RESULTS

Pressures

25 Steam pressure at throttle.. lb. per sq. in.
26 First receiver pressure.. lb. per sq. in.
27 Second receiver pressure.. lb. per sq. in.
28 H.p. jacket.. lb. per sq. in.
29 I.p. jacket.. lb. per sq. in.
30 L.p. jacket.. lb. per sq. in.
31 Barometer.........................in. Hg........................ lb. per sq. in.
32 Corresponding absolute pressure.. lb. per sq. in.
33 Vacuum in exhaust pipe near l.p. cylinder..............in. Hg.......... lb. per sq. in.
34 Corresponding absolute pressure near l.p. cylinder........................ lb. per sq. in.
35 Vacuum in condenser...................in. Hg........................ lb. per sq. in.
36 Corresponding absolute pressure in condenser........................ lb. per sq. in.

Temperatures

37 Steam at throttle valve.. deg. F.
38 Exhaust steam.. deg. F.
39 Water pumped.. deg. F.
40 Condensate leaving surface condenser.. deg. F.
41 Condensate or feedwater entering feedwater heaters........................ deg. F.
42 Condensate or feedwater leaving feedwater heaters........................ deg. F.
43 Temperature rise in feedwater heaters.. deg. F.
44 Condensate from jackets.. deg. F.
45 Condensate leaving 1st receiver.. deg. F.
46 Condensate leaving 2nd receiver.. deg. F.
47 Condensate from other drains.. deg. F.
48 Engine-room air.. deg. F.
49 External air.. deg. F.

Total Head

50 Discharge by gage.. lb. per sq. in.
51 Equivalent discharge head.. ft. of water
52 Vacuum or pressure shown by gage on suction main........................ ft. of water
53 Discharge head referred to datum line.. ft. of water
54 Suction head or suction vacuum referred to datum line........................ ft. of water
55 Total dynamic head pumped against.. ft. of water
56 Actual suction lift (if any) measured to highest point of discharge valve deck.. ft. of water

Quality of Steam Near Throttle

57 Dryness factor of steam.. percent
58 Superheat in steam (if any).. deg. F.

Total Steam Quantities

59 Total steam, as measured, supplied to engine and the auxiliaries concerned in its operation as specified.. lb.
60 Correction factor conforming to conditions agreed upon........................ percent
61 Total *estimated steam* supplied conforming to conditions agreed upon.............. lb.

Total Pump Quantities

62 Total quantity of water pumped by plunger displacement.......... U. S. gal. or cu. ft.
63 Equivalent total weight of water pumped.. lb.
64 Total quantity of water pumped as shown by other means of measurement.. U. S. gal. or cu. ft.

TESTS OF RECIPROCATING STEAM-DRIVEN PUMPS (*Continued*)

TEST DATA AND RESULTS—*Continued*

Item		
	Hourly Steam Quantities	
65	Steam, as measured, supplied per hour	lb.
66	*Estimated steam* supplied per hour	lb.
	Hourly Pump Quantities	
67	Quantity of water pumped per hour, by plunger displacement	U. S. gal. or cu. ft.
68	Equivalent weight of water pumped per hour	lb.
69	Quantity of water pumped per hour, as shown by other means of measurement	U. S. gal. or cu. ft.
70	Slip, in percentage of plunger displacement	percent
	Heat Supplied (Based on steam supplied, as measured, Item 59)	
71	Total heat per pound of steam at throttle	B.t.u.
72	Total heat in steam supplied	B.t.u.
73	Total heat returnable from engine to boiler	B.t.u.
74	Net heat supplied	B.t.u.
75	Net total heat supplied per hour	B.t.u.
	Heat Supplied (Based on total *estimated steam,* Item 61)	
76	Total heat per pound of steam at throttle	B.t.u.
77	Total heat in steam supplied	B.t.u.
78	Total heat returnable from engine to boiler	B.t.u.
79	Net heat supplied	B.t.u.
80	Net total heat supplied per hour	B.t.u.
	Speed	
81	Total number of revolutions	
82	Total number of single strokes	
83	Actual length of stroke, if direct-acting	ft.
84	Revolutions per minute	r.p.m.
85	Single strokes per minute.	
86	Piston and plunger speed per minute	ft. per min.
	Power	
87	Total work done during test	ft.-lb.
88	Water horsepower	W.Hp.
89	Combined hydraulic and mechanical efficiency	percent
90	Indicated steam horsepower	I.Hp.
91	Mean effective pressure (referred to l.p. cylinder if multi-expansion engine)	lb. per sq. in.
92	Indicated water horsepower	I.W.Hp.
	Economy Results (Based on steam supplied, as measured, Item 65)	
93	Steam supplied per I.Hp.-hr	lb.
94	Heat supplied per I.Hp.-hr	B.t.u.
95	Heat supplied per W.Hp.-hr	B.t.u.
	Economy Results (Based on *estimated steam,** Item 66)	
96	Steam supplied per I.Hp.-hr	lb.
97	Heat supplied per I.Hp.-hr	B.t.u.
98	Heat supplied per water Hp.-hr	B.t.u.
	Efficiency Results	
99	Thermal efficiency (referred to I.Hp.) based on steam as measured (2545 ÷ Item 94)	percent
100	Thermal efficiency (referred to I.Hp.) based on *estimated steam* * (2545 ÷ Item 97)	percent
	Duty	
101	Duty per 1000 lb. of steam as measured	ft.-lb.
102*	Duty per 1000 lb. of *estimated steam*	ft.-lb.
103	Duty per 1,000,000 B.t.u. based on steam supplied, as measured	ft.-lb.
104	Duty per 1,000,000 B.t.u. based on *estimated steam*	ft.-lb

* The correction factor used shall be stated in percentage, and the basis and method of determining it described.

12. TEST CODE FOR CENTRIFUGAL AND ROTARY PUMPS

Approved, 1927

This code applies to tests for determining performance of pump only. It does not apply to performance of piping, driving apparatus or other auxiliaries, except as they may affect results of pump test.

INSTRUMENTS AND APPARATUS required for measuring the head of water and speed of pump are:

a. Water column or gage glass. *b.* Pressure gage. *c.* Vacuum gage. *d.* Mercury column. *e.* Revolution counter. *f.* Tachometer.

Instruments and apparatus for measuring the quantity of water and amount of power input depend on methods to be used.

OPERATING CONDITIONS are: *a.* Quantity of water delivered by pump. *b.* Total dynamic head. *c.* Speed in r.p.m.

Speed always should be maintained as nearly constant and as nearly equal to nominal designed speed as possible. Prevailing total dynamic head under operating conditions may not be the designed head. If possible it should be brought to designed value by throttling by means of a valve in the discharge line, or other methods applicable to the local conditions. Throttling by a valve in the suction line close to the pump is not permissible. If operating head is greater than design head, additional delivery openings in discharge mains at a lower elevation will lower the operating head. Side outlets between pump and point of measurement of discharge should be blanked off.

Each of the various measurements for a given run shall be computed by averaging a series of instantaneous readings taken simultaneously at equal time intervals. Extreme fluctuations at any instant should be noted. The number of readings shall be such that the elimination or addition of a single reading, representing a maximum swing of the instrument, shall not affect the average by more than 1%.

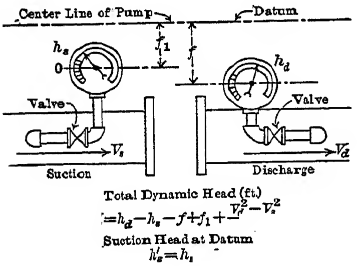


FIG. 1

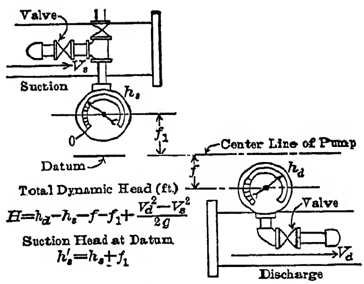
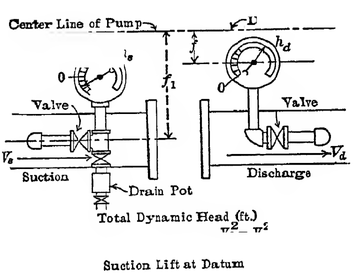


FIG. 2

Bourdon Gages. Absolute Suction Head above Atmospheric Pressure



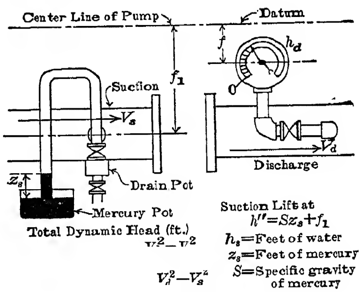


FIG. 3 FIG. 3A. Alternative to Fig. 3

Bourdon Gages. Absolute Suction Head below Atmospheric Pressure

HYDRAULIC AND MECHANICAL CONDITIONS.—The unit shall be in the best possible mechanical condition with bearings, stuffing boxes and internal running clearances properly adjusted. The water to be handled shall be clear cold water, free of air, gases or suspended solids. Unless otherwise stipulated, temperature of water during test shall not exceed 85° F.

For pumps handling fluids other than water, suitable corrections shall be made, depending on characteristics of the fluid; specific gravity and viscosity, referred to water, and boiling point, shall be determined. The dry weight and volume of solids in suspension, and the total weight and volume of the mixture shall be determined. Formulas in the code are based on the use of water whose specific gravity is unity.

DURATION OF TEST depends on method of driving pump, on method used to measure discharge, and also on how completely pump characteristic is determined. In determining characteristic, capacity should be varied in steps covering a range of at least 10 to 15% above and below normal tested delivery of pump, and curves plotted of the results.

STANDARD UNITS FOR VOLUME HEAD AND POWER.—The Standard Unit for Volume is the U. S. gallon or the cubic foot. The gallon unit is expressed in gallons per min. (g.p.m.) or million gallons per day of 24 hours (m.g.d.).

The cubic foot unit is expressed in cu. ft. per sec. One U. S. gallon = 231 cu. in. For temperatures not exceeding 85° F, specific gravity of water may be taken as unity. (For more exact computations, the specific gravity may be calculated by means of the weights of water at various temperatures as given in Table 2, p. 2–04.)

In computations, the following relations shall be used. 1 cu. ft. = 7.4805 gal.; 1 cu. ft. per sec. = 448.83 gal. per min.; 694.45 gal. per min. = 1,000,000 gal. per day of 24 hr.

The Unit for Measuring Head shall be the foot. 1 lb. per sq. in. = 144 ÷ weight of 1 cu. ft. of liquid. For water whose specific gravity is unity, 1 lb. per sq. in. = 2.307 ft.

When the temperature of water is such that the weight per cubic foot varies from 62.43 lb., or when liquid other than water is handled, the reading of gages calibrated in lb. per sq. in. must be corrected by dividing the reading by the specific gravity and multiplying by 2.307.

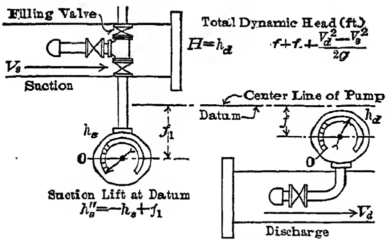


FIG. 4. Bourdon Gages. Absolute Suction Head below Atmospheric Pressure

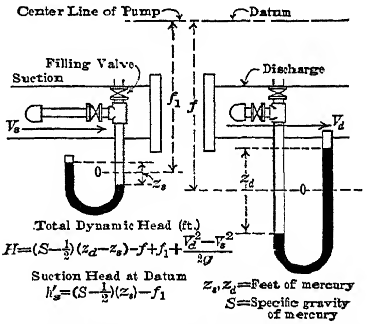
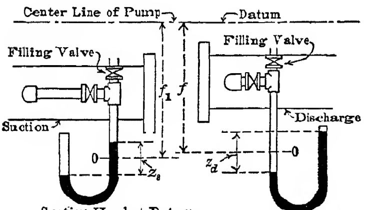


FIG. 5. Mercury U-tubes. Absolute Suction Head above Atmospheric Pressure



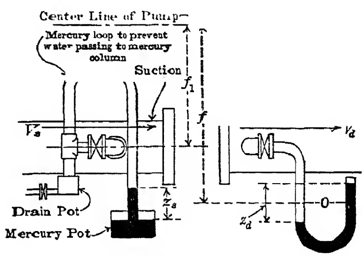
Suction Head at Datum

$H = (S - \frac{1}{2})(z_d + z_s) - f + f_1 + \frac{V_d^2 - V_s^2}{2g}$

Suction Lift at Datum, $h''_s = (S - \frac{1}{2})(z_s) + f_1$

z_s, z_d = Feet of mercury

S = Specific gravity of mercury



Suction Head at Datum

$H = (S - \frac{1}{2})(z_d) + Sz_s - f + f_1 + \frac{V_d^2 - V_s^2}{2g}$

Suction Lift at Datum, $h''_s = Sz_s + f_1$

FIG. 6 FIG. 6A. Alternative to Fig. 6

Mercury U-tubes. Absolute Suction Head below Atmospheric Pressure

The Unit of Power shall be the horsepower (550 ft.-lb. per sec.) delivered to the pump ed as shaft-horsepower. (S.Hp.)

NTS OF QUANTITY OF WATER are primary or secondary. The .easurement by volume or weight. Secondary methods, more nt by weir, Venturi meter, nozzle, or Pitot tube. istalled as to change either the coefficient of th ump.

·Net total head is the total dynamic head produced by the pump. It is the between the absolute head at discharge and suction nozzles, taking into acco nce in elevation of these nozzles with respect to a fixed datum, and any differe velocity heads at points of measurement. The standard method of measuring use of a water column or gage glass giving direct reading of surface elevation. If ble, indirect methods may be used, as a mercury or other fluid gage very gage or pressure measuring device must be attached by a smooth- 1/4 in. diam., whose axis is at right angles to the direction of flow, with the inside of the conduit. For great accuracy when the h operates is 50 ft. or less, ometers instead of Mercury gages shall under actual co aili comparison with the height of the water column.

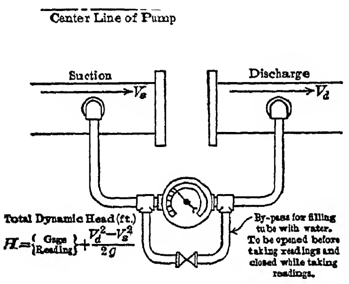


FIG. 7. Differential Gage, Bourdon Type

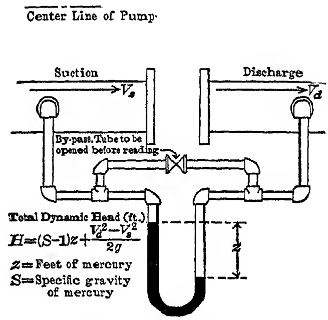
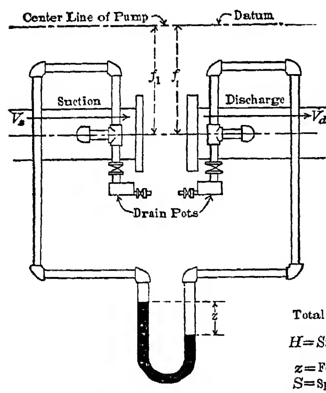
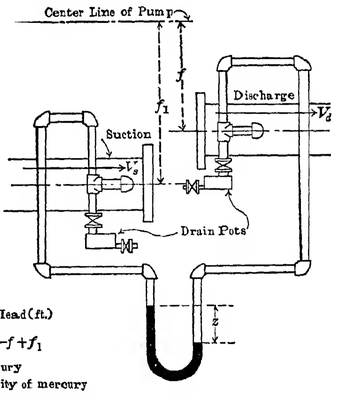


FIG. 8. Differential Gage, Mercury U-tube. Suction Head above or below Atmospheric Pressure. Discharge Head above Atmospheric Pressure





Total Dynamic Head (ft.)

$$H=Sz+\frac{V_d^2-V_s^2}{2g}-f+f_1$$

z = Feet of mercury

S = Specific gravity of mercury

Suction and Discharge Heads below Atmospheric Pressure

FIG. 10. Suction and Discharge at Different Levels

Differential Gage Mercury U-tube

Total Dynamic Head may be expressed by the formula

where H = total dynamic head; H_d = absolute discharge pressure head, ft.; H_s = absolute suction pressure head, ft.; V_d, V_s, = respectively mean discharge and suction velocity, ft. per sec., g = acceleration due to gravity, = 32.2.

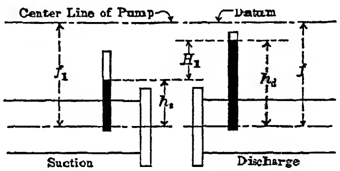
$$H_d = h_d' + h_a; \quad H_s = h_s' + h_a = -h_s'' + h_a$$

where h_a = atmospheric pressure head, ft., corresponding to barometer; h_d = discharge gage reading, ft., h_d' = discharge gage reading, ft., corrected to datum; h_s = suction gage reading, ft., positive when above, negative when below, atmospheric pressure; h_s'' = suction lift or negative suction pressure (below atmosphere), ft., corrected to datum $= - h_s'$.

The use of the various types of measuring instruments is shown in Figs. 1 to 13.

MEASUREMENT OF SPEED shall be taken by a revolution counter or an accurately calibrated tachometer.

MEASUREMENT OF POWER INPUT *i.e.*, shaft-horsepower of the pump, may be made by some form of transmission dynamometer or calibrated electric motor. Other methods involve measurement during the pump test of power input to the driving element and the previous or subsequent determination of the relation of power input to power output of this driving element under the identical conditions of the pump test.



Total Dynamic Head, ft., =

Suction Head at Datum = $h'_s = h_s - f_1$

FIG. 11. Water Gage. Suction and Discharge Pressures Above Atmospheric Pressure

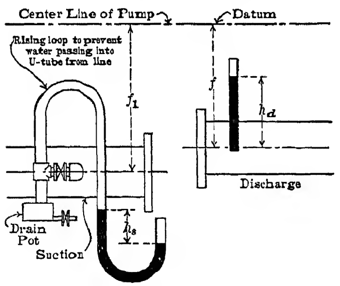
CALCULATION OF RESULTS.—Water Horsepower is found by the formula

$$\text{W. Hp.} = \{Q \times W \times (\text{Total head, ft.})\}/550$$
$$= \{\text{g.p.m.} \times (\text{total head, ft.})\}/3960$$

where W. Hp. = water horsepower; Q = cu. ft. per sec.; W = weight of water, lb. per cu. ft.; g.p.m. = gallons per minute. It is here assumed that the weight of water is 62.318 lb. per cu. ft.

Efficiency of a pump is ratio of energy converted into useful work to energy supplied to pump, or E = (W.Hp./S.Hp.)

RECORDS.—Test results usually are presented in the form of a single sheet upon which are drawn three curves showing respectively relation between efficiency and capacity,

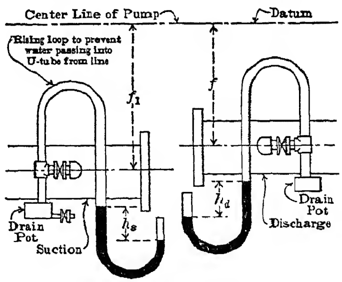


Total Dynamic Head, ft., =
$H = h_d + h_s + f_1 - f + \{(V_d^2$

Suction Lift at Datum =
$h''_s = h_s + f_1$

Water cannot be used if h_s is greater than height of rising loop.

FIG. 12. Water Gage. Absolute Suction Pressure below Atmospheric Pressure. Absolute Discharge Pressure above Atmospheric Pressure



Total Dynamic Head, ft., =
$H = h_s - h_d + f_1 - f +$ $V_s^2)/2g\}$

Suction Lift at Datum =
$h_s'' = h_s + f_1$

Water cannot be used if h_s is greater than rising loop. Read H, h_d, h_s, f and f_1 in feet.

FIG. 13. Absolute Suction and Discharge Pressures below Atmospheric Pressure

head and capacity, and shaft-horsepower and capacity. The values used in plotting these curves are those computed for a given definite speed of the pump, usually the speed of best efficiency, or that at which it was designed to operate.

If pump under test is of the rotary displacement type, its volumetric displacement corresponding to the speed developed in the test may be computed, and the difference between this displacement and the quantity actually pumped may be stated as slip, expressed as a percentage of displacement.

DATA AND RESULTS OF CENTRIFUGAL OR ROTARY PUMP TEST

Item	
	Pump Data
	ype. (Centrifugal or rotary, horizontal or vertical shaft, split or solid casing, enclosed or open-type impeller, single or multi-stage, spiral casing, with or without diffusion vanes, etc.)
8	Rated capacity
9	Rated head
10	Rated speed
11	Liquid pumped (clear, muddy, or sandy water, or other fluid, specific gravity, temperature, etc.)
12	Method employed for measuring quantity of water
13	Method employed for measuring power
14	Method employed for measuring head
15	Duration of period of measuring quantity of water
16	Duration of period of measuring power
17	Duration of period of measuring head
18	Size of discharge outlet
19	Size of suction inlet
20	Time of service and general condition of pump
	Driver Data
21	Type (motor, turbine, gear, belt, etc.)
22	Name of builder
23	Builder's type and serial number
24	Rated conditions (horsepower, speed, electrical or steam conditions, etc.)
25	Time in service
	Results
26	Volume of water pumped .... cu. ft. per sec .... g.p.m.
27	Discharge head absolute
	(a) Computed velocity in suction pipe at gage .... ft. per sec.
	(b) Computed velocity in discharge pipe at gage .... ft. per sec.
	(c) Head due to difference in velocity heads, plus or minus .... ft.
28	Suction head, absolute .... ft.
29	Total head on pump .... ft.
30	Speed .... r.p.m.
31	Water horsepower .... Hp.
32	Horsepower input to driver .... Hp.
33	Efficiency of driver .... percent
34	Shaft horsepower or horsepower input to pump .... Hp.
35	Pump efficiency .... percent
36	Specific speed of pump = r.p.m. $\times$ ($\sqrt{Q}/H^{3/4}$) at maximum efficiency point (from characteristic curve)

13. TEST CODE FOR INTERNAL COMBUSTION ENGINES

Approved, 1929

This code applies to all forms of internal combustion engines, but is limited to the engines alone. Separately-driven auxiliaries, essential to the operation of the engine, may be included in the test, but tests of such units must follow codes appropriate to them. Internal combustion engines vary widely in character and service, and operate under a wide range of conditions. No single code can prescribe tests in detail equally applicable to all conditions. Many details of the test must be left to the judgment of the engineer. The code includes:

a. Stationary engines and electric generating sets burning gaseous or liquid fuels including scavenging pumps of 2-cycle engines and injection air compressors, if used and separately driven. *b.* Marine engines, exclusive of propeller, and burning light or heavy liquid fuels. *c.* Aircraft engines, limited to block tests, exclusive of propeller, and burning light liquid fuels. *d.* Land vehicle engines, limited to block test and burning light and heavy liquid fuels. The code excludes: All sets where engine and driven unit are so combined that determination of the performance of the engine itself is impossible; all internal combustion engines whose construction or operating conditions prevent an output determination in terms of either brake horsepower or kilowatts;

internal combustion engines combined with vehicles, as vessels, air craft, motor cars, etc.; tests forming part of experimental investigations, research or development. The code is limited to tests in which the object is of a commercial nature.

MEASUREMENTS REQUIRED are:

a. Bore, stroke and clearance of power cylinders. *b.* Diameters of piston- and tail-rods of power cylinders of double-acting engines. *c.* Brake- or shaft-horsepower output. *d.* Kilowatt output if direct connected to a generator. *e.* Speed. *f.* Horsepower to drive independent cooling, water pump and fuel pump. *g.* Horsepower to drive independent scavenging pump or blower. *h.* Horsepower to drive independent injection air compressor. *i.* Cubic feet of gas supplied to gas burning engines, or pounds of liquid fuel supplied to liquid fuel engines. *j.* High calorific value of fuel.

Additional measurements that may be required are: 1. Cylinder diameters and strokes of injection air compressor, diameters of piston- and tail-rods. 2. Two-cycle engine scavenging pump, diameter and stroke and diameters of piston- and tail-rod. 3. Indicated horsepower of engine. 4. Explosions or fuel injections per minute. 5. Spray air pressure and temperature of air injection oil engines. 6. Exhaust back pressure at common exhaust nozzle. 7. Scavenging air pressure and temperature of 2-cycle engine. 8. Manifold vacuum of carburetor engine. 9. Jacket water or oil supply pressure and temperature at engine supply and main outlet; if jackets are divided, with separate supplies and discharges, the pressure at each. 10. Temperature, pressure, vacuum or hydraulic head, positive or negative, of supply of liquid fuel at carburetor connection of carburetor engine, at injection pump suction of injection oil engines, and pressure or vacuum of gas at stop valve of gas burning engine. 11. Total, static and velocity pressure, and temperature of cooling air supply of air-cooled engines. 12. Atmospheric pressure, temperature and humidity. 13. Pressure and temperature of lubricating oil supply to bearings at as many points as there are different supplies of pump-circulated lubricating oil, or in the crankcase of engines without pump, or having pump deliveries that do not permit of temperature measurement. 14. Amount of lubricant consumed. 15. Quantity of jacket water or oil supplied to engine as a whole or to separate jackets. 16. Quantity of cooling air supplied to air-cooled engines. 17. Quantity of jacket water evaporated in hopper-cooled engines. 18. Pounds of injection water supplied internally as liquid or steam, to cylinders of some oil engines. 19. Compression pressure in cylinders, when hot and cold, at normal speed at wide open air throttle.

INSTRUMENTS AND APPARATUS REQUIRED include:

a. Scales with or without tanks for weighing liquid fuel. *b.* Gas meters for measuring gaseous fuels, or gas metering methods with indirect observation apparatus. *c.* Gas calorimeters to determine heating value of gaseous fuel. *d.* Baumé oil hydrometer for petroleum liquid fuel and for indirect determination of calorific power. *e.* Pressure gages, mercury or water column. *f.* Thermometers. *g.* Barometers. *h.* Gas engine and oil engine indicators for working cylinders, and steam engine indicators for cylinders of 2-cycle scavenging pump, and low-pressure cylinders of injection air compressors. *i.* Pressure indicating or recording instruments for compression pressures or injection pressures in cylinders. *j.* Planimeters. *k.* Tachometers, revolution counters, or other apparatus for measuring speed or number of fuel admissions per minute. *l.* Absorption dynamometers of appropriate type, especially hydraulic brakes and electric dynamometers. *m.* If engine is direct connected to a generator, appropriate electrical instruments and apparatus, to provide suitable electrical load and measure it.

Dynamometers for testing all aircraft and land vehicle engines must be electric dynamometers. For brake-horsepower test of marine engines, hydraulic or electric dynamometers shall be used. For stationary engines and electric generating sets, electric or hydraulic dynamometers or other forms of brake may be used. For brake horsepower measurements, only a brake driven by direct coupling to, or mounted on, the engine shaft may be used.

With due precautions and under proper provisions, brake horsepower may be determined by substitution. The engine may be tested with a load such as a propeller, club or fan, which later may be tested for its speed-torque characteristic curve. The horsepower of the engine then is found by reading from this curve the horsepower and speed.

Tachometers of the centrifugal, liquid pumping, or magnetic type may be used after careful calibration. They must be connected by tachometer shaft or positive gearing to engine shaft. Continuous speed counters may be used if constant speed of considerable duration is maintained. Hand speed-counters placed directly against engine shaft may be used if handled carefully and well-made.

Calorimeters.—Unless otherwise agreed in advance, heating value of fuels to be used in calculations of results must be the *high value* products condensed, or the direct reading of a water calorimeter properly used. The following recommendations are made:

a. For gaseous fuel a standard form of gas calorimeter shall be used, and its determination used as heating value of gas as supplied. Heating value shall not be calculated from gas analysis. *b.* Heating value of liquid fuel shall be determined, in condition in which it is delivered to engine, in a standard bomb calorimeter, by a recognized physical or chemical laboratory. *c.* If previously agreed, heating values of gasoline may be estimated from the Baumé reading, by the Sherman and Kropf formulas, modified by Strong. These formulas (U. S. Bureau of Mines, Bull. No. 43) are:

Gasoline, $H = 18{,}320 + 40\,(B - 10)$; Kerosene, $H = 18{,}440 + 40\,(B - 10)$, where H = high heating value, B.t.u. per lb.; B = deg. Baumé.

Gas Measurements must be made with Venturi or drop-holder meter. Proportional meters, or anemometer-type meters are not satisfactory. Large capacity measurements are best made by some type of flow rate measurement. Physical or chemical properties of a mixed gas, determined for use in a metering calculation, shall be made by a recognized chemical laboratory previously agreed upon.

Liquid fuel must be weighed directly as used. Meters shall not be used.

PREPARATIONS.—Dimensions and physical conditions of all parts of engine shall be determined and recorded, and also of all parts external to engine, whose condition might affect test. Among external conditions so investigated may be included jacket water supply, external electric current supply for ignition system of electrically-ignited engines, sufficiency of fuel supply of liquid fuel engines, and sufficiency of lubricating oil supply.

DIMENSIONS.—In addition to those given in paragraph on measurement, the following data may be useful to identify an engine or to increase the value of future comparative analyses:

Dimensions of scavenging pump and injection air compressor, and of other fixed and running parts; diameter and length of all important bearings and pins, and of minor parts, such as valve lift, port area, area through mixing valves in gas-burning engines, carburetor choke area and nozzle sizes, injection pump plunger diameter and stroke, push rod or valve rocker clearances; all items of adjustment and timing, all nameplate information; dimensions of importance in judging the condition of engine as a machine, and useful in judging value of test, including all items of alignment, bearing clearances, pins, rods and slides, pistons and cylinders, rings in grooves, valve stems in guide, etc.

Internal Inspection should cover:

Jacket surface on the heat receiving side to insure freedom from water scale, rust or other deposits; combustion chamber walls including piston heads and all inward facing parts, which should be free from deposits; lubricating oil system, to insure freedom from deposits and cleanness of all flow passages; exhaust system, including expansion chambers, mufflers and pipes, to insure cleanness and arrangement that will not develop excessive back pressure; fuel supply and regulating system must be clean and free of all foreign matter, with passages fully open.

Leakage Tests should locate and correct all leakage. The more important leakages to be investigated are:

a. Cylinder leakage outward, best checked by an indicator. When there is clear space between compression and expansion lines of a diagram, taken when there is no combustion and when cooling water is not running, it may be assumed that leakage is excessive. In such case, and also if an indicator cannot be used to check leakage, the possible sources of leakage must be checked separately, *i.e.*, piston ring, cylinder, cylinder head gasket, air starting valve, relief valve, spray valve or spark plug seat, inlet or exhaust valve of 4-cycle engine, air scavenging valves of 2-cycle engines. *b.* Leakage from jackets through bad joints, cracks or porous spots, compressed air through leaky air starting valve, fuel oil past spray valve, and high pressure spray air. *c.* Miscellaneous leakages outside of cylinders, as in manifold system of carburetor engines between carburetor and inlet valve; fuel oil in injection oil engines at pump valves, pump plunger, oil delivery pipe, or spray valve; lubricating oil at pumps, tanks or in piping; spray air at compressor storage bottles, valve or piping; starting air at compressor storage reservoir, valves or piping; insulation of electrical circuits.

OPERATING CONDITIONS of the test should be agreed upon and defined. Such an agreement is regarded as essential to all internal combustion engine tests. The definition should include all of the items necessary in regard to service and to test procedure.

Variation.—Operating conditions to be maintained during test often are the ones that are most uncertain; thus, aircraft engines in service must work under a wide range of atmospheric temperature and pressure, and at any angle of inclination. Engines for land vehicles must operate on a fixed supply of jacket water or oil, with an air-cooled radiator, and external wall temperature varies with engine load, vehicle speeds, weather and climate. A statement of normal horsepower and speed for such engines is impossible. Marine engines always operate with a propeller load, under service conditions that cannot be reproduced in block tests. Even stationary engines in service operate under conditions that differ widely from those obtaining in a block test. Agreement on conditions of test is important, but details must be left to the judgment of the test engineer.

EFFECTIVE LOAD AND SPEED.—Operating conditions during test include mainly load and speed to be maintained. The agreement in all details should not be inconsistent with:

a. Stationary engines including generator sets shall be tested at constant speed under governor control, or as near constant as the governor will maintain it, and at whatever load may be required by agreement, or by specific object of the test. *b.* All land vehicle engines, except tractor engines fitted with speed governors and operating normally under governor control, shall be tested with throttle wide open, with brake torque regularly varied from zero to maximum and back to zero at whatever speeds may result. The horsepower-speed curve typical of the engine shall be determined

instead of horsepower at any given speed. Safe speed designated by engine builder must not be exceeded. If highest speed used is the safe limit, so state. Maximum torque applied in all cases shall be greater than that which will produce maximum horsepower, unless speed required exceeds safe speed. *c*. Marine and aircraft engines may be tested according to conditions prescribed for land vehicles, except that speed shall be maintained by brake torque and not by a governor. Marine engines may be tested at constant speed for full load operation fixed by their propellers.

Special attention should be given to questions of safety, such as safe speed or torsional vibration at a given speed within the normal range. Procedure in such cases should be by agreement.

STARTING, STOPPING AND DURATION.—Engine with all attachments shall be brought to a condition of steady operation, which shall continue long enough to permit preliminary observations to prove that steady state has been attained. If successive runs are to be made under different conditions, preliminary observations must be repeated for each run. An engine shall not be regarded as having attained its steady state until jacket and lubricating oil temperatures are substantially constant, nor until one hour after operating conditions have been imposed before the test observations are started.

For successive runs, under other conditions, minimum time for any engine shall be 10 min., actual time being determined by extent to which conditions are changed in each successive run. Maximum time to reach steady state shall not be greater than 24 hr.

Duration of test after establishment of steady operating conditions shall be by previous agreement. It should be greater for engines where reliability is important, and which require longest time to reach a steady state. Length of run shall be not less than period required to reach steady state, subject to the condition that length of run shall be great enough to insure accuracy of fuel measurement within 1%, except when metering of gaseous fuels is not improved by lengthened runs.

CALCULATION OF RESULTS.—**Fuel Consumption** for liquid fuel engines is stated in pounds of actual fuel measured and is subject to no correction.

For gaseous fuel engines, cubic feet of gas at prevailing pressure and temperature indicated or recorded by meter, or calculated from reading of flow rate devices, is stated in the report. This is to be corrected for pressure and temperature at which gas was delivered to calorimeter by the formula

$$Q_1 = Q \times (p_1/p_2) \times (t_2/t_1)$$

where Q_1 = corrected quantity of gas, cu. ft.; Q = quantity of gas as measured; p_1, p_2 = absolute pressure of gas at meter and calorimeter, respectively; t_1, t_2 = absolute temperature of gas at meter and calorimeter, respectively.

Heat Consumption is the product of heating value of fuel by fuel consumed per hour. Heating value for gaseous fuels is that found by the gas calorimeter, B.t.u. per cu. ft., high value. Heating value of liquid fuels is that given in report of physical or chemical laboratory, B.t.u. per lb., high value.

Indicated Horsepower if specified in agreement, shall be the gross I.Hp. as determined from indicator diagrams from working cylinders alone.

For 4-cycle engines, I.Hp. = $PLAN/(2 \times 33{,}000)$;

For 2-cycle engines, I.Hp. = $PLAN/33{,}000$,

where P = mean indicated pressure, lb. per sq. in.; L = length of stroke, ft.; A = area of piston, minus area of piston- or tail-rod, sq. in.; N = rev. per min. Total indicated horsepower of a double-acting cylinder is sum of the horsepowers developed in the two ends. Total indicated horsepower of a multi-cylinder engine is sum of the horsepowers developed in all cylinders.

Brake Horsepower is found by the formula

$$\text{B.Hp.} = 2\pi LWN/33{,}000$$

where W = net force on brake arm, lb.; L = distance between center of shaft and bearing point or weight center at end of brake arm, ft.; N = rev. per min. of shaft. If brake horsepower is calculated from electric measurements, 2% of full load capacity of generator at applicable speed shall be added to measured electric output to cover windage, friction and undetermined losses. Correction shall be made for conventional efficiency of generator.

Brake Mean Effective Pressure must be calculated from brake horsepower by the formulas

For 4-cycle engines, B.M.E.P. = $(\text{B.Hp.} \times 33{,}000 \times 2)/(LAN)$;

For 2-cycle engines, B.M.E.P. = $(\text{B.Hp.} \times 33{,}000)/(LAN)$.

Symbols have the same meaning as in formula for indicated horsepower.

Electrical Horsepower is kilowatt output at terminals divided by 0.746. With alternating-current generators, output shall be net output. When power for excitation or ventilation is taken directly from engine shaft, output indicated at terminal is net output.

(*Continued on page 16–49*)

DATA AND RESULTS OF INTERNAL-COMBUSTION ENGINE TEST AT CONSTANT SPEED

DESCRIPTIONS, DIMENSIONS, ETC.

Type of engine, 2- or 4-cycle, single- or double-acting, horizontal or vertical; if 4-cycle, the valve arrangement (L, T, or I head); if 2-cycle, the type of scavenging; if single-acting, whether crosshead or trunk piston; if multi-cylinder the arrangement of cylinders and cranks; gas or liquid fuel; if liquid fuel, class name, fixing manner of treating oil, as carburetor-type, air-injection, mechanical injection, or primary combustion..

Class and kind of service, stationary and special feature, marine, aircraft or vehicle....

Auxiliaries attached, as magneto, fuel-injection, fuel-circulating, lubricating-oil, jacket circulating, and scavenging pumps, spray-air and maneuvering or starting air compressors, radiator fans, oil or fuel coolers or heaters..............................

Auxiliaries, independent or separately driven, and power...........................

Type, capacity, maker and other features of auxiliaries............................

Rated brake horsepower of engine, or kw. of electric generating set, and speed.........

Grade of fuel for which designed, kind of gas or specification of liquid fuel, and fuel used in test...

Special structural features for fuel utilization.....................................

Special structural features of speed and power control, and governor or reversing gear....

Number of working cylinders...

Bore and stroke of power cylinders.

piston- and tail-rods

. constant for power

Crank-end hp. constant for power cylinders (stroke × net pis

Capacity of generator or apparatus consuming power of engine.

Characteristics of generator: d.c. or a.c.; volts; cycles; phase.

TEST DATA AND RESULTS

			Test No. 1	Test No. 2
23	Duration of test..............................	hr.		
	Pressures, Average			
24	Barometric pressure......in. Hg........	lb. per sq. in.		
25	Spray-air pressure (air-injection Diesel engines) average gage...............	lb. per sq. in.		
26	Exhaust back pressure at exhaust nozzle...	in. of water		
27	Jacket water supply pressure at crank-shaft center........................	lb. per sq. in.		
28	Manifold vacuum (carburetor engines)....	in. of water		
29	Gas pressure at meter (gaseous fuel) ....in. water..............	lb. per sq. in.		
30	Lubricating-oil pressure, circulating force-feed system at crank-shaft center......	lb. per sq. in.		
31	Scavenging-air pressure, average gage, in engine header (2-cycle engines)........	lb. per sq. in.		
	Temperatures			
32	Engine-room temperature...................	deg. F.		
33	Temperature of fuel (gaseous) at meter........	deg. F.		
34	Temperature of main air supply entering engine.	deg. F.		
35	Temperature of mixture at intake port (carburetor engine)........................	deg. F.		
36	Temperature of main jacket water or oil inlet....	deg. F.		
37	Temperature of main jacket water or oil outlet...	deg. F.		
38	Temperature of piston-cooling water or oil at inlet..................................	deg. F.		
39	Temperature of piston-cooling water or oil at outlet.	deg. F.		
	Fuel Properties			
40	Heating value of gas (high), at standard pressure and temperature..........	B.t.u. per cu. ft.		
41	Heating value (high) of liquid fuel.........	B.t.u. per lb.		
	Total Quantities			
42	Total gaseous fuel at meter pressure and temperature.......................	ft.		
43	Correction factor for gas [(absolute meter pressure × absolute standard temperature) ÷ (absolute standard pressure × absolute meter temperature)]..................................			
44	Total gaseous fuel at standard pressure and temperature (Item 42 × Item 43)..............	ft.		
45	Total liquid fuel......................................	lb.		

TESTS OF INTERNAL COMBUSTION ENGINE AT CONSTANT SPEED *(Continued)*

TEST DATA AND RESULTS—*Continued*

Item		Test No. 1	Test No. 2
	Hourly Quantities		
46	Total gaseous fuel per hr. at standard pressure and temperature .... cu. ft.	....	
47	Total liquid fuel per hour .... lb.	....	
	Heat Consumption		
48	Total heat in fuel supplied per hr. (Item 46 × Item 40; or Item 47 × Item 41) .... B.t.u.	....	
	Indicator Diagrams		
49	Mean indicated pressure, average of all power cylinders .... lb. per sq. in.	....	
50	Maximum pressure, average of all power cylinders .... lb. per sq. in.	....	
	Speed		
51	Revolutions per minute .... r.p.m.	....	
52	Piston speed (mean) of power pistons .... ft. per min.	....	
	Power		
53	Indicated horsepower of all power cylinders .... I.Hp.	....	
54	Brake horsepower developed by whole engine by brake or dynamometer measurement .... B.Hp.	....	
55	Brake mean effective pressure .... lb. per sq. in.	....	
56	Horsepower input to motor or output of engine driving independent scavenging pump .... Hp.	....	
57	Horsepower input to motor or output of engine driving independent injection air compressor .... Hp.	....	
58	Horsepower input to motors of other independent auxiliaries (tabulated) .... Hp.	....	
59	Net or actual brake-horsepower of engine {Item 55 − Items (56 + 57 + 58)} .... B.Hp.	....	
	Economy Results		
60	Fuel consumption per I.Hp.-hr.	....	
61*	Fuel consumption per B.Hp.-hr	....	
62	Heat consumed per I.Hp-hr. (high value) .... B.t.u.	....	
63*	Heat consumed per B.Hp.-hr. (high value) .... B.t.u.	....	
	Efficiency Results		
64	Indicated thermal efficiency .... percent	....	
65	Brake thermal efficiency .... percent	....	
	Specimen Diagrams		
66	Sample indicator diagram from each power cylinder	....	
	Electric Data		
67	Average volts, each phase .... volts	....	
68	Average amperes, each phase .... amp.	....	
69	Total electrical output corrected for winding 1, 2, or 3 phase .... kv-a.	....	
70	Power factor .... percent	....	
71	Total electric output .... kw.	....	
72	Separate excitation .... kw.	....	
73	Net electric output .... kw.	....	
	Electric Economy and Efficiency Results		
74	Fuel consumed per net kw-hr.	....	
75	Heat consumed per net kw-hr. (Item 48 ÷ Item 74) .... B.t.u.	....	

*According to actual contractual conditions, the B.Hp. will be either Item 54 or Item 59.

When exciting current is obtained from the generator through a motor, or from some outside source, net output is current delivered at terminals minus current supplied for excitation.

Thermal Efficiency is the fraction of the heat consumption converted into work according to the formulas

$$\text{Brake thermal efficiency, } e_b = 2545/Q_b$$

$$\text{Indicated thermal efficiency, } e_i = 2545/Q_i$$

where Q_b and Q_i = respectively, high heating value of the fuel multiplied by the pounds or cubic feet of fuel supplied per brake horsepower hour or per indicated horsepower hour.

DATA AND RESULTS should be reported in accordance with the form on p. 16–48. for horsepower tests made at constant speed, and on the form below for horsepower tests made over the whole speed range of a variable speed engine.

DATA AND RESULTS OF INTERNAL-COMBUSTION ENGINE TEST AT VARIABLE TORQUE AND SPEED

The following items are recorded together with Items Nos. 7, 8, 10, 17, 18 from the constant speed test form.

Item							
10	Rated horsepower and speed, or speed at maximum torque						
11	Maximum safe speed						
		Test No. 1			Test No. 2		
	Run	1	2	3	1	2	3
14	Brake-horsepower . . . B.Hp.	...	...	...	...	...	...
15	Speed . . . r.p.m.	...	...	...	...	...	...
16	Total fuel . . . lb.	...	...	...	...	...	...
17	Duration of run . . . hr.	...	...	...	...	...	...
18	Fuel per hour . . . lb.	...	...	...	...	...	...
19	Fuel per hr. per B.Hp . . . lb.	...	...	...	...	...	...
20	Heating value of fuel, high value . . . B.t.u. per lb.	...	...	...	...	...	...
21	Heat consumed per hr. (Item 18 × Item 20) . . . B.t.u.	...	...	...	...	...	...
22	Heat consumed per hr. per B.Hp. (Item 21/Item 14) . . . B.t.u.	...	...	...	...	...	...
23	Brake thermal efficiency (2545/Item 22) . . . percent	...	...	...	...	...	...
24	Mean effective pressure equivalent to B.Hp. . . . lb. per sq. in.	...	...	...	...	...	...
25	Torque at 1-ft. radius equivalent to B.Hp . . . ft.-lb.	...	...	...	...	...	...
26	Fig. 1 Curves of r.p.m. plotted horizontal, against vertical . . .	...	...	...	...	...	...
	(*a*) Brake-horsepower (Item 14) . . .	...	...	...	...	...	...
	(*b*) Fuel per hour (Item 18) . . .	...	...	...	...	...	...
	(*c*) Fuel per hour per B.Hp. (Item 19) . . .	...	...	...	...	...	...
	(*d*) Brake thermal efficiency (Item 23) . . .	...	...	...	...	...	...
	(*e*) Brake mean effective pressure (Item 24) . . .	...	...	...	...	...	...
	(*f*) Torque (Item 25) . . .	...	...	...	...	...	...

14. TEST CODE FOR GAS PRODUCERS

Approved, 1928

This code is intended primarily for tests of producers whose gas is to be used for power purposes; it also may be used for producers generating gas for metallurgical and other heating purposes. The term fuel in the code includes all possible fuels.

OBJECTS for which tests of producers are made include:

a. Maximum and most efficient capacities of the producer plant and of its several elements. *b.* The efficiency of the producer as a whole in making gas, and that of its several elements. *c.* Ability of the producer to use a particular fuel in a particular way. *d.* Labor and power required for operation. *e.* Quantity of cooling water required. *f.* Cleanness of gas delivered. *g.* Results obtained by using different kinds and sizes of fuel in different ways. *h.* Amount of manual and skilled labor and of power required to operate producer and each of its auxiliaries. *i.* Reliability of producer and of its several elements and auxiliaries. *j.* Causes for faulty operation of the producer.

MEASUREMENTS depend somewhat on the object of the test, but in general include:

a. General data relative to type of producer and of each of its auxiliaries and their principal and important dimensions. *b.* Sizes and weights of each of the solid fuels, weight of oil, and weight of tar supplied to producer, and their physical characteristics and rates of firing; weight of ashes and refuse; weight of unburned carbon recovered therefrom. *c.* Ultimate and approximate analyses and calorific values of fuel and tar supplied to producer, and of hot gas, clean gas and tar delivered by producer plant; the specific gravity of each, and the analyses of ashes and refuse. *d.* Weights of water and their total heats above or below 68° F. supplied to producer and to each of its auxiliaries and their rates of supply. *e.* Volume, pressure, temperature and humidity of air delivered to producer. *f.* Temperatures of water delivered to producer and each of its auxiliaries; of the steam generated, and its quality; of gas delivered to and by each unit of the plant wherever change of temperature occurs. *g.* Barometric pressure and temperature of atmosphere, pressures of steam at the steam jet, in evaporator or jacket in the boiler, and elsewhere; the suctions and pressures of the gas before and after passing each auxiliary, and on being measured for calorific value and quantity. *h.* Humidity of air and steam delivered to producer. *i.* Quantity of gas of ...ure, absolute pressure, analysis, humidity, and calorific value delivered, and the rate of generation. *j.* Amounts of gross power used in operating scrubbers, purifiers, pumps, exhausters, and each of the other auxiliaries, and in charging, rotating, poking, and cleaning

producer. *k.* Amounts of manual and skilled labor used in operating the producer and each of its auxiliaries. *l.* Scaling properties of the water. *m.* Wear and tear and length of life of fuel-feeding devices, grates, valves, cleaning materials used, and of engines, motors, and other auxiliaries. *n.* Size of coal as determined by screening a sample.

AND APPARATUS required for producer tests are:

a. Measuring rods, rules, or tapes. *b.* Platform scales and weighing vessels for weighing fuels, ashes and water. *c.* Fuel-sampling and analyzing appliances and hydrometers. *d.* Fuel calorimeter. *e.* Steam calorimeter. *f.* Gas calorimeter, both sampling and continuous. *g.* Gas analyzing apparatus. *h.* Tar determinator. *i.* Soot determinator. *j.* Moisture determinator. *k.* Gas meter, Venturi meter, pitometer, or orifice meter for measuring gas output, together with the necessary thermometers, thermometer wells, manometers or pressure gages. *l.* Manometers, draft and pressure gages. *m.* Water meters, or other water measuring apparatus for measuring feed and scrubber water, and steam meters or equivalents for measuring steam used by each unit. *n.* Thermometers and wells. *o.* Instruments for measuring or determining power required to rotate and poke producer, to operate water- and oil-fuel pumps, tar extractor, scrubber, purifier and each of the other motive appliances. *p.* Necessary connections for each instrument or apparatus. *q.* Necessary instruments, etc. for calibrating above apparatus and instruments.

DURATION OF TEST in full time and complete tests should be such that probable error in weighing coal does not exceed 2%, or that total consumption of fuel is at least twice the weight of fuel in the producer when in normal operation. No test should be shorter than 24 hours, after the producer has attained desired and constant operating temperatures and conditions. Determination of clinkering tendency of fuel and quality of ash in large producers will require tests of five or more days duration.

Intermittent producers require the fuel bed to be entirely removed and rebuilt at regular intervals. Duration of test then should be that of one or more of the regular commercial operating cycles. Or if a complete cleaning and renewal occurs before total fuel consumption stipulated above occurs, the duration of test should be time elapsing between a corresponding number of successive renewals of the fuel bed.

STARTING AND STOPPING should occur at the conclusion of the time of regular cleaning. Both starting and stopping should be preceded for a period of not less than 8 hr. by the same regular working conditions as will characterize the test as a whole. Nc clinker should be on the walls or in the producer at beginning and end of test.

Continuous Producers with Grate and No Ash Bed.—Remove ashes and clinker from grate and lower part of furnace space. Stir fuel bed by means of a poker introduced through poker holes. Replenish producer with fuel to working depth; fill hopper level full. Note time and consider it as starting time. Then clean ash pit and proceed with regular work of the test, using weighed fuel.

To stop test, repeat cleaning operations, bring ash zone to the same depth and uniformity of content as at start, replenish producer with fuel to the same measured working depth as at start, and end test by filling hopper level full and noting time. Remove ashes and refuse from ash pit, weigh and sample.

Continuous Producers with Supporting Ash Beds.—Remove ashes until top of ash bed is lowered to normal working point. Introduce poker and break down any bridge or crust that may have formed, and make fuel bed homogeneous. Replenish producer with fuel to the working depth, which should be measured; fill hopper level full. Note time and consider this starting time. Then proceed with regular work of test, using weighed fuel.

To stop test, repeat above operations, ending with replenishing producer and filling hopper with weighed fuel. Note time and consider this as stopping time. Ashes and refuse finally removed, unless already dry, are to be dried before weighing. They may be weighed both wet and dry, and the water accredited to the water used during the test.

Intermittent Producers.—Thoroughly clean producer of its entire contents. Introduce a weighed supply of fuel, ignite it and build fuel bed to working condition, using weighed fuel. Note time and consider it as starting time. Proceed with regular work of the test.

When time approaches for ending test, burn fuel bed as low as practicable to prepare for cleaning. Note time and consider this stopping time. Completely empty producer, quench fire remaining in live coals, separate and weigh coke and ashes, and deduct weight of the coke from fuel as charged. Dry, weigh, and sample for analysis, the ashes and refuse. Analyze them for their contents of moisture, carbon, iron and earthy inorganic matter, making allowance for moisture in determining net weight of dry ashes.

THICKNESS OF FUEL BED AND ASH BED may be determined by the simultaneous use of several rods inserted vertically through observation holes in top of producer. The rods should be of $^3/_4$-in. pipe, with a stop-pin 3 in. long, 2 ft. from the upper end and long enough to extend down to the ash pit.

Top level of the green coal zone and height of fuel in producer at each place will be shown by distance rod is inserted below observation hole when resting on the fuel. When rod is forced down

to the ash pit, the depth of ash zone, thickness of incandescent and of green-fuel zones may be determined by length and position of red-hot portion of the rod. The gas zone may be noted by e sooty deposit on the upp ... results from the several rods should be reported.

NOTATION.—The following notation is used in the discussion below:

Standard Conditions are a temperature of ° F. and a pressure of 29.9212 in. of mercury at 32° F. (approximately 14.70 lb. per sq. in.).

$A_1, A_2 \ldots A_n$ = percentage of each combustible constituent of gas (standard conditions)

c_1, c_2 = sum of proportions, by volume, of constituent gases containing 1 atom and 2 atoms, respectively, of carbon

C_1 = carbon in 1 lb. of coal as fired

C_2 = carbon in 1 lb. of ashes and refuse

C_3 = carbon in ashes from 1 lb. of coal

C_4 = carbon gasified per 1 lb. of coal

C_5 = carbon in 1 cu. ft. of gas (standard conditions)

C_c = carbon in 1 cu. ft. of clean, dry gas, lb.

C_f = carbon in 1 lb. of fuel

$C_{p1}, C_{p2} \ldots C_{pn}$ = respectively, specific heat of each combustible constituent gas at constant pressure

C_n = percentage of net carbon in fuel

C_s = lb. of carbon per lb. of fuel in soot

C_t = lb. of carbon per lb. of fuel in tar

$d_1, d_2 \ldots d_n$ = respectively, density of each constituent gas

E = efficiency

E_c = cold gas efficiency

E_h = hot gas efficiency

$G_1, G_2 \ldots G_n$ = respectively, percentage of each constituent in gas delivered (standard conditions)

H_a = heat lost in combustible in ashes and refuse

H_c = B.t.u. per lb. of coal

H_g = B.t.u. per 1 cu. ft. of dry gas

H_m = Heat in moisture leaving producer

H_o = desired B.t.u. output of producer per hr.

H_r = Heat of scrubber water

H_s = Sensible heat in gas as delivered

H_w = Total heat above 68° F. of water vapor in gas at partial pressure due to its temperature

$h_1, h_2 \ldots h_n$ = respectively, calorific value of constituent gases (Standard conditions)

h_f = higher calorific value of 1 lb. of fuel, B.t.u.

h_g = higher calorific value of 1 lb. of dry gas

h_h = higher calorific value of gas, B.t.u. per cu. ft. (Standard conditions)

h_r = total heat of 1 lb. of superheated steam above 68° F. at temperature of gas leaving producer, and at its partial pressure

h_s = higher heat value of fuel (or equivalent heat) used to generate steam supplied from outside source

m = percentage of moisture in gas

P = absolute pressure of gas, lb. per sq. in. (Standard conditions)

P_a = standard atmospheric pressure

P_w = pressure of saturated water vapor at temperature t

t = temperature of gas as delivered, deg. F.

t_a = absolute temperature of gas, deg. F.

t_i = temperature of scrubber inlet water

t_o = temperature of scrubber outlet water

V = gas delivered per 1 lb. of fuel, cu. ft.

V_b = volume of gas generated per 1 lb. of dry fuel burned, cu. ft.

V_c = volume of clean gas per 1 lb. of fuel, cu. ft.

V_d = volume of dry gas, cu. ft.

V_e = equivalent volume (standard conditions)

V_m = measured volume

V_s = observed volume of gas leaving scrubber

W_a = weight of combustible in ashes, lb.

W_b = weight of dry gas per 1 lb. of fuel burned

W_c = weight of coal per hour to generate desired output

W_d = weight of dry gas per 1 lb. of net carbon gasified

W_f = weight of dry fuel

W_g = gross weight of dry gas delivered

W_r = weight of scrubber water

$w_1, w_2 \ldots w_n$ = weight of each of constituent gases, lb. per cu. ft. (standard conditions)

w_a = weight of ash in 1 lb. of coal, by analysis

w_c = weight of carbon per 100 cu. ft. of gas

w_g = weight of 1 cu. ft. of dry gas

SPOT TESTS are made where the cost of making full time test is prohibitive, or where frequent check is desired on the capacity, efficiency and character of output of producer at any time. Spot tests require an appliance for continuously sampling gas or for taking a series of spot samples, apparatus for analyzing gas produced and a continuous gas calorimeter or equivalent.

Average calorific value of gas may be calculated from volumetric analysis under standard conditions by using appropriate values of calorific value of the constituent gases. Average volumetric analysis for test period then is converted into an analysis by weight from which the total weight of carbon in 1 cu. ft. of standard gas may be determined.

Total weight of carbon in 1 cu. ft. of gas standard is $C_5 = 0.03127\ c_1 + 0.06298\ c_2$

Determine ultimate analysis and calorific value of the coal as actually fired, and also the analysis of the carbon content of the dried ash. Then weight of carbon lost in ashes per 1 lb. of coal as fired is $C_3 = w_a\{C_2/(1 - C_2)\}$.

Weight of carbon gasified per 1 lb. of coal is $C_4 = C_1 - C_3$

Cu. ft. of gas per 1 lb. of coal is $V = C_4/C_5$

Cold gas efficiency is $E_c = (C_4 \times \text{B.t.u. per cu. ft. of gas})/(C_5 \times H_c)$

B.t.u. output per lb. of coal is cu. ft. gas per lb. of coal × B.t.u. per cu. ft. of gas

Weight of coal to be fired per hour to generate desired output is $W_c = H_o/(E_c \times H_c)$. Weighing the coal input over any desired period of steady load, therefore, will permit determination of output capacity.

Spot Test with Tar Producing Fuels may be made if tar and soot per cu. ft. of outlet gas are determined. The above methods will apply only where cooling water is recirculated and tar and soot either are gasified, or separated and determined per cu. ft. of gas delivered. If tar is wasted, it first must be collected, weighed and sampled, and its weight compared with weight of fuel used during the same time, to determine amount of tar lost per lb. of coal gasified. If water is not recirculated, considerable CO_2 will be absorbed by the cold water washing the gas. Carbon so lost can be determined by comparing CO_2 content of gas samples taken simultaneously before and after passing the coolers, or by metering cooling water and analyzing it to determine the amount of carbon carried away as CO_2 in the water.

Short Tests may be made by determining items indicated in the record, p. 16–55. In such tests, it is unnecessary to meter steam used or gas made. Volume of gas made is determined by knowing amount of carbon gasified per hr., and average amount of carbon in 1 cu. ft. of gas as it leaves producer, including carbon in tar and soot. Amount of water vapor coming from all sources must equal water equivalent to hydrogen, free and combined, in gas, tar and water going into the gas-main from the producer. The amount of hydrogen coming from the fuel and from moisture in the air are known. The balance of the hydrogen enters producer as steam from boiler and as vapor from ash pit, water seal and elsewhere. From these data, gas and tar produced per ton of coal, total heat in gas, both sensible and potential, and thermal efficiency can be calculated.

Analyses and Calorific Test of Gas Output * should be determined by chemical analysis, calorimetric tests, or both. Continuous samples should be taken from delivery pipe at point nearest producer and other points as needed. Condensible tarry vapors and soots should be filtered out and their weights determined. Calorimetric tests of clean gas should be made with a Junkers type of calorimeter. Approximate determination of the composition of clean gas may be made with an Orsat apparatus and a complete determination with a Hempel apparatus. Determinations should be made not more than one-half hour apart, the time for collecting sample being not less than one quarter hour. If results of averages only are desired, taking of the sample may be made continuous and extended to several hours.

Measurement of Steam may be by weighing the water before it is evaporated, providing there are no leaks or other uses made of the steam; by a suitable steam meter; or by the Rateau or Napier formulas.†

CALCULATIONS. Total Volume of Gas Delivered should be calculated from chemical analyses of fuel and gas, or measured by the pitometer or orifice meter. With the latter it will be the continued product of the effective area, sq. ft., of the delivery pipe at the Pitot tube, or of the orifice, the average velocity of gas over that area, ft. per min., and the duration of the test in minutes. With unclean gases, calculation from chemical analysis is more likely to give correct results.

Equivalent volume of gas at standard atmospheric pressure is

$$V_e = (V_m \times P \times 35.89)/t_a$$

Gas leaving the scrubber is saturated with water vapor at the pressure due to its temperature. Volume of dry gas is

$$V_d = V_s \times \{(P - P_w)/P_a\} \times \{527.6/(t + 459)\}$$

The weight of dry gas per lb. of net carbon gasified in the producer is

$$W_d = \frac{11CO_2 + 8O_2 + 7(CO + N_2 + C_2H_4) + 4CH_4 + 0.5H_2}{3(CO_2 + CO + CH_4 + 2C_2H_4)}$$

where CO_2, O_2, etc. are the percentages by volume of the dry gas per lb. of net carbon gasified.

Dry gas per pound of fuel burned is

$$W_b = (W_d \times C_n)/100; \quad C_n = C_f - (C_t + C_s + C_1)$$

C_f, C_t, C_s and C_1 are taken in percent.

Volume of gas per 1 lb. of fuel is $V = W_b/w_g$. Values of w_g may be calculated from the values in Table 1 as shown below.

Weight of carbon per 100 cu. ft. of gas at standard conditions is

$$w_c = 0.03127\ (CO + CO_2 + CH_4) + 0.06284\ C_2H_4 + 0.6313\ C_2H_6$$

where CO, CO_2, etc. = percentage by volume of each constituent gases containing carbon; the numerical factors are the pounds of carbon in 1 cu. ft. of the several constituent gases.

Volume of clean gas delivered per lb. of fuel is

$$V_c = \{C_f - (C_t + C_s + C_a)\}/C_c$$

The percentage of tar and soot, C_t and C_s, is total weight of tar and soot including that collected from tar drips, soot catchers and scrubber water, divided by total weight of dry fuel used.

* See Analysis of Fuel Gas, S. W. Parr and F. E. Vandaveer, Univ. of Ill. Bull., vol. xxii, No. 8, Oct. 20, 1924.

† Rateau formula, $Q = 3.6P(16.3 - 0.96 \log P)$; Napier formula, $Q = PA/70$ (approx); Q = lb. steam per sec.; A = orifice area, sq. in.; P = absolute pressure, lb. per sq. in.

MOISTURE IN GAS is found by passing a sample of gas through a chloride of calcium tube and weighing the amount of moisture collected and absorbed, the gas first passing through a heater to raise its temperature to that of gas leaving producer, or to not less than 250° F. The gas first should have all tarry vapors removed by inserting a plug of cotton in the inlet end of the sampling tube, providing the temperature of the gases is below 212° F., and at the exit end of the tube. The tube should be kept at a temperature above the dewpoint of the gas, if the gas is extremely hot. The sample of gas from which tarry vapors have been removed should be metered.

Net Volume of Dry Gas Delivered is the difference between the volume delivered by the producer and that used by gas engines or other gas-consuming auxiliaries in the plant. If steam is generated in a special boiler instead of by waste heat from the producer, fuel and labor costs of the boiler should be charged to the producer. If the producer and its auxiliaries use purchased electric or other power, the cost of such power should be charged to the producer.

Gross Weight of Dry Gas may be obtained by

$$W_g = (G_1 w_1 + G_2 w_2 \ldots . G_n w_n)/100$$

Values of $w_1, w_2 \ldots . w_n$ should be at standard conditions. (See column 2, Table 1.)

Calorific Value of Dry Gas per cu. ft. should be obtained by a Junkers type of calorimeter. It also may be obtained by

$$H_g = A_1 h_1 + A_2 h_2 \ldots . A_n h_n$$

For values of $h_1, h_2 \ldots . h_n$, see Table 2.

Sensible Heat of the gas as delivered may be obtained by

$$H_s = (t - 68)\ (A_1 d_1 C_{p1} + A_2 d_2 C_{p2} + \ldots . A_n d_n C_{pn}) + H_w$$

Results may be calculated and reported per cubic foot of gas (saturated, moist or dry), either at specified or at standard temperature and pressure, or they may be reported per pound of fuel gasified (either as fired or dry). Sensible heat of fuel is product of its specific heat per lb. (0.20 to 0.24 for coal) and temperature difference between the fuel and the surrounding air, above or below 68° F. (*Continued on page 16–58*)

Table 1.—Specific Density of Gases

Gas	Lb. per Cu. Ft. at 68° F. and 29.9212 in. Hg at 32° F.	Lb. per Cu. Ft. at 32° F. and 29.9212 in. Hg at 32° F.	Lb. per Cu. Ft. at 60° F. and 30 in. Hg at 32° F.	Grams per Liter at 0° C. and 760 mm. Hg*
H_2	0.005228	0.005611	0.005323	0.08988
O_2	0.08312	0.08921	0.08462	1.4290
N_2	0.07274	0.07807	0.07406	1.2506
CO	0.07273	0.07806	0.07405	1.2504
CO_2	0.11499	0.12341	0.11707	1.9769
CH_4	0.04169	0.04475	0.04245	0.7168
C_2H_4	0.07331	0.07868	0.07464	1.2604
C_2H_6	0.07891	0.08469	0.08034	1.3566
C_3H_6	0.10913	0.11712	0.11110	1.8761
C_6H_6	0.20257	0.21740	0.20623	3.4825
H_2S	0.08846	0.09493	0.09005	1.5207
SO_2	0.17025	0.18272	0.17333	2.9269

* The figures given in the last column were supplied by the U. S. Bureau of Standards as the most reliable on file in 1927. The figures in the other columns have been derived from those in the last column.

Table 2.—Calorific Value of Gases

Gas	B.t.u. per Cu. Ft. at 68° F. and 29.9212 in. of Hg at 32° F.		B.t.u. per Cu. Ft. at 60° F. and 30 in. Hg at 32° F.		Calories per Liter at 32° F. and 29.9212 in. Hg at 32° F.		Calories per Gram-molecule *
	Dry	Saturated	Dry	Saturated	Dry	Saturated	
H_2	320.2	314	325.2	319	3.052	3.03	68.4
CO	318.8	313	323.7	318	3.039	3.02	68.0
H_2S	620	628	630	638	6.100	6.06	132.4
CH_4	......	......	......	......	......	......	210.8 or 212
C_2H_4	......	......	......	......	......	......	332.0 or 331.6
C_2H_6	......	......	......	......	......	......	368.4 or 370.0
C_3H_6	......	......	......	......	......	......	496.8 or 490.0
C_6H_6	....	......	......	......	......	......	787.2 or 784.0

* The data in this column were obtained from the U. S. Bureau of Standards as the most reliable on file in 1927. The data in the other columns were derived therefrom.

DATA AND RESULTS OF GAS-PRODUCER TEST

Only items marked * are used in report of short test

Item		
	Type of producer	
	Rated size, power, or capacity of producer	
9	Duration of test	
10	Trade name, kind, and size of solid fuel	
11	Trade name and kind of oil fuel	
	(a) Oil fuel, specific gravity at 68 deg. F	deg. A.P.I.
	(b) Oil fuel, specific gravity	referred to water
	(c) Oil fuel, weight per gal. at 68 deg. F	lb.
12	Outside diameter and height of producer	ft.
13	Inside diameter at hearth	ft.
14	Maximum inside diameter of lining of producer	ft.
15	Minimum inside diameter of lining of producer	ft.
16	Actual net diameter of gasification zone	ft.
17	Height from lowest level of grate to top of crown	ft.
18	Height from lowest level of grate to center of gas outlet	ft.
19	Vertical distance between highest and lowest levels of inclined grate	ft.
20	Gross dimension of grate, or of area upon which fuels and ashes are supported	ft.
21	Corresponding projected area of grate	sq. ft.
22	Gross diameter of central air-inlet through grate	ft.
23	Corresponding area of same	sq. ft.
24	Gross area of grate, if inclined	sq. ft.
25	Area of air spaces in grate	sq. ft.
26	Ratio of air spaces in grate to net, or gross, grate area	
27	Diameter of steam-blast inlet, or blower	in.
28	Corresponding area of same	sq. in.
29	Type of blower used	
30	Diameter of steam pipe to blast inlet, or blower	in.
31	Diameter of gas outlet	in.
32	Corresponding area of same	sq. in.
33	Area of fuel bed at maximum diameter	sq. ft.
34	Actual net area of fuel bed at gasification zone	sq. ft.
35	Area of water-heating surface in vaporizer, or jacket	sq. ft.
36	Method of poking producer	
37	Method of rotating producer	
38	Method of generating steam for producer	
39	Method of cooling gas	
40	Method of scrubbing and purifying gas	
41	Method of supplying water for cooling and cleaning	

TEST DATA AND RESULTS

Pressures

Item		
*42	Steam pressure, gage, lb. per sq. in. (a) in boiler .... (b) in vaporizer .... (c) as delivered to producer..	
43	Gas pressure, or suction, in main at point where gas is measured	in. of water
44	Pressure, or suction, at top of producer	in. of water
45	Pressure, or suction, beyond scrubber	in. of water
46	Pressure, or suction, beyond purifier	in. of water
47	Draft pressure, or suction, in ash pit, or in bottom of producer	in. of water
48	Barometric pressure of atmosphere .... in. of mercury at	deg. F.
49	Corresponding barometric pressure of atmosphere at 32 deg. F	in. Hg
50	Corresponding barometric pressure of atmosphere at 68 deg. F	in. Hg

Miscellaneous

Item		
*51	Relative humidity of air	percent
	Humidity of steam as delivered to the producer	percent wet
53	Average depth of fuel bed	
54	Average depth of ash bed	
55	Number of cleanings	
56	Average length of cleanings	
57	Average intervals between cleanings	
58	Average intervals between pokings	
59	Average length of pokings	
60	Ashes, clinkers, and refuse, wet or dry	lb.

Temperatures

Item		
61	Temperatures of feedwater entering vaporizer of producer	deg. F.
62	Temperature of feedwater entering outside steam generator or waste-heat boiler	deg. F.
63	Temperature of air in room	deg. F.
*64	Temperature of air entering producer	deg. F.
*65	Temperature of gas leaving producer	deg. F.
66	Temperature of tar and soot as delivered	deg. F.

(Continued on following page)

DATA AND RESULTS OF GAS-PRODUCER TESTS—*Continued*

Item

Temperatures—*Continued*

67 Temperature of gas in main at point where gas is measured .... deg. F.
68 Temperature of gas entering scrubber .... deg. F.
69 Temperature of gas leaving scrubber .... deg. F.
70 Temperature of gas entering purifier .... deg. F.
71 Temperature of gas leaving purifier .... deg. F.
72 Temperature of water entering scrubber .... deg. F.
73 Temperature of water leaving scrubber .... deg. F.
74 Temperature of water entering purifier .... deg. F.
75 Temperature of water leaving purifier .... deg. F.
76 Temperature of cooling water, or seal–water, entering producer .... deg. F.
77 Temperature of cooling water leaving producer .... deg. F.
78 Temperature of ashes when being removed .... deg. F.
*79 Temperature of: (*a*) steam delivered to producer .... deg. F.
(*b*) air-steam mixture in ashpit or bottom of producer .... deg. F.

Proximate Analysis of Fuel Supplied to Producer

*80 Moisture .... percent
*81 Volatile matter and its nature .... percent
*82 Fixed carbon .... percent
*83 Ash .... percent
*84 Sulphur, separately determined .... percent

100 percent

Item	Ultimate Analysis of Fuel	As Fired	Dry Coal
*85	Carbon (C)	percent	percent
*86	Hydrogen (H_2)	percent	percent
	Oxygen (O_2)	percent	percent
	Nitrogen (N_2)	percent	percent
*89	Sulphur (S)	percent	percent
*90	Ash	percent	percent
			100 percent

Analysis of Ashes and Refuse

*91 Carbon .... percent
*92 Inorganic or earthy matter and iron .... percent
*93 Moisture .... percent
*94 Nature and texture of ashes .... percent
*95 Fusing temperature of ashes .... percent
(*a*) Initial deformation temperature of ashes .... deg. F.
(*b*) Softening temperature of ashes .... deg. F.
(*c*) Fluid temperature of ashes .... deg. F.

Analysis of Dry Gas by Volume

*96 Carbon dioxide (CO_2) .... percent
*97 Carbon monoxide (CO) .... percent
*98 Oxygen (O_2) .... percent
*99 Hydrogen (H_2) .... percent
*100 Methane, or marsh gas (CH_4) .... percent
*101 Ethylene, or olefiant gas (C_2H_4) .... percent
*102 Hydrogen sulphide (H_2S) .... percent
*103 Sulphur dioxide (SO_2) .... percent
*104 Ammonia (NH_3) .... percent
*105 Nitrogen (N_2), by difference .... percent
*106 Total combustible in dry gas .... percent

Calorific Values by Calorimeter or Analysis

*107 Higher calorific value of 1 lb. of fuel as fired .... B.t.u.
*108 Higher calorific value of 1 lb. of dry fuel .... B.t.u.
*109 Higher calorific value of 1 lb. of combustible by analysis .... B.t.u.
*110 Higher calorific value of 1 cu. ft. of dry gas at 68 deg. and 29.9212 in. Hg .... B.t.u.
Lower calorific value of 1 cu. ft. of dry gas at 68 deg. and 29.9212 in. Hg .... B.t.u.

TOTAL, UNIT, AND HOURLY QUANTITIES AND RATES

Total Quantities

112 Gross weight of fuel charged, corrected for estimated differences in weights of fuel in producer at beginning and end of test .... lb.
113 Moisture in fuel (Item 80) .... percent
*114 Weight of dry fuel charged into producer, or dry fuel gasified .... lb.
*115 Dry fuel gasified: (*a*) per sq. ft. of main fuel bed .... lb.
(*b*) per hour .... lb.
116 Gross weight of fuel-as-fired used by auxiliary boiler for supplying steam to producer and its blower .... lb.
117 Gross weight of fuel-as-fired used by auxiliary boiler for supplying steam to pumps supplying water to producer, scrubber and purifier .... lb.

DATA AND RESULTS OF GAS-PRODUCER TESTS—*Continued*

Item

Total Quantities—*Continued*

118 Total gross weight of fuel-as-fired supplied to plant.... lb.
119 Total weight of dry fuel supplied to plant.... lb.
120 Weight of wet ashes and refuse: (a) from producer.... lb.
(b) from auxiliary boilers.... lb.
(c) from plant.... lb.
121 Weight of dry ashes and refuse: (a) from producer.... lb.
(b) from auxiliary boilers.... lb.
(c) from plant.... lb.
122 Dry ashes and refuse to dry fuel: (a) from producer.... percent
(b) from auxiliary boilers.... percent
(c) from plant.... percent
123 Weight of gross combustible charged: (a) into producer.... lb.
(b) into auxiliary boilers.... lb.
(c) into plant.... lb.
124 Weight of unburnt carbon in 1 lb. of dry ashes and refuse:
(a) from producer.... lb.
(b) from auxiliary boiler.... lb.
(c) from plant.... lb.
125 Weight of net combustible charged: (a) into producer.... lb.
(b) into auxiliary boilers.... lb.
(c) into plant.... lb.
126 Volume of air entering producer at..........deg. F.... cu. ft.
127 Humidity of air entering producer.... percent
128 Weight of dry air entering producer.... lb.
*129 Volume of gross gas delivered at..........deg. F. and..........in. of water-pressure (or suction).... cu. ft.
*130 Specific gravity of dry gas....
131 Weight of dry gas delivered at..........deg. F. and..........in. of water-pressure (or suction).... lb.
132 Moisture in gas leaving producer, mixed with 1 lb. of dry gas.... lb.
133 Moisture in gas leaving scrubber, mixed with 1 lb. of dry gas.... lb.
134 Equivalent volume of gross dry gas at temperature of 68 deg. F. and pressure of 29.9212 in. of mercury at 32 deg. F.... cu. ft.
135 Equivalent volume of gross dry gas at 68 deg. F. and 29.9212 in. Hg
(a) per lb. of dry fuel.... cu. ft.
(b) per lb. of net combustible.... cu. ft.
136 Equivalent volume of net dry gas at 68 deg. F. and 29.9212 in. Hg
(a) per lb. of dry fuel.... cu. ft.
(b) per lb. of net combustible.... cu. ft.
137 Weight of tar and soot delivered.... lb.
*138 Percentage of tar and soot in gas, referred to dry fuel.... percent
139 Residual tar and soot in clean dry gas.... grains per cu. ft.
140 Heat supplied to producer as steam: (a) from vaporizer.... B.t.u.
(b) from waste-heat boiler.... B.t.u.
(c) from separately fired boilers.... B.t.u.

Water Supplied, Total Quantities

141 To ashpit and water-seal for vaporization.... lb.
142 To ashpit and water-seal for cooling.... lb.
143 To vaporizer of producer or boiler supplying steam to producer.... lb.
144 As moisture in fuel.... lb.
145 As moisture in air.... lb.
146 Steam supplied to air-blower and to producer.... lb.
147 Total water and steam supplied to producer and used for gasification.... lb.
148 Total water supplied to producer and its auxiliaries for cooling.... lb.
149 Total water supplied to scrubber.... lb.
150 Total water supplied to purifier.... lb.
151 Total water supplied to auxiliary boiler for steam for steam pumps.... lb.
152 Total water supplied to waste-heat boiler.... lb.
153 Total water supplied for other auxiliaries.... lb.
154 Total water supplied to producer and its auxiliaries.... lb.

Efficiencies

155 Gross efficiency of producer, based on fuel as fired and steam supplied.... percent
156 Net efficiency of producer, based on fuel as fired and steam supplied.... percent
157 Gross efficiency of producer, based on dry fuel and steam supplied.... percent
158 Net efficiency of producer based on dry fuel and steam supplied.... percent
159 Gross efficiency of producer, based on net combustible and steam supplied.... percent
160 Net efficiency of producer, based on net combustible and steam supplied.... percent

(Continued on following page)

DATA AND RESULTS OF GAS-PRODUCER TESTS—*Continued*

Heat Balance Based on 1 lb. of Dry Fuel and the Products Therefrom

Item			
161	Higher calorific value of 1 lb. of dry fuel	B.t.u. =	100 percent
162	Higher heating value of dry gas from 1 lb. of dry fuel	B.t.u. =	... percent
163	Sensible heat in hot dry gas above 68 deg. F.	B.t.u. =	... percent
164	Total heat of moisture in gas above 68 deg. F.	B.t.u. =	... percent
165	Heat removed by water of scrubber	B.t.u. =	... percent
166	Heat removed by purifier	B.t.u. =	... percent
167	Heat lost in residual tar and soot of dry gas	B.t.u. =	... percent
168	Heat lost as unburnt fuel in ashes	B.t.u. =	... percent
169	Heat lost by radiation and otherwise unaccounted for	B.t.u. =	... percent
170	Sensible heat of fuel above or below 68 deg. F.	B.t.u. =	... percent
171	Sensible heat of air above or below 68 deg. F	B.t.u. =	... percent
172	Total heat above 68 deg. F. in water and steam supplied	B.t.u. =	... percent
173	Sensible heat of ashes and refuse	B.t.u. =	... percent
174	Heat lost to cooling water in producer and auxiliaries	B.t.u. =	... percent

Power Required for Auxiliaries for

Item		Per Hour		
175	Supplying air, as by turbo-blower	lb. steam	........Hp.	
176	Water pumps	lb. steam	........Hp.	
177	Tar pumps	lb. steam	........Hp.	.kw.
178	Scrubbers	lb. steam	........Hp.	.kw.
179	Purifiers	lb. steam	........Hp.	.kw.
180	Poking fire	lb. steam	........Hp.	.kw.
181	Rotating producer	lb. steam	........Hp.	
182	Exhauster	lb. steam	........Hp.	
183	Charging producer	lb. steam	........Hp.	
184	Total power required	lb. steam	........Hp.	..kw.

Manual Labor Supplied and the Cost Thereof During.........hours

Item		Man-hours	Rate of pay, cts. per hr.	Cost
185	Generating steam	....	....	..
	(*a*) for producer and its blower	....	....	$.
	(*b*) for pumps	....	....	$.
186	Charging producer	....	....	$.
187	Poking fire	....	....	$.
188	Rotating producer	....	....	$.
189	Cleaning producer	....	....	$.
190	Operating auxiliaries	....	....	$.
191	Recovering unburnt fuel	....	....	$.
192	Other operations, as removal of ashes	....	....	$.
193	Total labor supplied for operation of plant.	....	....	$.

Gross and Net Costs, Exclusive of Overhead

Item		Rates	Per 1000 cu. ft. Net Gas at 68 deg., 29.9212 in. Hg	Per 1,000,000 B.t.u. of Available Heat Energy in Net Gas at 68 deg., 29.9212 in. Hg	Per 1,000,000 B.t.u. of Available Heat Energy in Hot Gas at .... deg. F.
194	Fuel as fired	$.... per 2000 lb.			
195	Fuel as fired	$.... per gallon			
196	Water at	$.... per M. cu. ft.			
197	High pressure steam at..	$.... per M. lb.			
198	Exhaust steam at	$.... per M. lb.			
199	Electricity, a.c. or d.c. at	$.... per kw.-hr.			
200	Total labor supplied				
201	Total gross cost of labor, fuel, water, steam, and current used				

202 Total gross cost of labor, fuel, steam, and current used per ton (2000 lb.) or gallons consumed..

(*Continued on following page*)

EFFICIENCIES.—The hot gas efficiency of the producer is the ratio of the total heat carried by the gas (heat of combustion + sensible heat above 68° F.) to the sum of the heat supplied in its production. The total heat of hot unclean gases includes: Heats of combustion of the constituent gases and of the carbonaceous and tarry vapors, liquids

DATA AND RESULTS OF GAS-PRODUCER TESTS—*Continued*

Item	Credits	Per 1000 cu. ft. Net Gas	Per 1,000,000 B.t.u. of Available Heat Energy in Net Gas
203	High pressure steam at $...... per M. lb.	....	....
204	Low pressure steam at $...... per M. lb.	....	....
205	Tar and soot at....... $...... per M. lb.	....	....
206	Ammoniacal liquor (sp. gr. = ...) at..... $...... per M. cu. ft.	....	....
207	Total credits..........	....	....
208	Total net cost of labor, fuel, water, steam, current used.......	....	....
209	Total net cost of labor, fuel, water, steam, and current used per ton (2000 lb.), or per gallon consumed.. $........		
	Summary		
210	Total capacity of plant in tons (2000 lb.), or gallons, of fuel consumed in 24 hr........		
211	Net heat output in millions of B.t.u. delivered in 24 hr........................ B.t.u.		
212	Total net cost of generating same, exclusive of overhead charges............ $........		

and solids, their superheat above the temperatures corresponding to their partial pressures. the latent heats of the constituent vapors, including water vapor, and their sensible heats above 68° F. In determining efficiency, total heat used to generate steam supplied to the producer from an outside source, must be added to heat of the fuel supplied in the same time.

Gross-dry-fuel is 1 lb. less weight of moisture in 1 lb. as determined by analysis. Net-dry-fuel is gross-dry-fuel less weight of unburned carbon in ashes and refuse produced by 1 lb. of fuel-as-fired. Gross combustible is 1 lb. less sum of weights of moisture and ash in 1 lb. as determined by analysis. Net combustible is gross combustible less weight of unburned carbon in ashes and refuse produced by 1 lb. of fuel-as-fired.

Efficiency may be referred to gross or net gas delivered (saturated, moist or dry), to fuel-as-fired, to gross-dry- or net-dry-fuel, or to gross or net combustible. The calorific value used must be that of the fuel as stated. The gases may be taken as saturated, moist or dry under standard conditions. Efficiency may be calculated by

$$E = h_h \times V/(h_f + h_s)$$

Efficiency of conversion and cleaning, or gross efficiency, is found by using total gross volume of gas delivered. Efficiency of plant, or net efficiency, is found by using net volume of gas delivered. Net volume is gross volume less the actual gas used, or its equivalent, to operate the auxiliaries. If total heat of cold gas be taken, the efficiency is the cold gas efficiency. If the total heat of the hot gas be taken, the efficiency is the hot gas efficiency. Cold gas efficiency and hot gas efficiency are determined by the following formulas:

$$E_c = (A_1 h_1 + A_2 h_2 \ldots + A_n h_n)/(h_f + h_s)$$
$$E_h = \{(A_1 h_1 + A_2 h_2 \ldots + A_n h_n) + H_s\}/(h_f + h_s)$$

$A_1, A_2 \ldots A_n$ are taken as percentages of cold gas delivered per lb. of fuel; $h_1, h_2 \ldots h_n$ are based on cold gas. Unless otherwise stated, higher calorific value always should be used. Cold and hot gas efficiencies, both gross and net, however, may be determined for both higher and lower calorific values of the gas.

HEAT BALANCE quantities based on dry gas are computed as follows:

Heating value of dry gas $= H_g = V_b \times h_g$.

Sensible heat $= H_s = V_b \times C_{p(\text{mean})} \times (t - 68)$. (See also p. 16–54.)

Heat carried off by scrubber water $= H_r = W_r \times (t_o - t_i)/W_f$.

Heat in moisture leaving producer $= H_m = W_b \times m \times h_r$.

Heat lost in combustible in ashes and refuse $= H_a = (W_a/W_f) \times 14{,}544$. The calorific value of the combustible in the ashes may be checked by calorimeter or by chemical analysis.

If excess steam is generated in a waste heat boiler by waste heat of the producer, and if tar, soot and ammonia are recovered along with the gas, their amounts should be included and reported in the output, and their net values deducted in the financial report of the cost of making gas.

DATA AND RESULTS should be reported in the form p. 16–55, using only those items that conform to the object of the test.

15. TEST CODE FOR CENTRIFUGAL COMPRESSORS AND EXHAUSTERS

Approved, 1934

This code covers the conducting and recording of tests on centrifugal compressors and exhausters, whose purpose is to determine all or some of the following essential quantities, reduced to some specified conditions:

1. Inlet and discharge pressures, expressed as total pressures in inches of water for fans, and pounds per square inch for compressors and exhausters.

2. Quantity of air or gas compressed and delivered, expressed in units of volume per unit of time, under the specified conditions of density, temperature, and pressure at the intake of the machine. The unit for this quantity usually is the cubic foot per minute.

3. Power applied to the shaft of the compressor or exhauster or fan. The unit for this quantity usually is the horsepower.

4. The rate at which steam, electrical energy or fuel, depending upon the method of drive, is consumed by the driving element. The unit for this quantity usually is pounds of steam per hour, kilowatts, or pounds of fuel per hour. This also may be expressed as the pounds of steam, kilowatt-hours, pounds of fuel, or available B.t.u. per 100 cu. ft. of air or gas at contract inlet conditions.

PREPARATIONS FOR TESTS.—Dimensions and physical condition of the machine to be tested, as listed in the report form, should be ascertained and reported, and also similar information concerning all associated parts of the plant which may have a bearing on the results of the test.

Items which should be recorded in this connection are: Internal condition, particularly with reference to clearance in wearing rings, presence and extent of deposits on impellers and casing, and other factors that can affect the performance of the machine; dimensions of inlet and discharge openings, particularly where pressure readings are taken; number of stages in the machine and whether it is uncooled or provided with internal or external cooling. A sketch showing the location of the various instruments and testing apparatus in relation to the machine under test should be prepared.

Preliminary Tests may be run to: *a.* Determine whether the machine is in suitable condition for conduct of acceptance test. *b.* Check all instruments. *c.* Train personnel.

OPERATING CONDITIONS.—The most important factors affecting operating conditions are: *a.* Inlet gas temperature, pressure and composition. *b.* Quantity of air or gas handled by machine. *c.* Total pressure rise or pressure ratio. *d.* Speed of the machine.

Operating conditions should be as near as possible to those specified. Test results with gas having practically the same value of C_p/C_v as specified, may be corrected to specified conditions when volume, speed and density at inlet are within 3%, 5% and 10%, respectively, of the corresponding specified conditions, and when the combined variations result in (*inlet volume rate/speed*) differing by not more than 4% from the specified values.

STARTING AND STOPPING.—The machine should be operated under test conditions sufficiently long to establish uniform conditions of pressure, volume, temperature and speed. The duration of the test depends on method of driving the machine, on how completely the compressor characteristic is to be determined, and on the uniformity of the various readings. Each test in which volume, pressure and speed are intended to be constant shall run for at least 30 minutes. During this period at least six readings of all instruments which have an important bearing in the calculation of results shall be taken. The readings of each set should be taken as nearly simultaneously as possible. The test must be continued until a period of 30 consecutive minutes produces a consistent set of records.

INSTRUMENTS AND APPARATUS required are: *a.* Barometer. *b.* Thermometers. *c.* Pressure gages, or mercury or water U-tubes. *d.* Differential gages. *e.* Impact tubes, pitot tubes, manometers. *f.* Flow nozzles. *g.* Tachometers, revolution counters, or other speed counting devices. *h.* Appropriate electrical instruments. *i.* Appropriate instruments for measuring total steam flow, pressure, temperature, vacuum, etc., if performance of a turbine or engine is to be determined. *j.* Hydrometer or psychrometer. *k.* Gas analysis apparatus. *l.* Dynamometer for determining power absorbed at coupling of machine, if separate performance data are required for driving and driven elements.

Power delivered to the compressor or exhauster shaft, when direct connected, shall be the power output of the driving element. When not direct connected, corrections shall be made for the losses between driving element and the machine under test. Without special precautions, the electric dynamometer may be an unreliable instrument. It is recommended that the machine under test be driven by a direct-current dynamometer and that the dynamometer horsepower be checked by the calculation of horsepower from the electrical readings by the method of losses. It is recommended further that a dynamometer should not be used for less than 33% of its rated

horsepower. If a transmission dynamometer is not available, the preferred method of determining power input to the compressor shall be the use of a direct connected calibrated electric motor, provided the calibration tests of this motor are made in accordance with the standard rules of the A.I.E.E. and are frequently re-checked. Curves shall be made up from the calibration tests, translating kilowatt input into shaft horsepower output. Power input to the motor during a compressor test then shall be measured in accordance with the standard rules of the A.I.E.E. and the corresponding power output as shown on the calibration curve shall be taken as the power input to the compressor.

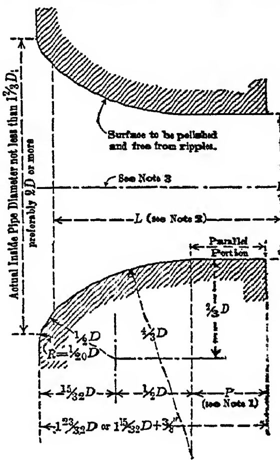
COMPUTATION OF RESULTS.—Pressure Rise shall be the difference between the absolute discharge total pressure and the absolute total pressure at the inlet to the machine. If the difference between the velocity pressure at the inlet and discharge is less than 0.2% of the pressure rise, as defined above, static pressures given by pressure gages may be used.

Pressure Ratio shall be the absolute discharge total pressure divided by the absolute inlet total pressure.

The Volume Rate discharged by compressors and exhausters shall be computed by the flow nozzle method.

EFFICIENCY of a compressor or exhauster is the ratio of the energy converted into useful work to the energy supplied. Efficiency $= (P_g/P) \times 100$, where P = shaft horsepower input, as determined by dynamometer or calibrated electric motor, and P_g = gas horsepower output as explained below.

Efficiency may be computed on the basis either of adiabatic or isothermal compression. Overall efficiency is the ratio of theoretical (adiabatic or isothermal) horsepower to the horsepower input of the driving element.



Note 1. $P = {}^3/_4\ D$ for nozzles $1\ {}^1/_2$ in. and over $= {}^1/_2 D + {}^3/_8$ in. for smaller nozzles.

Note 2. For pipe diameters of $1.66D$ to $1.9D$, L is to be made such as to make diameter of intersection of inlet curve with nozzle face equal to $1.05\times$ actual inside pipe diameter.

Note 3. Nozzle to be centered with pipe by means of close fitting bolts, dowels or counterbore, or careful setting and checking; within ${}^1/_{25}\ D$ for pipe diameters $3D$ to $4D$, and within $0.01D$ for pipe diameters less than $4D$.

FIG. 1. Flow Nozzle

Gas and Air Horsepower for Adiabatic Compression and Delivery is determined as follows:

$$P_g = [144\ k/\{33{,}000\ (k-1)\}]\ p_i\ q_i\ X_k \quad \ldots\ldots\ldots \quad [1]$$

where $k = C_p/C_v$ (specific heat ratio); p_i = total pressure at nozzle inlet, lb. per sq. in., abs.; q_i = volume rate at inlet, cu. ft. per min.; X_k = a constant $= r^{(k-1)/k} - 1$; r = pressure ratio. For normal air, or any gas where $(k-1)/k = 0.283$, formula [1] becomes

$$P_g = 0.01542\ p_i\ q_i\ X \quad \ldots\ldots\ldots\ldots \quad [2]$$

where $X = r^{0.283} - 1$.

Gas Horsepower for Isothermal Compression and Delivery is determined by

$$P_g = (144\ p_i/33{,}000) \times q_i \times \log_e (p_f/p_i) = 0.01005\ p_i\ q_i \times \log_{10} (p_f/p_i) \quad \ldots \quad [3]$$

where p_f = discharge pressure. Other notation as before.

Conversion of Test Results to Conditions Specified may be necessary when it is not possible to maintain the same speed for each of the series of tests, or where conditions make tests impractical at the speed at which characteristic curves are desired, or where it is not possible to have conditions of pressure, temperature and gas density in accordance with specified conditions. Test readings then may be corrected to specified conditions by the method given below. Corrections should be applied only within a reasonably narrow range. It is desirable to maintain the ratio (*inlet volume rate/speed*) as nearly as possible the same as when compressor is operating under specified conditions.

Table 1.—Formulas for Flow of Air and Gas through Nozzles

ı the formulas below, the various symbols have the following significance: efficient of discharge = 0.97 for nozzle throat diameter of 0.5 to 1.0 in.; 0.98 above 1 to 2 in.; 0.99 above 2 up to 6 in.; 0.995 above 6 in. C_p, C_v = specific heat at constant pressure and constant volume, respectively. D diam., in. $k = C_p/C_v$. $K = (k-1)/k$ = 0.283 for air and diatomic gases. p_1 at nozzle inlet. p_s = static pressure at nozzle inlet. p_2 = static pressure at nozzle outlet absolute pressure to which volume flow is to be computed or referred. p_b = barometric e. q_1 = volume of flow at conditions in inlet pipe corresponding to p_s and T_1. flow corresponding to conditions p_3 and T_3. $r = p_1/p_2$ for inlet total pressure static pressure. T_1 = absolute temperature at nozzle inlet. T_3 = absolute tem hich volume flow is to be computed or referred. V = velocity, ft. per sec., in inlet p_s is measured. w = flow, lb. per sec. $X = (r^{0.283} - 1)$. Δ = differential pressure across nozzle = $p_1 - p_2$, or $p_s - p_2$. ρ_1 = density in inlet pipe, lb. per cu. ft. corresponding to a given gas at T_1 and p_1 or p_s. All pressures are in lb. per sq. in. absolute. All temperatures are absolute = deg. F. + 459.6.

AFFECTED FLOW.—Absolute Pressure of Low-pressure Region more than 50% of Inlet Pressure.

Inlet Total Pressure	Divisor for Inlet Static Pressure (See Note 1)
Formulas for Gases and Air with Small Pressure Difference.	
FORMULAS FOR ANY PERFECT GAS WITH $\Delta <$ APPROXIMATELY 0.10 p_1	
$w = 0.5239\, cD^2 \sqrt{\Delta p_2 \rho_1/p_1}$	$\sqrt{1 - (D/D_1)^4}$ *
$q_3 = (31.43\, cD^2 T_3/p_3 T_1) \sqrt{\Delta p_2 p_1/\rho_1}$	$\sqrt{1 - (D/D_1)^4 \{1 - (2/k)(\Delta/p_s)\}}$ †
FORMULAS FOR AIR	
$w = 0.8596\, cD^2 \sqrt{\Delta p_2/T_1}$	$\sqrt{1 - (D/D_1)^4}$ *
$q_3 = (19.16\, cD^2 T_3/p_3) \sqrt{\Delta p_2/T_1}$	$\sqrt{1 - (D/D_1)^4 \{1 - (1.434/p_s)\}}$
Formulas for Gases and Air Accurate to within 0.2% up to Critical Pressure.	
FORMULAS FOR ANY PERFECT GAS WITH Δ BETWEEN APPROXIMATELY 10% AND 50% OF p_1 OR p_s	
$w = 0.5258\, cD^2 \sqrt{\rho_1 \Delta \{1 - (3/2k)(\Delta/p_1)\}}$ $q_3 = (31.55\, cD^2 p_1 T_3/T_1 p_3) \sqrt{(\Delta/\rho_1)\{1 - (3/2k)(\Delta/p_1)\}}$	$\sqrt{1 - (D/D_1)^4 \{1 - (2/k)(\Delta/p_s)\}}$
FORMULAS FOR PERFECT DIATOMIC GASES WITH Δ BETWEEN APPROXIMATELY 10% AND 50% OF p_1 OR p_s	
$w = 0.5258\, cD^2 \sqrt{\rho_1 \Delta \{1 - 1.0755\,(\Delta/p_1)\}}$ ($\sqrt{(\Delta/\rho_1)\{1 - 1.0755\,(\Delta/p_1)\}}$	$\sqrt{1 - (D/D_1)^4 \{1 - (1.434/p_s)\}}$
FORMULAS FOR AIR WITH Δ BETWEEN APPROXIMATELY 10% AND 50% OF p_1 OR p_s	
$w = 0.8627\, cD^2 \sqrt{(\Delta/T_1)(p_2 - 0.0755\,\Delta)}$ $q_3 = (19.23\, cD^2 T_3/p_3) \sqrt{(\Delta/T_1)(p_2 - 0.0755\,\Delta)}$	$\sqrt{1 - (D/D_1)^4 \{1 - (1.434\,\Delta/p_s)\}}$
Theoretical Formulas for Gases and Air.	
FORMULAS FOR ANY PERFECT GAS WITH Δ BETWEEN APPROXIMATELY 10% AND 50% OF p_1 OR p_s	
$w = (0.5250\, cD^2 p_2/\sqrt{K}) \sqrt{\rho_1/p_1} \sqrt{(p_1/p_2)^K \{(p_1/p_2)^K - 1\}}$ $q_3 = (31.50\, cD^2 p_2 T_3/p_3 T_1 \sqrt{K})$ $\times \sqrt{p_1/\rho_1} \sqrt{(p_1/p_2)^K \{(p_1/p_2)^K - 1\}}$	$\sqrt{1 - (D/D_1)^4 \{(p_s/p_2)^K/(p_s/p_2)\}^2}$ $K = (k-1)/k$
FORMULAS FOR PERFECT DIATOMIC GASES WITH Δ BETWEEN APPROXIMATELY 10% AND 50% OF p_1 OR p_s	
$w = 0.987\, cD^2 p_2 \sqrt{\rho_1/p_1} \sqrt{X(X+1)}$ $q_3 = (59.22\, cD^2 p_2 T_3/p_3 T_1) \sqrt{p_1/\rho_1} \sqrt{(X+1)X}$	$\sqrt{1 - (D/D_1)^4 \{(X+1)/(p_s/p_2)\}^2}$ Here $r = p_s/p_2$; $X = (r^{0.283} - 1)$
FORMULAS FOR AIR WITH Δ BETWEEN APPROXIMATELY 10% AND 50% OF p_1 OR p_s	
$w = 1.619\, cD^2 p_2 \sqrt{\{X(X+1)/T_1\}}$ $q_3 = \{(36.09\, cD^2 p_2 T_3)/(p_3 \sqrt{T_1})\} \sqrt{X(X+1)}$	$\sqrt{1 - (D/D_1)^4 \{(X+1)(p_s/p_2)\}^2}$ Here $r = p_s/p_2$; $X = (r^{0.283} - 1)$

* Divisor for any perfect gas, or air, with Δ less than 2% of p_s, and D/D_1 less than 1/3.
† Divisor for any perfect gas, or air, with Δ less than 10% of p_s, and any D/D_1.

Note 1.—When inlet pressure is measured as static pressure use the proper formula for inlet total pressure with the following changes: 1. Substitute p_s for p_1; 2. Divide the right-hand term of the formula by the divisor given by the formula in the static inlet pressure column.

Table 1.—Formulas for Flow of Air and Gas through Nozzles.—*Continued*

UNAFFECTED FLOW.—Absolute Pressure of Low Pressure Region Less than about 56% of Inlet Pressure.

Inlet Total Pressure	Inlet Static Pressure
FORMULAS FOR ANY PERFECT GAS	First compute approximate flow by using p_s for p_1 in formula for total inlet pressure. Then compute approximate velocity V by one of the following:
$w = 0.5250\,cD^2\sqrt{p_1\rho_1}\{2/(k+1)\}^{1/(k-1)}\sqrt{k/(k+1)}$	$V = 144\,q_1/60\,D_1^2(\pi/4)$
$q_3 = 31.50\,cD^2\,(T_3 p_1/p_3\,T_1)$ $\times\sqrt{p_1/\rho_1}\{2/(k+1)\}^{1/(k-1)}\sqrt{k/(k+1)}$	$= 144\,w/\rho_1\,D_1^2(\pi/4)$.
FORMULAS FOR ANY PERFECT DIATOMIC GAS	ρ_1 = density in inlet pipe corresponding to T_1 and p_s. Then
$w = 0.2539\,cD^2\sqrt{p_1/\rho_1}$	$(p_1 - p_2) = \rho_1 V^2/144\,(2g)$
$q_3 = 15.23\,cD^2(T_3 p_1/p_3\,T_1)\sqrt{p_1/\rho_3}$	$= \rho_1\,V^2/9266$.
FORMULAS FOR AIR	This is added to p_s, giving p_1, which is used in the proper formula for total inlet pressure.
$w = 0.4165\,cD^2\,p_1/\sqrt{T_1}$	
$q_3 = 9.284\,cD^2\,p_1\,T_3/p_3\sqrt{T_1}$	

In the following formulas, subscript c denotes specified conditions, subscript t denotes test conditions, subscript tc denotes values which the test shows that the machine will give when operating under specified conditions.

The corrected value X_{tc} is found from the test value X_t by the formula

[4]

where the symbols have the same significance as in formulas [1] to [3] and where n = revolutions per minute of the machine under test and ρ = density of gas. $r_t = p_{ft}/p_{it}$, from which the value of X_t may be found.

The corresponding pressure ratio, $r_{tc} = p_{ftc}/p_{ic}$ then is found from the corrected value X_{tc} and compared with specified ratio $r_c = p_{fc}/p_{ic}$. The corrected pressure rise $(p_{ftc} - p_{ic})$ should approximate the specified pressure rise $(p_{fc} - p_{ic})$.

If the gas analysis under specified and test conditions is the same, so that molecular and gas constant R are the same, formula [4] reduces to

[5]

Corrected power P_{tc} is given by

$$P_{tc} = \ldots (k/k - \ldots \times \frac{\ldots}{p_{it}} \times \frac{\ldots}{q_{it}} \times \ldots$$

If the specified and test gas analysis is the same, then

$$P_{tc} \ldots \qquad [7]$$

The above formulas give the power which the machine will consume when operating at specified inlet conditions of volume, pressure, temperature and density, and speed, and on the basis of the corrected pressure ratio, r_{tc} actually produced by the machine.

If the corrected discharge pressure, p_{ftc}, produced approximates specified discharge pressure, p_{fc}, the specified power, P_c, shall be multiplied by the ratio (X_{tc}/X_c). The product, $P_c \times (X_{tc}/X_c)$, should approximate the power P_{tc}.

When tests are conducted to obtain the complete characteristics of the machine, a series of tests should be conducted as follows: *a*. Inlet suction or discharge pressure (not both), and speed are held constant while the volume is changed for each point on the curve. *b*. Similar series are run for different speeds.

The performance of the unit or its characteristic is best presented in curves using inlet volume as abscissa and discharge pressure, horsepower and efficiency as ordinates. Curves should show the relation between each of the following pairs of quantities for each constant speed: 1. q_i and p_f, or $(p_f - p_i)$. 2. q_i and P. 3. q_i and efficiency. Steam consumption in pounds per hour or kilowatts may be used as ordinate, instead of the horsepower, in presenting the q_i and P relations, and the efficiency then should represent that of the overall unit, including the driving element.

FLOW-MEASUREMENT WITH NOZZLES.*—Flow may be measured by means of the nozzle shown in Fig. 1, with an accuracy within 2%, with thoroughly experienced personnel. The formulas given in Table 1 are used to calculate results.

* Tentative, 1935.

REPORT OF TESTS OF COMPRESSORS AND EXHAUSTERS

Item

DESCRIPTION AND DIMENSIONS

8 Type of compressor (radial or axial flow, open or shrouded, etc.).
(a) Serial number and builder's model number.
(b) Number of stages...
(c) Type of cooling (casing, diaphragm, inter- and after-coolers, giving sizes, capacities, etc.)...

Type of driver (whether motor, turbine, motor and gear, internal combustion engine, etc., giving capacity and descriptive information, guarantees, etc.)...

10 Governor (whether arranged to maintain constant pressure, constant volume, or constant speed, and whether operated by water, air, oil, or mechanically)...

11 Rated discharge pressure........lb. per sq. in., ...in. Hg, or in. water..

12 Rated inlet pressure...............lb. per sq. in., ...in. Hg, or in. water..

13 Rated inlet temperature...deg. F.

14 Rated inlet gas density...lb. per cu. ft.

15 Composition of gas for which intended...

16 Rated speed...r.p.m.

17 Rated volume at inlet conditions...cu. ft. per min.

18 Diameter or dimensions of inlet opening...in.

19 Area of inlet opening...sq. ft.

20 Diameter or dimensions of discharge opening...in.

21 Area of discharge opening...sq. ft.

22 Instrument numbers, titles and ranges (sketch of locations)...

AVERAGES OF DATA AS RECORDED

GENERAL

23 Test number...

24 Date...

25 Time...

26 Conditions peculiar to particular test...

27 Readings by...

28 Barometer...in. Hg

29 Room temperature...deg. F.

30 Elevation of test ins...ft.

COMPRESSOR

31 Total inlet pressure (above or below atmosphere)......in. water, in. Hg or lb. per sq. in.

32 Inlet temperature......... (a) Dry bulb.......deg. F.; (b) Wet bulb.......deg. F.

33 Total discharge pressure (above or below atmosphere)...in. water, in. Hg, or lb. per sq. in.

34 Discharge temperature...deg. F.

35 Composition of gas (if other than air)...percent by volume

36 Speed...r.p.m.

37 Shaft horsepower (dynamometer reading only)...S.Hp.

38 Cooling water quantities in gal. per min. or lb. per hr.:
(a) Jackets......... (b) Intercoolers 1, 2, 3, etc.........; (c) Aftercooler........

39 Water temperatures in deg. F.:
(a) To jackets... (b) From jackets...
(c) To intercooler 1, 2, 3, etc... (d) From intercooler 1, 2, 3, etc...
(e) To aftercooler... (f) From aftercooler...

40 Throat diameter of flow nozzle...in.

41 Diameter of approach pipe...in.

42 Nozzle inlet total pressure...in. Hg, in. water, or lb. per sq. in.

43 Nozzle throat static pressure...in. water, in. Hg, or lb. per sq. in.

44 Nozzle inlet temperature...deg. F.

DRIVER

a. Steam turbine

45 Pressure of steam to throttle...lb. per sq. in., abs.

46 Quality of steam to throttle...deg. F., or percent moisture

47 Exhaust pressure...lb. per sq. in., abs.

48 Exhaust quality...deg. F., or percent moisture

49 Steam flow to turbine throttle...lb. per hr.

50 Output in horsepower by guarantee or separate test...Hp.

51 For more detailed list see Test Code for Steam Turbines

b. Electric motor

52 Potential...volts

53 Current...amperes

54 Frequency...cycles per sec.

55 Power input...kw., phase 1, 2, 3

56 Speed...r.p.m.

57 Excitation...

58 Output in horsepower by guarantee or separate test...Hp.

59 For more detailed list see appropriate A.I.E.E. codes...

REPORT OF TESTS OF COMPRESSORS AND EXHAUSTERS (*Continued*)

Item

AVERAGE OF DATA AS RECORDED.—*Continued*

DRIVER.—*Continued*

c. *Internal-combustion engine, etc.*

60 Calorific value of fuel........B.t.u. per lb.
61 Fuel consumption........lb. per hr.
62 Exhaust temperature........deg. F.
63 Speed........r.p.m.
64 Output in horsepower by guarantee or separate test........
65 For more detailed list, see appropriate Internal Combustion Engine code........

PERFORMANCE UNDER ACTUAL TEST CONDITIONS

66 Test number........
67 Date........
68 Time........
69 Conditions peculiar to particular test........
70 Barometer........in. Hg
71 Total inlet absolute pressure........in. Hg, or lb. per sq. in.
72 Inlet temperature........deg. F.
73 Inlet density of gas........lb. per cu. ft.
74 Discharge absolute pressure........in. Hg, or lb. per sq. in.
75 Volume at inlet conditions........cu. ft. per min.
76 Shaft horsepower (by direct measurement, guarantee, or separate test data of driver) S.Hp.
77 Adiabatic (or isothermal) horsepower........Hp.
78 Adiabatic (or isothermal) efficiency........percent
79 Power consumption (converted to equivalent in net input in horsepower of driver)......
80 Overall adiabatic (or isothermal) efficiency........percent
81 Power consumption..........lb. of steam per hr.........kw. or lb. of fuel per hr.
82 Unit power consumption..........lb. of steam; ....kw.-hr.; ....lb. of fuel, or available B.t.u. per 100 cu. ft. of initial gas

PERFORMANCE INDICATED FOR SPECIFIED CONDITIONS

83 Speed........specified r.p.m.
84 Inlet density of gas........specified........lb. per cu. ft.
85 Inlet absolute pressure........specified........in. Hg, or lb. per sq. in.
86 Discharge absolute pressure........specified........in. Hg, or lb. per sq. in.
87 Volume (at inlet conditions) indicated by test for above specified conditions..cu.ft. per min.
88 Volume (at inlet conditions) guaranteed or predicted for above specified conditions........cu. ft. per min.
89 Power consumption........specified

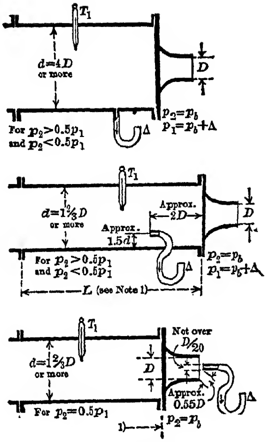
Nozzle Outlet Pressure, where it affects the flow, is the static pressure of an unrestricted region into which the discharge takes place. For example, when a nozzle under pressure discharges to atmosphere, the nozzle outlet pressure is that of the atmosphere as given by the barometer reading. When a nozzle discharges into a reservoir so large that the pressure on the walls is the same wherever measured, the nozzle outlet pressure is the static pressure in the reservoir.

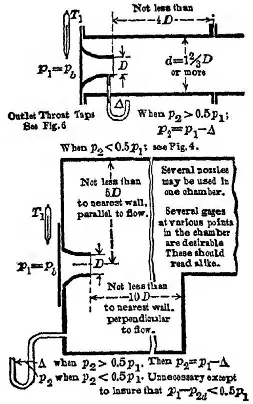
Nozzle Inlet Total Pressure in an inlet pipe whose diameter is $4D$, or more (D = nozzle diameter), may be taken as either the static or total pressure. Static or total pressure also may be used where the fluid enters nozzle from a large space, as when air flows from outside into an exhausted reservoir. If inlet pipe is less than $4D$, total pressure is higher than static pressure. It may be taken with an impact tube, pointing upstream, either in the pipe just ahead of the nozzle or in the jet outside. The reading will be the same in either position. See Fig. 2 for arrangement of apparatus.

Nozzle Inlet Static Pressure may be used instead of inlet total pressure, but the formulas are more complicated. See Fig. 3 for an arrangement of apparatus.

Affected and Unaffected Flow.—When p_2 is more than about 0.5 p_1, flow increases with decrease of p_2. The flow then is *affected.* When p_2 is less than about 0.5 p_1, a change of p_2 does not change the flow, which then is *unaffected.* In either affected or unaffected flow, the pressure rise of the machine under test may be wasted in order to make the flow measurement test, or the flow may be measured while the machine is in service. See Figs. 2 to 5 for arrangement of apparatus for the several classes.

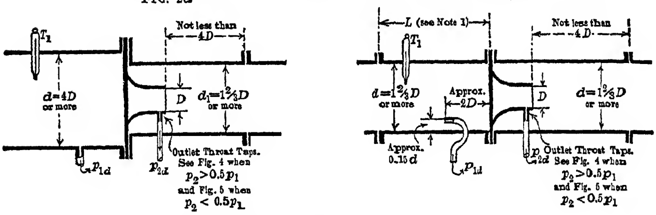
Pressures.—With affected flow, if discharge is into an unrestricted region, nozzle throat pressure is the same as outlet pressure, *i.e.*, pressure in unrestricted region. If nozzle outlet pressure is observed with throat taps, nozzle discharge coefficient is practically the same as for unrestricted discharge. Throat pressure observations should be made with at least two, and preferably four, sets of pressure holes, each with its own dif-





where pressure rise is wasted in order to make flow measurement test.

Fig. 2a Fig. 2b



Arrangements for measuring flow with machine in service.

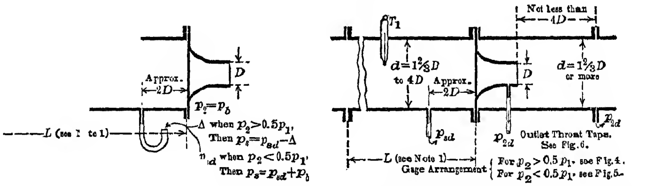
Fig. 2c

Note 1. $L = 10\,D$ where there is a preceding symmetrical irregularity in the pipe, as a reducer; $L = 20\,D$ if there is a preceding unsymmetrical irregularity, as an ell or tee.

Fig. 2. Flow Measurement with Inlet Total Pressure

ferential U-tube, whose readings should agree within 1.5%. Apparatus set-up and operating conditions should be refined until this agreement is attained.

Nozzle diameters should be selected that will give pressures ranging from 10 to 40 in. of water or 10 to 28 in. of mercury. High-pressure air discharging to atmosphere may be measured with high initial pressure, giving unaffected flow, or a throttle may be used to give affected flow, and nozzle diameters selected accordingly. Throttle valves should be of the butterfly or globe type (not a gate) and should be at least 20 nozzle diameters upstream from nozzle.



Note 1. $L = 10\,D$ where there is a preceding symmetrical irregularity in the pipe, as a reducer; $L = 20\,D$ where there is a preceding unsymmetrical irregularity, as an ell or tee.

Fig. 3. Flow Measurement with Inlet Static Pressure

TEMPERATURES less than 300° F. should be measured with a mercury-in-glass thermometer with bare bulb exposed to the fluid, and located 6 to 10 nozzle diameters upstream from the nozzle. T_1 = thermometer reading + 459.6.

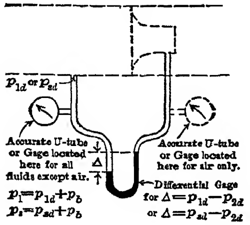


FIG. 4

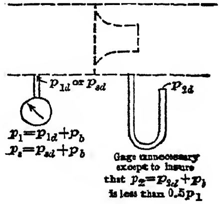


FIG. 5

Gage Arrangements for Flow Measurement with Machine in Service

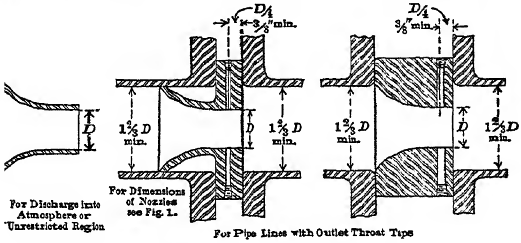


FIG. 6. Forms of Nozzles

ROTARY MOTION of the fluid should be eliminated by guide vanes or straighteners (Fig. 7) whose discharge ends are 5 to 15 nozzle diameters upstream from nozzle.

SELECTION OF FORMULA, when result desired is in volume instead of weight of fluid, should be on the basis of: 1. *Inlet volume* at conditions of pressure and temperature at inlet of machine under test, invariably used for tests of compressing and circulating apparatus; substitute values of inlet temperature and pressure for p_3 and T_3 in the formulas. 2. *Atmospheric conditions*, substitute values of p_b and T_b for p_3 and T_3. 3. *Volume under any conditions*, substitute values for such conditions for p_3 and T_3.

STANDARD CONDITIONS OF AIR are taken as 14.7 lb. per sq. in. pressure, 68° F. temperature, and 0.075 lb. per cu. ft. density. These conditions represent average air and make some allowance for humidity.

DENSITY OF PERFECT GASES.—Where the density in the inlet pipes is unknown, it may be calculated by one of the following formulas. These formulas also may be combined with the formulas in Table 1 to form new formulas in terms of factors such as specific gravity, etc., which are known or can be determined, instead of in terms of the inlet density ρ_1. This is done by substituting for ρ_1 in the formulas of Table 1, the values given by one of the formulas below.

Based on density ρ_0 at 68° F. and 14.7 lb. per sq. in.

$$\rho_1 = 527.6\, p_1\, \rho_0 / 14.7\, T_1.$$

Based on specific gravity G, compared to air without moisture

$$\rho_1 = \{(527.6 \times 0.752)/14.7\}\{p_1\, G/T\}$$

on specific heat C_p

$$\rho_1 = (144/J)(p_1/T_1)\,[1/\{C_p(k-1)/k\}]$$

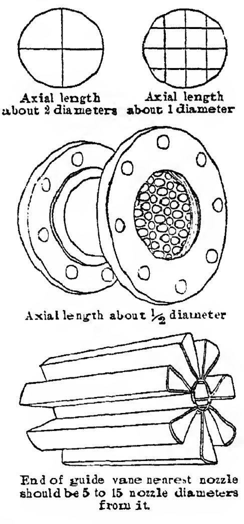


FIG. 7. Guide Vanes to Prevent Rotary Motion

Based on R, from equation, $144\ pv = RT$

$$\rho_1 =$$

Based on molecular weight M

$$\rho_1 = (527.6 \times 0.752\, p_1\, M)/(14.7 \times 28.989\, T_1).$$

For a mixture of gases, density at standard conditions ρ_0, or specific gravity, may be computed from analysis and the first or second formula used.

If the gas contains water vapor,

$$\rho_1 = \rho_g\, \{1 - (Hp_w/p_1)\} + H\rho_w,$$

where ρ_1, ρ_g = density, lb. per cu. ft., of moist and dry gas, respectively, at p_1 and T_1; ρ_w = density, lb. per cu. ft., of saturated water vapor at T_1; H = relative humidity of gas at nozzle inlet = 1.00 when gas is saturated at T_1; p, p_w = saturation pressure of water vapor at T and T_1 respectively; p_1 = absolute pressure at nozzle; T = dewpoint of water vapor at inlet, where gas is saturated at some temperature T less than T_1; T_1= temperature at nozzle inlet.

In some cases gas, saturated at a given temperature, may decrease in temperature to T_1 at nozzle inlet, with certainty of complete removal of the condensed moisture. Under some such circumstances, H may be inferred to be 1.00 at T_1. Pressures and temperatures may be in any units, if the same units are used throughout.

DATA AND RESULTS should be reported in the form shown on p. 16–64.

Section 17

MATHEMATICAL TABLES

TABLE OF CONTENTS

MATHEMATICAL TABLES

LOGARITHMS

Logarithms (abbreviation *log*).—The log of a number is the exponent of the power to which it is necessary to raise a fixed number to produce the given number. The fixed number is called the *base*. Thus if the base is 10, the log of 1000 is 3, for $10^3 = 1000$. There are two systems of logs in general use, the *common*, in which the base is 10, and the Napierian, or *hyperbolic*, in which the base is 2.718281828. . . . The Napierian base is commonly denoted by e, as in the equation $e^y = x$, in which y is the Napierian log of x. The abbreviation $\log_e$ is commonly used to denote the Napierian log.

In any system of logs, the log of 1 is 0; the log of the base, taken in that system, is 1. In any system the base of which is greater than 1, the logs of all numbers greater than 1 are positive and the logs of all numbers less than 1 are negative.

The modulus of any system is equal to the reciprocal of the Napierian log of the base of that system. The modulus of the Napierian system is 1, that of the common system is 0.4342945. The log of a number in any system equals the modulus of that system $\times$ Napierian log of the number. The *hyperbolic* or *Napierian* log of any number equals the common log $\times$ 2.3025851.

Every log consists of two parts, an integral part called the *characteristic*, or index, and the decimal part, or *mantissa*. The mantissa only is given in the usual tables of common logs, with the decimal point omitted. The characteristic is one less than the number of figures to the left of the decimal point in the number whose log is to be found. The characteristic of numbers from 1 to 9.99+ is 0, from 10 to 99.99+ is 1, from 100 to 999+ is 2, from 0.1 to 0.99+ is −1, from 0.01 to 0.099+ is −2, etc. Thus,

log of 2000 is 3.30103;	log of 0.2	is −1.30103,	or	9.30103 − 10
" " 200 " 2.30103;	" " 0.02	" −2.30103,	"	8.30103 − 10
" " 20 " 1.30103;	" " 0.002	" −3.30103,	"	7.30103 − 10
" " 2 " 0.30103;	" " 0.0002	" −4.30103,	"	6.30103 − 10

The minus sign is frequently written above the characteristic thus: log 0.002 = $\bar{3}.30103$. The characteristic only is negative, the decimal part, or mantissa, being always positive.

When a log consists of a negative index and a positive mantissa, it is usual to write the negative sign over the index, or to add 10 to the index, and indicate the subtraction of 10 from the resulting logarithm. Thus log 0.2 = $\bar{1}.30103$, may be written 9.30103−10. The difference between a logarithm and 10 is its *arithmetical complement* or **cologarithm**.

In tables of logarithmic sines, etc., the −10 is generally omitted, as being understood.

RULES FOR USE OF THE TABLE OF COMMON LOGARITHMS.—To Find the Log of a Decimal Fraction or of a Whole Number and a Decimal.—First find the log of the quantity as if there were no decimal point, then prefix the index according to rule; the index is one less than the number of figures to the left of the decimal point.

EXAMPLE. log of 3.14159.	log of 3.141 =	0.497068.	Diff.	138.
From proportional parts	5 =	690		
" " "	09 =	1242		
	log 3.14159	0.4971494		

If the number is a decimal less than unity, the index is negative and is one more than the number of zeros to the right of the decimal point. Log of 0.0682 = $\bar{2}.833784$ = 8.833784 − 10.

To Find the Number Corresponding to a Given Log.—Find in the table the log nearest to the decimal part of the given log and take the first four digits of the required number from the column N and the top of the column containing the log which is the next less than the given log. To find the 5th and 6th digits subtract the log in the table from the given log, and multiply the difference by 100, and divide by the figure in the Diff. column opposite the log; annex the quotient to

the four digits already found, and place the decimal point according to the rule; the number of figures to the left of the decimal point is one greater than the index. The number corresponding to a log is called the anti-logarithm.

Find the anti-log of.............................. 0.497150
Next lowest log in table corresponds to 3141........ 0.497068 Diff. = 82.

Tabular diff. = 138; 82 ÷ 138 = 0.59+

The index being 0, the number is therefore 3.14159+.

Multiplication by Means of Logarithms.—Add together the logs of the two numbers to be multiplied. The sum is the log of the product.

a. Where both factors are greater than unity.

EXAMPLE. $31 \times 1274 = 39{,}494$.

Solution. $\log 31 + \log 1274 = 1.491362 + 3.105169 = 4.596531 = \log 39{,}494$.

b. Where one or more factors are less than unity, the logs with a negative characteristic can be handled most conveniently by adding and subtracting 10.

EXAMPLE. $.000028961 \times .084507 = .0000024474$.

Solution. $\log .000028961 + \log .084507 = \bar{5}.461814 + \bar{2}.926893.$

$$= (5.461814 - 10) + (8.926893 - 10) = 14.388707 - 20 = \bar{6}.388707$$
$$= \log .0000024474.$$

Division by Means of Logarithms.—Subtract the log of the divisor from the log of the dividend. The remainder is the log of the quotient.

a. When the divisor is smaller than the dividend.

EXAMPLE. $2987 \div 63 = 47.41284$.

Solution. $\log 2987 - \log 63 = 3.475235 - 1.799341 = 1.675894 = \log 47.41284$.

b. When the divisor is larger than the dividend, add and subtract as many tens as may be necessary to the log of the dividend and proceed as before.

EXAMPLE. $.000672 \div 263 = .00000255513$.

Solution. $\log .000672 - \log 263 = \bar{4}.827369 - 2.419956$
$$= 6.827369 - 2.419956 - 10 = \bar{6}.407413 = \log .00000255513.$$

c. The log of a fraction is obtained by subtracting the log of the denominator from the log of the numerator. Thus,

$$\log \frac{a}{b} = \log a - \log b.$$

To Raise a Number to Any Given Power.—Multiply the log of the number by the exponent of the number, and find the number whose log is the product.

a. Where the exponent consists of one figure.

EXAMPLE. $16.2^3 = 4251.528$.

Solution. $3 \times \log 16.2 = 3 \times 1.209515 = 3.628545 = \log 4251.528$.

b. Where the exponent consists of two or more figures, it is best to multiply the characteristic and mantissa separately.

EXAMPLE. $.005624^{.37} = .147067$.

Solution. $.37 \times \log .005624 = .37 \times \bar{3}.750045 = .37 \times (-3) + .37 \times .750045$
$$= -1.11 + .277517 = \bar{1}.167517 = \log .147067.$$

c. Where the number is a fraction, first find the log of the fraction and then multiply it by the exponent.

EXAMPLE. $\left(\frac{276}{.032}\right)^{.72} = 681.9396$.

Solution. $.72 (\log 276 - \log .032) = .72 \{2.440909 - (8.505150 - 10)\}$
$$= .72 \times 3.935759 = 2.833746 = \log 681.9396.$$

To Extract Any Root of a Number.—Divide the log of the number by the index of the root, and find the number whose log is the quotient.

To extract the root of a decimal: *a.* When the root index is positive and evenly divisible into the negative characteristic of the log of the number, the division may be performed with the negative characteristic written in its usual place.

EXAMPLE. $\sqrt[4]{.0006954} = .16239$.

Solution. $\log .0006954 \div 4 = \bar{4}.842235 \div 4 = \bar{1}.210559 = \log .16239$.

b. When the root index is positive and not even divisible into the negative characteristic, add to the log of the number, and indicate the subtraction from it, the smallest integral multiple of the root which will eliminate the negative characteristic. Divide the result by the root index and ascertain the number whose log corresponds to the quotient.

EXAMPLE. $\sqrt[3.1]{.00002785} = .03393$.

Solution. $\log .00002785 = \bar{5}.444825;\ \bar{5}.444825 \div 3.1 = \{(2 \times 3.1) + \bar{5}.444825 - 2 \times 3.1\} \div 3.1$
$$= \{1.644825 - 6.2\} \div 3.1 = \bar{2}.530589 = \log .03393.$$

c. When the root index is negative, determine the excess of the negative characteristic over the positive mantissa. Divide the result by the root index and ascertain the number whose log corresponds to the quotient.

EXAMPLE. $\sqrt[-4]{.000003976} = 22.394$.

Solution. $\log .000003976 = \bar{6}.599446 = .599446 - 6 = -5.400554;$
$$-5.400554 \div (-4) = 1.350138 = \log 22.394.$$

Solution of Exponential Equations.—In an exponential equation, the unknown quantity is the exponent; thus $a^x = b$. This may be transformed to $\log a^x = \log b$, or $x \log a = \log b$, whence $x = \log b \div \log a$.

a. When the base is greater than unity, put the equation in the form $x = \log b \div \log a$. Then $\log x = \log (\log b) - \log (\log a)$.

Example. $32.6^x = 14.632$.

Solution. $\log x = \log (\log 14.632) - \log (\log 32.6) = \log 1.165303 - \log 1.513218$
$= .066439 - .179901 = \bar{1}.886538$; $x = .77008$.

b. When both the known quantities are decimal, put the equation in the form $x = \log b \div \log a$. Subtract the positive mantissa from the negative characteristic in both divisor and dividend, obtaining negative remainders. Change the signs of divisor and dividend and proceed as in Case *a*.

Example. $.0729^x = .2693$.

Solution. $x = \log .2693 \div \log .0729 = \bar{1}.430236 \div \bar{2}.862728$
$= (-.569764) \div (-1.137272) = .569764 \div 1.137272$;
$\log x = \log .569764 - \log 1.137272 = \bar{1}.755695 - .055864 = \bar{1}.699831$
$x = .50099$.

c. When only one of the known quantities is a decimal, put the equation in the form $x = \dfrac{\text{og}}{\log a}$.
Subtract the positive mantissa from the negative characteristic of the numerator or denominator as the case may be, and rewrite the fraction with the remainder so obtained as the new numerator or denominator. Make both numerator and denominator positive, but write a minus sign in front of the fraction, to signify that the result will be a negative quantity. Solve the fraction by logarithms and write a minus sign in front of the result.

Example. $.726^x = 802.7$.

Solution. $x = \dfrac{\log 802.7}{\log .726} = \dfrac{2.904553}{\bar{1}.860937} = \dfrac{2.904553}{-.139063} = -\dfrac{2.904553}{.139063}$.

$\log x = -(\log 2.904553 - \log .139063) = -(.463079 - \bar{1}.143211) = -(1.319868)$.
$x = -20.8867$.

d. When the exponent is negative and one of the known quantities is less than unity, put the equation in the form $(-x) = \dfrac{\log b}{\log a}$. Subtract the positive mantissa from the negative characteristic, as in Case *c*, and multiply both sides of the resulting equation by (-1). Find the value of x as in Case *c*.

Example. $10.78^{-x} = .09431$.

Solution. $x = \dfrac{\log .09431}{\log 10.78} = \dfrac{\bar{2}.974558}{1.032619} = -\dfrac{1.025442}{1.032619}$.

$\log x = .010911 - .013940 = \bar{1}.996971$. $x = .99305$.

Table 1. Logarithms of Numbers from 1 to 100

N	Log	N	Log	N	Log	N	Log	N	Log
1	0.000000	21	1.322219	41	1.612784	61	1.785330	81	1.908485
2	0.301030	22	1.342423	42	1.623249	62	1.792392	82	1.913814
3	0.477121	23	1.361728	43	1.633468	63	1.799341	83	1.919078
4	0.602060	24	1.380211	44	1.643453	64	1.806180	84	1.924279
5	0.698970	25	1.397940	45	1.653213	65	1.812913	85	1.929419
6	0.778151	26	1.414973	46	1.662758	66	1.819544	86	1.934498
7	0.845098	27	1.431364	47	1.672098	67	1.826075	87	1.939519
8	0.903090	28	1.447158	48	1.681241	68	1.832509	88	1.944483
9	0.954243	29	1.462398	49	1.690196	69	1.838849	89	1.949390
10	1.000000	30	1.477121	50	1.698970	70	1.845098	90	1.954243
11	1.041393	31	1.491362	51	1.707570	71	1.851258	91	1.959041
12	1.079181	32	1.505150	52	1.716003	72	1.857332	92	1.963788
13	1.113943	33	1.518514	53	1.724276	73	1.863323	93	1.968483
14	1.146128	34	1.531479	54	1.732394	74	1.869232	94	1.973128
15	1.176091	35	1.544068	55	1.740363	75	1.875061	95	1.977724
16	1.204120	36	1.556303	56	1.748188	76	1.880814	96	1.982271
17	1.230449	37	1.568202	57	1.755875	77	1.886491	97	1 986772
18	1.255273	38	1.579784	58	1.763428	78	1.892095	98	1.991226
19	1.278754	39	1.591065	59	1.770852	79	1.897627	99	1.995635
20	1.301030	40	1.602060	60	1.778151	80	1.903090	100	2.000000

See pp. 17–06 to 17–23 for a complete table of six-place logarithms.

Table 2—Common Logarithms of Numbers

N	0	1	2	3	4	5	6	7	8	9	Diff.
100	**000000**	**000434**	**000868**	**001301**	**001734**	**002166**	**002598**	**003029**	**003461**	**003891**	432
1	004321	004751	005181	005609	006038	006466	006894	007321	007748	008174	428
2	008600	009026	009451	009876	010300	010724	011147	011570	011993	012415	424
3	012837	013259	013680	014100	014521	014940	015360	015779	016197	016616	420
4	017033	017451	017868	018284	018700	019116	019532	019947	020361	020775	416
5	021189	021603	022016	022428	022841	023252	023664	024075	024486	024896	412
6	025306	025715	026125	026533	026942	027350	027757	028164	028571	028978	408
7	029384	029789	030195	030600	031004	031408	031812	032216	032619	033021	404
8	033424	033826	034227	034628	035029	035430	035830	036230	036629	037028	400
9	037426	037825	038223	038620	039017	039414	039811	040207	040602	040998	397
110	**041393**	**041787**	**042182**	**042576**	**042969**	**043362**	**043755**	**044148**	**044540**	**044932**	393
1	045323	045714	046105	046495	046885	047275	047664	048053	048442	048830	390
2	049218	049606	049993	050380	050766	051153	051538	051924	052309	052694	386
3	053078	053463	053846	054230	054613	054996	055378	055760	056142	056524	383
4	056905	057286	057666	058046	058426	058805	059185	059563	059942	060320	379
5	060698	061075	061452	061829	062206	062582	062958	063333	063709	064083	376
6	064458	064832	065206	065580	065953	066326	066699	067071	067443	067815	373
7	068186	068557	068928	069298	069668	070038	070407	070776	071145	071514	370
8	071882	072250	072617	072985	073352	073718	074085	074451	074816	075182	366
9	075547	075912	076276	076640	077004	077368	077731	078094	078457	078819	363
120	**079181**	**079543**	**079904**	**080266**	**080625**	**080987**	**081347**	**081707**	**082067**	**082426**	360
1	082785	083144	083503	083861	084219	084576	084934	085291	085647	086004	357
2	086360	086716	087071	087426	087781	088136	088490	088845	089198	089552	355
3	089905	090258	090611	090963	091315	091667	092018	092370	092721	093071	352
4	093422	093772	094122	094471	094820	095169	095518	095866	096215	096562	349
5	096910	097257	097604	097951	098298	098644	098990	099335	099681	100026	346
6	100371	100715	101059	101403	101747	102091	102434	102777	103119	103462	343
7	103804	104146	104487	104828	105169	105510	105851	106191	106531	106871	341
8	107210	107549	107888	108227	108565	108903	109241	109579	109916	110253	338
9	110590	110926	111263	111599	111934	112270	112605	112940	113275	113609	335

Proportional Parts

Diff.	1	2	3	4	5	6	7	8	9
434	43.4	86.8	130.2	173.6	217.0	260.4	303.8	347.2	390.6
432	43.2	86.4	129.6	172.8	216.0	259.2	302.4	345.6	388.8
430	43.0	86.0	129.0	172.0	215.0	258.0	301.0	344.0	387.0
428	42.8	85.6	128.4	171.2	214.0	256.8	299.6	342.4	385.2
426	42.6	85.2	127.8	170.4	213.0	255.6	298.2	340.8	383.4
424	42.4	84.8	127.2	169.6	212.0	254.4	296.8	339.2	381.6
422	42.2	84.4	126.6	168.8	211.0	253.2	295.4	337.6	379.8
420	42.0	84.0	126.0	168.0	210.0	252.0	294.0	336.0	378.0
418	41.8	83.6	125.4	167.2	209.0	250.8	292.6	334.4	376.2
416	41.6	83.2	124.8	166.4	208.0	249.6	291.2	332.8	374.4
414	41.4	82.8	124.2	165.6	207.0	248.4	289.8	331.2	372.6
412	41.2	82.4	123.6	164.8	206.0	247.2	288.4	329.6	370.8
410	41.0	82.0	123.0	164.0	205.0	246.0	287.0	328.0	369.0
408	40.8	81.6	122.4	163.2	204.0	244.8	285.6	326.4	367.2
406	40.6	81.2	121.8	162.4	203.0	243.6	284.2	324.8	365.4
404	40.4	80.8	121.2	161.6	202.0	242.4	282.8	323.2	363.6
402	40.2	80.4	120.6	160.8	201.0	241.2	281.4	321.6	361.8
400	40.0	80.0	120.0	160.0	200.0	240.0	280.0	320.0	360.0
398	39.8	79.6	119.4	159.2	199.0	238.8	278.6	318.4	358.2
396	39.6	79.2	118.8	158.4	198.0	237.6	277.2	316.8	356.4
394	39.4	78.8	118.2	157.6	197.0	236.4	275.8	315.2	354.6
392	39.2	78.4	117.6	156.8	196.0	235.2	274.4	313.6	352.8
390	39.0	78.0	117.0	156.0	195.0	234.0	273.0	312.0	351.0
388	38.8	77.6	116.4	155.2	194.0	232.8	271.6	310.4	349.2
386	38.6	77.2	115.8	154.4	193.0	231.6	270.2	308.8	347.4
384	38.4	76.8	115.2	153.6	192.0	230.4	268.8	307.2	345.6
382	38.2	76.4	114.6	152.8	191.0	229.2	267.4	305.6	343.8
380	38.0	76.0	114.0	152.0	190.0	228.0	266.0	304.0	342.0
378	37.8	75.6	113.4	151.2	189.0	226.8	264.6	302.4	340.2
376	37.6	75.2	112.8	150.4	188.0	225.6	263.2	300.8	338.4
374	37.4	74.8	112.2	149.6	187.0	224.4	261.8	299.2	336.6
372	37.2	74.4	111.6	148.8	186.0	223.2	260.4	297.6	334.8
370	37.0	74.0	111.0	148.0	185.0	222.0	259.0	296.0	333.0
368	36.8	73.6	110.4	147.2	184.0	220.8	257.6	294.4	331.2
366	36.6	73.2	109.8	146.4	183.0	219.6	256.2	292.8	329.4
364	36.4	72.8	109.2	145.6	182.0	218.4	254.8	291.2	327.6
362	36.2	72.4	108.6	144.8	181.0	217.2	253.4	289.6	325.8
360	36.0	72.0	108.0	144.0	180.0	216.0	252.0	288.0	324.0

N	0	1	2	3	4	5	6	7	8	9	Diff.
130	**113943**	**114277**	**114611**	**114944**	**115278**	**115611**	**115943**	**116276**	**116608**	**116940**	333
1	117271	117603	117934	118265	118595	118926	119256	119586	119915	120245	330
2	120574	120903	121231	121560	121888	122216	122544	122871	123198	123525	328
3	123852	124178	124504	124830	125156	125481	125806	126131	126456	126781	325
4	127105	127429	127753	128076	128399	128722	129045	129368	129690	130012	323
5	130334	130655	130977	131298	131619	131939	132260	132580	132900	133219	321
6	133539	133858	134177	134496	134814	135133	135451	135769	136086	136403	318
7	136721	137037	137354	137671	137987	138303	138618	138934	139249	139564	316
8	139879	140194	140508	140822	141136	141450	141763	142076	142389	142702	314
9	143015	143327	143639	143951	144263	144574	144885	145196	145507	145818	311
140	**146128**	**146438**	**146748**	**147058**	**147367**	**147676**	**147985**	**148294**	**148603**	**148911**	309
1	149219	149527	149835	150142	150449	150756	151063	151370	151676	151982	307
2	152288	152594	152900	153205	153510	153815	154120	154424	154728	155032	305
3	155336	155640	155943	156246	156549	156852	157154	157457	157759	158061	303
4	158362	158664	158965	159266	159567	159868	160168	160469	160769	161068	301
5	161368	161667	161967	162266	162564	162863	163161	163460	163758	164055	299
6	164353	164650	164947	165244	165541	165838	166134	166430	166726	167022	297
7	167317	167613	167908	168203	168497	168792	169086	169380	169674	169968	295
8	170262	170555	170848	171141	171434	171726	172019	172311	172603	172895	293
9	173186	173478	173769	174060	174351	174641	174932	175222	175512	175802	291
150	**176091**	**176381**	**176670**	**176959**	**177248**	**177536**	**177825**	**178113**	**178401**	**178689**	289
1	178977	179264	179552	179839	180126	180413	180699	180986	181272	181558	287
2	181844	182129	182415	182700	182985	183270	183555	183839	184123	184407	285
3	184691	184975	185259	185542	185825	186108	186391	186674	186956	187239	283
4	187521	187803	188084	188366	188647	188928	189209	189490	189771	190051	281
5	190332	190612	190892	191171	191451	191730	192010	192289	192567	192846	279
6	193125	193403	193681	193959	194237	194514	194792	195069	195346	195623	278
7	195900	196176	196453	196729	197005	197281	197556	197832	198107	198382	276
8	198657	198932	199206	199481	199755	200029	200303	200577	200850	201124	274
9	201397	201670	201943	202216	202488	202761	203033	203305	203577	203848	272

PROPORTIONAL PARTS

Diff.	1	2	3	4	5	6	7	8	9
358	35.8	71.6	107.4	143.2	179.0	214.8	250.6	286.4	322.2
356	35.6	71.2	106.8	142.4	178.0	213.6	249.2	284.8	320.4
354	35.4	70.8	106.2	141.6	177.0	212.4	247.8	283.2	318.6
352	35.2	70.4	105.6	140.8	176.0	211.2	246.4	281.6	316.8
350	35.0	70.0	105.0	140.0	175.0	210.0	245.0	280.0	315.0
348	34.8	69.6	104.4	139.2	174.0	208.8	243.6	278.4	313.2
346	34.6	69.2	103.8	138.4	173.0	207.6	242.2	276.8	311.4
344	34.4	68.8	103.2	137.6	172.0	206.4	240.8	275.2	309.6
342	34.2	68.4	102.6	136.8	171.0	205.2	239.4	273.6	307.8
340	34.0	68.0	102.0	136.0	170.0	204.0	238.0	272.0	306.0
338	33.8	67.6	101.4	135.2	169.0	202.8	236.6	270.4	304.2
336	33.6	67.2	100.8	134.4	168.0	201.6	235.2	268.8	302.4
334	33.4	66.8	100.2	133.6	167.0	200.4	233.8	267.2	300.6
332	33.2	66.4	99.6	132.8	166.0	199.2	232.4	265.6	298.8
330	33.0	66.0	99.0	132.0	165.0	198.0	231.0	264.0	297.0
328	32.8	65.6	98.4	131.2	164.0	196.8	229.6	262.4	295.2
326	32.6	65.2	97.8	130.4	163.0	195.6	228.2	260.8	293.4
324	32.4	64.8	97.2	129.6	162.0	194.4	226.8	259.2	291.6
322	32.2	64.4	96.6	128.8	161.0	193.2	225.4	257.6	289.8
320	32.0	64.0	96.0	128.0	160.0	192.0	224.0	256.0	288.0
318	31.8	63.6	95.4	127.2	159.0	190.8	222.6	254.4	286.2
316	31.6	63.2	94.8	126.4	158.0	189.6	221.2	252.8	284.4
314	31.4	62.8	94.2	125.6	157.0	188.4	219.8	251.2	282.6
312	31.2	62.4	93.6	124.8	156.0	187.2	218.4	249.6	280.8
310	31.0	62.0	93.0	124.0	155.0	186.0	217.0	248.0	279.0
308	30.8	61.6	92.4	123.2	154.0	184.8	215.6	246.4	277.2
306	30.6	61.2	91.8	122.4	153.0	183.6	214.2	244.8	275.4
304	30.4	60.8	91.2	121.6	152.0	182.4	212.8	243.2	273.6
302	30.2	60.4	90.6	120.8	151.0	181.2	211.4	241.6	271.8
300	30.0	60.0	90.0	120.0	150.0	180.0	210.0	240.0	270.0
298	29.8	59.6	89.4	119.2	149.0	178.8	208.6	238.4	268.2
296	29.6	59.2	88.8	118.4	148.0	177.6	207.2	236.8	266.4
294	29.4	58.8	88.2	117.6	147.0	176.4	205.8	235.2	264.6
292	29.2	58.4	87.6	116.8	146.0	175.2	204.4	233.6	262.8
290	29.0	58.0	87.0	116.0	145.0	174.0	203.0	232.0	261.0
288	28.8	57.6	86.4	115.2	144.0	172.8	201.6	230.4	259.2
286	28.6	57.2	85.8	114.4	143.0	171.6	200.2	228.8	257.4
284	28.4	56.8	85.2	113.6	142.0	170.4	198.8	227.2	255.6
282	28.2	56.4	84.6	112.8	141.0	169.2	197.4	225.6	253.8
280	28.0	56.0	84.0	112.0	140.0	168.0	196.0	224.0	252.0

N											
160	**204120**	**204391**	**204663**	**204934**	**05204**	**05475**	**205746**	**06016**	**206286**	**06556**	271
1	206826	207096	207365	07634	07904	08173	08441	08710	08979	09247	269
2	209515	209783	210051	10319	10586	10853	11121	11388	1654	1921	267
3	212188	212454	212720	12986	13252	13518	13783	14049	4314	4579	266
4	214844	215109	215373	15638	15902	16166	16430	16694	16957	7221	264
5	217484	217747	218010	18273	18536	18798	19060	19323	19585	19846	262
6	220108	220370	220631	20892	21153	21414	21675	21936	22196	22456	261
7	222716	222976	223236	23496	23755	224015	24274	224533	24792	25051	259
8	225309	225568	225826	26084	26342	226600	26858	27115	27372	27630	258
9	227887	228144	228400	28657	28913	229170	29426	29682	29938	30193	256
170		**230704**	**230960**	**231215**	**231470**	**231724**	**231979**	**232234**	**232488**	**232742**	255
1	232996	233250	233504	33757	34011	34264	34517	34770	35023	235276	253
2	235528	235781	236033	36285	36537	36789	37041	37292	37544	237795	252
3	238046	238297	238548	38799	39049	39299	39550	39800	40050	40300	250
4	240549	240799	241048	241297	41546	41795	42044	42293	42541	42790	249
5	243038	243286	243534	243782	244030	44277	44525	44772	45019	45266	248
6	245513	245759	246006	246252	246499	46745	46991	47237	47482	47728	246
7	247973	248219	248464	248709	248954	49198	249443	49687	49932	50176	245
8	250420	250664	25090	251151	251395	251638	251881	52125	52368	252610	243
9	252853	253096	25333	253580	253822	254064	254306	54548	54790	55031	242
180	**255273**	**255514**	**255755**	**255996**	**256237**	**256477**	**256718**	**256958**	**57198**	**257439**	241
1	257679	257918	25815	25839	25863	258877	259116	259355	25959	259833	239
2	260071	260310	26054	26078	26102	261263	261501	261739	261976	26221	238
3	262451	262688	262925	26316	26339	263636	263873	264109	264346	264582	237
4	264818	265054	26529	26552	26576	265996	266232	266467	266702	266937	235
5	267172	267406	26764	26787	26811	26834	26857	268812	269046	269279	234
6	269513	269746	26998	27021	27044	27067	27091	271144	27137	271609	233
7	271842	272074	27230	27253	27277	27300	27323	27346	2736	27392	232
8	274158	274389	27462	27485	27508	27531	27554	27577	27600	27623	230
9	276462	276692	27692	27715	27738	277609	27783	27806	27829	27852	229
190	**278754**	**278982**	**27921**	**27943**	**279667**	**279895**	**28012**	**28035**	**28057**	**280806**	228
1	281033	281261	2814	28171	28194	28216	28239	28262	28284	28307	227
2	283301	283527	2837	28397	28420	28443	28465	28488	28510	28533	226
3	285557	285782	2860	28623	28645	28668	28690	28713	287354	28757	225
4	287802	288026	2882	2884	2886	2889	28914	28936	28958	28981	223
5	290035	290257	2904	2907	29092	2911	29136	29159	29181	29203	222
6	292256	292478	2926	29292	29314	29336	29358	29380	2940	29424	221
7	294466	294687	2949	2951	29534	29556	29578	29600	296226	29644	220
8	296665	296884	2971	2973	29754	29776	29797	29819	2984	29863	219
				29951	299		30016	30037	3005	30081	218

Proportional Parts

Diff.							7		
278	27.8	55.6	83.4	111.2	139.0	166.8	194.6	222.4	250.2
276	27.6	55.2	82.8	110.4	138.0	165.6	193.2	220.8	248.4
274	27.4	54.8	82.2	109.6	137.0	164.4	191.8	219.2	246.6
272	27.2	54.4	81.6	108.8	136.0	163.2	190.4	217.6	244.8
270	27.0	54.0	81.0	108.0	135.0	162.0	189.0	216.0	243.0
268	26.8	53.6	80.4	107.2	134.0	160.8	187.6	214.4	
266	26.6	53.2	79.8	106.4	133.0	159.6	186.2	212.8	
264	26.4	52.8	79.2	105.6	132.0	158.4	184.8	211.2	
262	26.2	52.4	78.6	104.8	131.0	157.2	183.4	209.6	
260	26.0	52.0	78.0	104.0	130.0	156.0	182.0	208.0	
258	25.8	51.6	77.4	103.2	129.0	154.8	180.6	206.4	
256	25.6	51.2	76.8	102.4	128.0	153.6	179.2	204.8	
254	25.4	50.8	76.2	101.6	127.0	152.4	177.8	203.2	
252	25.2	50.4	75.6	100.8	126.0	151.2	176.4	201.6	
250	25.0	50.0	75.0	100.0	125.0	150.0	175.0	200.0	
248	24.8	49.6	74.4	99.2	124.0	148.8	173.6	198.4	
246	24.6	49.2	73.8	98.4	123.0	147.6	172.2	196.8	
244	24.4	48.8	73.2	97.6	122.0	146.4	170.8	195.2	
242	24.2	48.4	72.6	96.8	121.0	145.2	169.4	193.6	
240	24.0	48.0	72.0	96.0	120.0	144.0	168.0	192.0	
238	23.8	47.6	71.4	95.2	119.0	142.8	166.6	190.4	
236	23.6	47.2	70.8	94.4	118.0	141.6	165.2	188.8	
234	23.4	46.8	70.2	93.6	117.0	140.4	163.8	187.2	
232	23.2	46.4	69.6	92.8	116.0	139.2	162.4	185.6	
			69.0	92.0	115.0	138.0	161.0	184.0	

N	0	1	2	3	4	5	6	7	8	9	Diff.
200	**301030**	**301247**	**301464**	**301681**	**301898**	**302114**	**302331**	**302547**	**302764**	**302980**	217
1	303196	303412	303628	303844	304059	304275	304491	304706	304921	305136	216
2	305351	305566	305781	305996	306211	306425	306639	306854	307068	307282	215
3	307496	307710	307924	308137	308351	308564	308778	308991	309204	309417	213
4	309630	309843	310056	310268	310481	310693	310906	311118	311330	311542	212
5	311754	311966	312177	312389	312600	312812	313023	313234	313445	313656	211
6	313867	314078	314289	314499	314710	314920	315130	315340	315551	315760	210
7	315970	316180	316390	316599	316809	317018	317227	317436	317646	317854	209
8	318063	318272	318481	318689	318898	319106	319314	319522	319730	319938	208
9	320146	320354	320562	320769	320977	321184	321391	321598	321805	322012	207
210	**322219**	**322426**	**322633**	**322839**	**323046**	**323252**	**323458**	**323665**	**323871**	**324077**	206
1	324282	324488	324694	324899	325105	325310	325516	325721	325926	326131	205
2	326336	326541	326745	326950	327155	327359	327563	327767	327972	328176	204
3	328380	328583	328787	328991	329194	329398	329601	329805	330008	330211	203
4	330414	330617	330819	331022	331225	331427	331630	331832	332034	332236	202
5	332438	332640	332842	333044	333246	333447	333649	333850	334051	334253	
6	334454	334655	334856	335057	335257	335458	335658	335859	336059	336260	201
7	336460	336660	336860	337060	337260	337459	337659	337858	338058	338257	200
8	338456	338656	338855	339054	339253	339451	339650	339849	340047	340246	199
9	340444	340642	340841	341039	341237	341435	341632	341830	342028	342225	198
220	**342423**	**342620**	**342817**	**343014**	**343212**	**343409**	**343606**	**343802**	**343999**	**344196**	197
1	344392	344589	344785	344981	345178	345374	345570	345766	345962	346157	196
2	346353	346549	346744	346939	347135	347330	347525	347720	347915	348110	195
3	348305	348500	348694	348889	349083	349278	349472	349666	349860	350054	194
4	350248	350442	350636	350829	351023	351216	351410	351603	351796	351989	193
5	352183	352375	352568	352761	352954	353147	353339	353532	353724	353916	
6	354108	354301	354493	354685	354876	355068	355260	355452	355643	355834	192
7	356026	356217	356408	356599	356790	356981	357172	357363	357554	357744	191
8	357935	358125	358316	358506	358696	358886	359076	359266	359456	359646	190
9	359835	360025	360215	360404	360593	360783	360972	361161	361350	361539	189
230	**361728**	**361917**	**362105**	**362294**	**362482**	**362671**	**362859**	**363048**	**363236**	**363424**	188
1	363612	363800	363988	364176	364363	364551	364739	364926	365113	365301	
2	365488	365675	365862	366049	366236	366423	3666[illegible]0	36[illegible]96	366983	367169	187
3	367356	367542	367729	367915	368101	368287	368[illegible]3	368659	368845	369030	186
4	369216	369401	369587	369772	369958	370143	370328	370513	370698	370883	185
5	371068	371253	371437	371622	371806	371991	372175	372360	372544	372728	184
6	372912	373096	373280	373464	373647	373831	374015	374198	374382	374565	
7	374748	374932	375115	375298	375481	375664	375846	376029	376212	376394	183
8	376577	376759	376942	377124	377306	377488	377670	377852	378034	378216	182
9	378398	378580	378761	378943	379124	379306	379487	379668	379849	380030	181

PROPORTIONAL PARTS

Diff.	1	2	3	4	5	6	7	8	9
228	22.8	45.6	68.4	91.2	114.0	136.8	159.6	182.4	205.2
226	22.6	45.2	67.8	90.4	113.0	135.6	158.2	180.8	203.4
224	22.4	44.8	67.2	89.6	112.0	134.4	156.8	179.2	201.6
222	22.2	44.4	66.6	88.8	111.0	133.2	155.4	177.6	199.8
220	22.0	44.0	66.0	88.0	110.0	132.0	154.0	176.0	198.0
218	21.8	43.6	65.4	87.2	109.0	130.8	152.6	174.4	196.2
216	21.6	43.2	64.8	86.4	108.0	129.6	151.2	172.8	194.4
214	21.4	42.8	64.2	85.6	107.0	128.4	149.8	171.2	192.6
212	21.2	42.4	63.6	84.8	106.0	127.2	148.4	169.6	190.8
210	21.0	42.0	63.0	84.0	105.0	126.0	147.0	168.0	189.0
208	20.8	41.6	62.4	83.2	104.0	124.8	145.6	166.4	187.2
206	20.6	41.2	61.8	82.4	103.0	123.6	144.2	164.8	185.4
204	20.4	40.8	61.2	81.6	102.0	122.4	142.8	163.2	183.6
202	20.2	40.4	60.6	80.8	101.0	121.2	141.4	161.6	181.8
200	20.0	40.0	60.0	80.0	100.0	120.0	140.0	160.0	180.0
198	19.8	39.6	59.4	79.2	99.0	118.8	138.6	158.4	178.2
196	19.6	39.2	58.8	78.4	98.0	117.6	137.2	156.8	176.4
194	19.4	38.8	58.2	77.6	97.0	116.4	135.8	155.2	174.6
192	19.2	38.4	57.6	76.8	96.0	115.2	134.4	153.6	172.8
190	19.0	38.0	57.0	76.0	95.0	114.0	133.0	152.0	171.0
188	18.8	37.6	56.4	75.2	94.0	112.8	131.6	150.4	169.2
186	18.6	37.2	55.8	74.4	93.0	111.6	130.2	148.8	167.4
184	18.4	36.8	55.2	73.6	92.0	110.4	128.8	147.2	165.6
182	18.2	36.4	54.6	72.8	91.0	109.2	127.4	145.6	163.8
180	18.0	36.0	54.0	72.0	90.0	108.0	126.0	144.0	162.0

N	0	1	2	3	4	5	6	7	8	9	Diff.
240	**380211**	**380392**	**380573**	**380754**	**380934**	**381115**	**381296**	**381476**	**381656**	**381837**	181
1	382017	382197	382377	382557	382737	382917	383097	383277	383456	383636	180
2	383815	383995	384174	384353	384533	384712	384891	385070	385249	385428	179
3	385606	385785	385964	386142	386321	386499	386677	386856	387034	387212	178
4	387390	387568	387746	387924	388101	388279	388456	388634	388811	388989	
5	389166	389343	389520	389698	389875	390051	390228	390405	390582	390759	177
6	390935	391112	391288	391464	391641	391817	391993	392169	392345	392521	176
7	392697	392873	393048	393224	393400	393575	393751	393926	394101	394277	
8	394452	394627	394802	394977	395152	395326	395501	395676	395850	396025	175
9	396199	396374	396548	396722	396896	397071	397245	397419	397592	397766	174
250	**397940**	**398114**	**398287**	**398461**	**398634**	**398808**	**398981**	**399154**	**399328**	**399501**	173
1	399674	399847	400020	400192	400365	400538	400711	400883	401056	401228	
2	401401	401573	401745	401917	402089	402261	402433	402605	402777	402949	172
3	403121	403292	403464	403635	403807	403978	404149	404320	404492	404663	171
4	404834	405005	405176	405346	405517	405688	405858	406029	406199	406370	
5	406540	406710	406881	407051	407221	407391	407561	407731	407901	408070	170
6	408240	408410	408579	408749	408918	409087	409257	409426	409595	409764	169
7	409933	410102	410271	410440	410609	410777	410946	411114	411283	411451	
8	411620	411788	411956	412124	412293	412461	412629	412796	412964	413132	168
9	413300	413467	413635	413803	413970	414137	414305	414472	414639	414806	167
260	**414973**	**415140**	**415307**	**415474**	**415641**	**415808**	**415974**	**416141**	**416308**	**416474**	
1	416641	416807	416973	417139	417306	417472	417638	417804	417970	418135	166
2	418301	418467	418633	418798	418964	419129	419295	419460	419625	419791	165
3	419956	420121	420286	420451	420616	420781	420945	421110	421275	421439	
4	421604	421768	421933	422097	422261	422426	422590	422754	422918	423082	164
5	423246	423410	423574	423737	423901	424065	424228	424392	424555	424718	
6	424882	425045	425208	425371	425534	425697	425860	426023	426186	426349	163
7	426511	426674	426836	426999	427161	427324	427486	427648	427811	427973	162
8	428135	428297	428459	428621	428783	428944	429106	429268	429429	429591	
9	429752	429914	430075	430236	430398	430559	430720	430881	431042	431203	161
270	**431364**	**431525**	**431685**	**431846**	**432007**	**432167**	**432328**	**432488**	**432649**	**432809**	
1	432969	433130	433290	433450	433610	433770	433930	434090	434249	434409	160
2	434569	434729	434888	435048	435207	435367	435526	435685	435844	436004	159
3	436163	436322	436481	436640	436799	436957	437116	437275	437433	437592	
4	437751	437909	438067	438226	438384	438542	438701	438859	439017	439175	158
5	439333	439491	439648	439806	439964	440122	440279	440437	440594	440752	
6	440909	441066	441224	441381	441538	441695	441852	442009	442166	442323	157
7	442480	442637	442793	442950	443106	443263	443419	443576	443732	443889	
8	444045	444201	444357	444513	444669	444825	444981	445137	445293	445449	156
9	445604	445760	445915	446071	446226	446382	446537	446692	446848	447003	155
280	**447158**	**447313**	**447468**	**447623**	**447778**	**447933**	**448088**	**448242**	**448397**	**448552**	
1	448706	448861	449015	449170	449324	449478	449633	449787	449941	450095	154
2	450249	450403	450557	450711	450865	451018	451172	451326	451479	451633	
3	451786	451940	452093	452247	452400	452553	452706	452859	453012	453165	153
4	453318	453471	453624	453777	453930	454082	454235	454387	454540	454692	
5	454845	454997	455150	455302	455454	455606	455758	455910	456062	456214	152
6	456366	456518	456670	456821	456973	457125	457276	457428	457579	457731	
7	457882	458033	458184	458336	458487	458638	458789	458940	459091	459242	151
8	459392	459543	459694	459845	459995	460146	460296	460447	460597	460748	
9	460898	461048	461198	461348	461499	461649	461799	461948	462098	462248	150

PROPORTIONAL PARTS

Diff.	1	2	3	4	5	6	7	8	9
182	18.2	36.4	54.6	72.8	91.0	109.2	127.4	145.6	163.8
180	18.0	36.0	54.0	72.0	90.0	108.0	126.0	144.0	162.0
178	17.8	35.6	53.4	71.2	89.0	106.8	124.6	142.4	160.2
176	17.6	35.2	52.8	70.4	88.0	105.6	123.2	140.8	158.4
174	17.4	34.8	52.2	69.6	87.0	104.4	121.8	139.2	156.6
172	17.2	34.4	51.6	68.8	86.0	103.2	120.4	137.6	154.8
170	17.0	34.0	51.0	68.0	85.0	102.0	119.0	136.0	153.0
168	16.8	33.6	50.4	67.2	84.0	100.8	117.6	134.4	151.2
166	16.6	33.2	49.8	66.4	83.0	99.6	116.2	132.8	149.4
164	16.4	32.8	49.2	65.6	82.0	98.4	114.8	131.2	147.6
162	16.2	32.4	48.6	64.8	81.0	97.2	113.4	129.6	145.8
160	16.0	32.0	48.0	64.0	80.0	96.0	112.0	128.0	144.0
158	15.8	31.6	47.4	63.2	79.0	94.8	110.6	126.4	142.2
156	15.6	31.2	46.8	62.4	78.0	93.6	109.2	124.8	140.4

N	0	1	2	3	4	5	6	7	8	9	Diff.
290	**462398**	**462548**	**462697**	**462847**	**462997**	**463146**	**463296**	**463445**	**463594**	**463744**	
1	463893	464042	464191	464340	464490	464639	464788	464936	465085	465234	149
2	465383	465532	465680	465829	465977	466126	466274	466423	466571	466719	
3	466868	467016	467164	467312	467460	467608	467756	467904	468052	468200	148
4	468347	468495	468643	468790	468938	469085	469233	469380	469527	469675	
5	469822	469969	470116	470263	470410	470557	470704	470851	470998	471145	147
6	471292	471438	471585	471732	471878	472025	472171	472318	472464	472610	146
7	472756	472903	473049	473195	473341	473487	473633	473779	473925	474071	
8	474216	474362	474508	474653	474799	474944	475090	475235	475381	475526	
9	475671	475816	475962	476107	476252	476397	476542	476687	476832	476976	145
300	**477121**	**477266**	**477411**	**477555**	**477700**	**477844**	**477989**	**478133**	**478278**	**478422**	
1	478566	478711	478855	478999	479143	479287	479431	479575	479719	479863	144
2	480007	480151	480294	480438	480582	480725	480869	481012	481156	481299	
3	481443	481586	481729	481872	482016	482159	482302	482445	482588	482731	143
4	482874	483016	483159	483302	483445	483587	483730	483872	484015	484157	
5	484300	484442	484585	484727	484869	485011	485153	485295	485437	485579	142
6	485721	485863	486005	486147	486289	486430	486572	486714	486855	486997	
7	487138	487280	487421	487563	487704	487845	487986	488127	488269	488410	141
8	488551	488692	488833	488974	489114	489255	489396	489537	489677	489818	
9	489958	490099	490239	490380	490520	490661	490801	490941	491081	491222	140
310	**491362**	**491502**	**491642**	**491782**	**491922**	**492062**	**492201**	**492341**	**492481**	**492621**	
1	492760	492900	493040	493179	493319	493458	493597	493737	493876	494015	139
2	494155	494294	494433	494572	494711	494850	494989	495128	495267	495406	
3	495544	495683	495822	495960	496099	496238	496376	496515	496653	496791	
4	496930	497068	497206	497344	497483	497621	497759	497897	498035	498173	138
5	498311	498448	498586	498724	498862	498999	499137	499275	499412	499550	
6	499687	499824	499962	500099	500236	500374	500511	500648	500785	500922	137
7	501059	501196	501333	501470	501607	501744	501880	502017	502154	502291	
8	502427	502564	502700	502837	502973	503109	503246	503382	503518	503655	136
9	503791	503927	504063	504199	504335	504471	504607	504743	504878	505014	
320	**505150**	**505286**	**505421**	**505557**	**505693**	**505828**	**505964**	**506099**	**506234**	**506370**	
1	506505	506640	506776	506911	507046	507181	507316	507451	507586	507721	135
2	507856	507991	508126	508260	508395	508530	508664	508799	508934	509068	
3	509203	509337	509471	509606	509740	509874	510009	510143	510277	510411	134
4	510545	510679	510813	510947	511081	511215	511349	511482	511616	511750	
5	511883	512017	512151	512284	512418	512551	512684	512818	512951	513084	133
6	513218	513351	513484	513617	513750	513883	514016	514149	514282	514415	
7	514548	514681	514813	514946	515079	515211	515344	515476	515609	515741	
8	515874	516006	516139	516271	516403	516535	516668	516800	516932	517064	132
9	517196	517328	517460	517592	517724	517855	517987	518119	518251	518382	
330	**518514**	**518646**	**518777**	**518909**	**519040**	**519171**	**519303**	**519434**	**519566**	**519697**	131
1	519828	519959	520090	520221	520353	520484	520615	520745	520876	521007	
2	521138	521269	521400	521530	521661	521792	521922	522053	522183	522314	
3	522444	522575	522705	522835	522966	523096	523226	523356	523486	523616	130
4	523746	523876	524006	524136	524266	524396	524526	524656	524785	524915	
5	525045	525174	525304	525434	525563	525693	525822	525951	526081	526210	129
6	526339	526469	526598	526727	526856	526985	527114	527243	527372	527501	
7	527630	527759	527888	528016	528145	528274	528402	528531	528660	528788	
8	528917	529045	529174	529302	529430	529559	529687	529815	529943	530072	128
9	530200	530328	530456	530584	530712	530840	530968	531096	531223	531351	

PARTS

Diff.	1	2	3	4	5	6	7	8	9
154	15.4	30.8	46.2	61.6	77.0	92.4	107.8	123.2	138.6
152	15.2	30.4	45.6	60.8	76.0	91.2	106.4	121.6	136.8
150	15.0	30.0	45.0	60.0	75.0	90.0	105.0	120.0	135.0
148	14.8	29.6	44.4	59.2	74.0	88.8	103.6	118.4	133.2
146	14.6	29.2	43.8	58.4	73.0	87.6	102.2	116.8	131.4
144	14.4	28.8	43.2	57.6	72.0	86.4	100.8	115.2	129.6
142	14.2	28.4	42.6	56.8	71.0	85.2	99.4	113.6	127.8
140	14.0	28.0	42.0	56.0	70.0	84.0	98.0	112.0	126.0
138	13.8	27.6	41.4	55.2	69.0	82.8	96.6	110.4	124.2
136	13.6	27.2	40.8	54.4	68.0	81.6	95.2	108.8	122.4
134	13.4	26.8	40.2	53.6	67.0	80.4	93.8	107.2	120.6
132	13.2	26.4	39.6	52.8	66.0	79.2	92.4	105.6	118.8
130	13.0	26.0	39.0	52.0	65.0	78.0	91.0	104.0	117.0
128	12.8	25.6	38.4	51.2	64.0	76.8	89.6	102.4	115.2

N	0	1	2	3	4	5	6	7	8	9	Diff.
340	**531479**	**531607**	**531734**	**531862**	**531990**	**532117**	**532245**	**532372**	**532500**	**532627**	
1	532754	532882	533009	533136	533264	533391	533518	533645	533772	533899	127
2	534026	534153	534280	534407	534534	534661	534787	534914	535041	535167	
3	535294	535421	535547	535674	535800	535927	536053	536180	536306	536432	126
4	536558	536685	536811	536937	537063	537189	537315	537441	537567	537693	
5	537819	537945	538071	538197	538322	538448	538574	538699	538825	538951	
6	539076	539202	539327	539452	539578	539703	539829	539954	540079	540204	125
7	540329	540455	540580	540705	540830	540955	541080	541205	541330	541454	
8	541579	541704	541829	541953	542078	542203	542327	542452	542576	542701	
9	542825	542950	543074	543199	543323	543447	543571	543696	543820	543944	124
350	**544068**	**544192**	**544316**	**544440**	**544564**	**544688**	**544812**	**544936**	**545060**	**545183**	
1	545307	545431	545555	545678	545802	545925	546049	546172	546296	546419	
2	546543	546666	546789	546913	547036	547159	547282	547405	547529	547652	123
3	547775	547898	548021	548144	548267	548389	548512	548635	548758	548881	
4	549003	549126	549249	549371	549494	549616	549739	549861	549984	550106	
5	550228	550351	550473	550595	550717	550840	550962	551084	551206	551328	122
6	551450	551572	551694	551816	551938	552060	552181	552303	552425	552547	
7	552668	552790	552911	553033	553155	553276	553398	553519	553640	553762	121
8	553883	554004	554126	554247	554368	554489	554610	554731	554852	554973	
9	555094	555215	555336	555457	555578	555699	555820	555940	556061	556182	
360	**556303**	**556423**	**556544**	**556664**	**556785**	**556905**	**557026**	**557146**	**557267**	**557387**	120
1	557507	557627	557748	557868	557988	558108	558228	558349	558469	558589	
2	558709	558829	558948	559068	559188	559308	559428	559548	559667	559787	
3	559907	560026	560146	560265	560385	560504	560624	560743	560863	560982	119
4	561101	561221	561340	561459	561578	561698	561817	561936	562055	562174	
5	562293	562412	562531	562650	562769	562887	563006	563125	563244	563362	
6	563481	563600	563718	563837	563955	564074	564192	564311	564429	564548	
7	564666	564784	564903	565021	565139	565257	565376	565494	565612	565730	118
8	565848	565965	566034	566202	566320	566437	566555	566673	566791	566909	
9	567026	567144	567262	567379	567497	567614	567732	567849	567967	568084	
370	**568202**	**568319**	**568436**	**568554**	**568671**	**568788**	**568905**	**569023**	**569140**	**569257**	117
1	569374	569491	569608	569725	569842	569959	570076	570193	570309	570426	
2	570543	570660	570776	570893	571010	571126	571243	571359	571476	571592	
3	571709	571825	571942	572058	572174	572291	572407	572523	572639	572755	116
4	572872	572988	573104	573220	573336	573452	573568	573684	573800	573915	
5	574031	574147	574263	574379	574494	574610	574726	574841	574957	575072	
6	575188	575303	575419	575534	575650	575765	575880	575996	576111	576226	115
7	576341	576457	576572	576687	576802	576917	577032	577147	577262	577377	
8	577492	577607	577722	577836	577951	578066	578181	578295	578410	578525	
9	578639	578754	578868	578983	579097	579212	579326	579441	579555	579669	114
380	**579784**	**579898**	**580012**	**580126**	**580241**	**580355**	**580469**	**580583**	**580697**	**580811**	
1	580925	581039	581153	581267	581381	581495	581608	581722	581836	581950	
2	582063	582177	582291	582404	582518	582631	582745	582858	582972	583085	
3	583199	583312	583426	583539	583652	583765	583879	583992	584105	584218	
4	584331	584444	584557	584670	584783	584896	585009	585122	585235	585348	113
5	585461	585574	585686	585799	585912	586024	586137	586250	586362	586475	
6	586587	586700	586812	586925	587037	587149	587262	587374	587486	587599	
7	587711	587823	587935	588047	588160	588272	588384	588496	588608	588720	112
8	588832	588944	589056	589167	589279	589391	589503	589615	589726	589838	
9	589950	590061	590173	590284	590396	590507	590619	590730	590842	590953	

PROPORTIONAL PARTS

Diff.	1	2	3	4	5	6	7	8	9
128	12.8	25.6	38.4	51.2	64.0	76.8	89.6	102.4	115.2
126	12.6	25.2	37.8	50.4	63.0	75.6	88.2	100.8	113.4
124	12.4	24.8	37.2	49.6	62.0	74.4	86.8	99.2	111.6
122	12.2	24.4	36.6	48.8	61.0	73.2	85.4	97.6	109.8
120	12.0	24.0	36.0	48.0	60.0	72.0	84.0	96.0	108.0
118	11.8	23.6	35.4	47.2	59.0	70.8	82.6	94.4	106.2
116	11.6	23.2	34.8	46.4	58.0	69.6	81.2	92.8	104.4
114	11.4	22.8	34.2	45.6	57.0	68.4	79.8	91.2	102.6

N	0	1	2	3	4	5	6	7	8	9	Diff.
390	**591065**	**591176**	**591287**	**591399**	**591510**	**591621**	**591732**	**591843**	**591955**	**592066**	
1	592177	592288	592399	592510	592621	592732	592843	592954	593064	593175	111
2	593286	593397	593508	593618	593729	593840	593950	594061	594171	594282	
3	594393	594503	594614	594724	594834	594945	595055	595165	595276	595386	
4	595496	595606	595717	595827	595937	596047	596157	596267	596377	596487	
5	596597	596707	596817	596927	597037	597146	597256	597366	597476	597586	110
6	597695	597805	597914	598024	598134	598243	598353	598462	598572	598681	
7	598791	598900	599009	599119	599228	599337	599446	599556	599665	599774	
8	599883	599992	600101	600210	600319	600428	600537	600646	600755	600864	109
9	600973	601082	601191	601299	601408	601517	601625	601734	601843	601951	
400	**602060**	**602169**	**602277**	**602386**	**602494**	**602603**	**602711**	**602819**	**602928**	**603036**	
1	603144	603253	603361	603469	603577	603686	603794	603902	604010	604118	108
2	604226	604334	604442	604550	604658	604766	604874	604982	605089	605197	
3	605305	605413	605521	605628	605736	605844	605951	606059	606166	606274	
4	606381	606489	606596	606704	606811	606919	607026	607133	607241	607348	
5	607455	607562	607669	607777	607884	607991	608098	608205	608312	608419	107
6	608526	608633	608740	608847	608954	609061	609167	609274	609381	609488	
7	609594	609701	609808	609914	610021	610128	610234	610341	610447	610554	
8	610660	610767	610873	610979	611086	611192	611298	611405	611511	611617	
9	611723	611829	611936	612042	612148	612254	612360	612466	612572	612678	106
410	**612784**	**612890**	**612996**	**613102**	**613207**	**613313**	**613419**	**613525**	**613630**	**613736**	
1	613842	613947	614053	614159	614264	614370	614475	614581	614686	614792	
2	614897	615003	615108	615213	615319	615424	615529	615634	615740	615845	
3	615950	616055	616160	616265	616370	616476	616581	616686	616790	616895	105
4	617000	617105	617210	617315	617420	617525	617629	617734	617839	617943	
5	618048	618153	618257	618362	618466	618571	618676	618780	618884	618989	
6	619093	619198	619302	619406	619511	619615	619719	619824	619928	620032	
7	620136	620240	620344	620448	620552	620656	620760	620864	620968	621072	104
8	621176	621280	621384	621488	621592	621695	621799	621903	622007	622110	
9	622214	622318	622421	622525	622628	622732	622835	622939	623042	623146	
420	**623249**	**623353**	**623456**	**623559**	**623663**	**623766**	**623869**	**623973**	**624076**	**624179**	
1	624282	624385	624488	624591	624695	624798	624901	625004	625107	625210	103
2	625312	625415	625518	625621	625724	625827	625929	626032	626135	626238	
3	626340	626443	626546	626648	626751	626853	626956	627058	627161	627263	
4	627366	627468	627571	627673	627775	627878	627980	628082	628185	628287	
5	628389	628491	628593	628695	628797	628900	629002	629104	629206	629308	102
6	629410	629512	629613	629715	629817	629919	630021	630123	630224	630326	
7	630428	630530	630631	630733	630835	630936	631038	631139	631241	631342	
8	631444	631545	631647	631748	631849	631951	632052	632153	632255	632356	101
9	632457	632559	632660	632761	632862	632963	633064	633165	633266	633367	
430	**633468**	**633569**	**633670**	**633771**	**633872**	**633973**	**634074**	**634175**	**634276**	**634376**	
1	634477	634578	634679	634779	634880	634981	635081	635182	635283	635383	
2	635484	635584	635685	635785	635886	635986	636087	636187	636287	636388	
3	636488	636588	636688	636789	636889	636989	637089	637189	637290	637390	
4	637490	637590	637690	637790	637890	637990	638090	638190	638290	638389	100
5	638489	638589	638689	638789	638888	638988	639088	639188	639287	639387	
6	639486	639586	639686	639785	639885	639984	640084	640183	640283	640382	
7	640481	640581	640680	640779	640879	640978	641077	641177	641276	641375	
8	641474	641573	641672	641771	641871	641970	642069	642168	642267	642366	
9	642465	642563	642662	642761	642860	642959	643058	643156	643255	643354	99

PROPORTIONAL

Diff.	1	2	3	4	5	6	7	8	9
112	11.2	22.4	33.6	44.8	56.0	67.2	78.4	89.6	100.8
110	11.0	22.0	33.0	44.0	55.0	66.0	77.0	88.0	99.0
108	10.8	21.6	32.4	43.2	54.0	64.8	75.6	86.4	97.2
106	10.6	21.2	31.8	42.4	53.0	63.6	74.2	84.8	95.4
104	10.4	20.8	31.2	41.6	52.0	62.4	72.8	83.2	93.6
102	10.2	20.4	30.6	40.8	51.0	61.2	71.4	81.6	91.8
100	10.0	20.0	30.0	40.0	50.0	60.0	70.0	80.0	90.0
98	9.8	19.6	29.4	39.2	49.0	58.8	68.6	78.4	88.2

N	0	1	2	3	4	5	6	7	8	9	Diff.
440	**643453**	**643551**	**643650**	**643749**	**643847**	**643946**	**644044**	**644143**	**644242**	**644340**	
1	644439	644537	644636	644734	644832	644931	645029	645127	645226	645324	
2	645422	645521	645619	645717	645815	645913	646011	646110	646208	646306	
3	646404	646502	646600	646698	646796	646894	646992	647089	647187	647285	98
4	647383	647481	647579	647676	647774	647872	647969	648067	648165	648262	
5	648360	648458	648555	648653	648750	648848	648945	649043	649140	649237	
6	649335	649432	649530	649627	649724	649821	649919	650016	650113	650210	
7	650308	650405	650502	650599	650696	650793	650890	650987	651084	651181	
8	651278	651375	651472	651569	651666	651762	651859	651956	652053	652150	97
9	652246	652343	652440	652536	652633	652730	652826	652923	653019	653116	
450	**653213**	**653309**	**653405**	**653502**	**653598**	**653695**	**653791**	**653888**	**653984**	**654080**	
1	654177	654273	654369	654465	654562	654658	654754	654850	654946	655042	
2	655138	655235	655331	655427	655523	655619	655715	655810	655906	656002	96
3	656098	656194	656290	656386	656482	656577	656673	656769	656864	656960	
4	657056	657152	657247	657343	657438	657534	657629	657725	657820	657916	
5	658011	658107	658202	658298	658393	658488	658584	658679	658774	658870	
6	658965	659060	659155	659250	659346	659441	659536	659631	659726	659821	
7	659916	660011	660106	660201	660296	660391	660486	660581	660676	660771	95
8	660865	660960	661055	661150	661245	661339	661434	661529	661623	661718	
9	661813	661907	662002	662096	662191	662286	662380	662475	662569	662663	
460	**662758**	**662852**	**662947**	**663041**	**663135**	**663230**	**663324**	**663418**	**663512**	**663607**	
1	663701	663795	663889	663983	664078	664172	664266	664360	664454	664548	
2	664642	664736	664830	664924	665018	665112	665206	665299	665393	665487	94
3	665581	665675	665769	665862	665956	666050	666143	666237	666331	666424	
4	666518	666612	666705	666799	666892	666986	667079	667173	667266	667360	
5	667453	667546	667640	667733	667826	667920	668013	668106	668199	668293	
6	668386	668479	668572	668665	668759	668852	668945	669038	669131	669224	
7	669317	669410	669503	669596	669689	669782	669875	669967	670060	670153	93
8	670246	670339	670431	670524	670617	670710	670802	670895	670988	671080	
9	671173	671265	671358	671451	671543	671636	671728	671821	671913	672005	
470	**672098**	**672190**	**672283**	**672375**	**672467**	**672560**	**672652**	**672744**	**672836**	**672929**	
1	673021	673113	673205	673297	673390	673482	673574	673666	673758	673850	
2	673942	674034	674126	674218	674310	674402	674494	674586	674677	674769	92
3	674861	674953	675045	675137	675228	675320	675412	675503	675595	675687	
4	675778	675870	675962	676053	676145	676236	676328	676419	676511	676602	
5	676694	676785	676876	676968	677059	677151	677242	677333	677424	677516	
6	677607	677698	677789	677881	677972	678063	678154	678245	678336	678427	
7	678518	678609	678700	678791	678882	678973	679064	679155	679246	679337	91
8	679428	679519	679610	679700	679791	679882	679973	680063	680154	680245	
9	680336	680426	680517	680607	680698	680789	680879	680970	681060	681151	
480	**681241**	**681332**	**681422**	**681513**	**681603**	**681693**	**681784**	**681874**	**681964**	**682055**	
1	682145	682235	682326	682416	682506	682596	682686	682777	682867	682957	
2	683047	683137	683227	683317	683407	683497	683587	683677	683767	683857	90
3	683947	684037	684127	684217	684307	684396	684486	684576	684666	684756	
4	684845	684935	685025	685114	685204	685294	685383	685473	685563	685652	
5	685742	685831	685921	686010	686100	686189	686279	686368	686458	686547	
6	686636	686726	686815	686904	686994	687083	687172	687261	687351	687440	
7	687529	687618	687707	687796	687886	687975	688064	688153	688242	688331	
8	688420	688509	688598	688687	688776	688865	688953	689042	689131	689220	89
9	689309	689398	689486	689575	689664	689753	689841	689930	690019	690107	
490	**690196**	**690285**	**690373**	**690462**	**690550**	**690639**	**690728**	**690816**	**690905**	**690993**	
1	691081	691170	691258	691347	691435	691524	691612	691700	691789	691877	
2	691965	692053	692142	692230	692318	692406	692494	692583	692671	692759	
3	692847	692935	693023	693111	693199	693287	693375	693463	693551	693639	88
4	693727	693815	693903	693991	694078	694166	694254	694342	694430	694517	
5	694605	694693	694781	694868	694956	695044	695131	695219	695307	695394	
6	695482	695569	695657	695744	695832	695919	696007	696094	696182	696269	
7	696356	696444	696531	696618	696706	696793	696880	696968	697055	697142	
8	697229	697317	697404	697491	697578	697665	697752	697839	697926	698014	87
[illegible]	[illegible]	[illegible]	[illegible]	[illegible]	[illegible]	[illegible]	69862[illegible]	698709	698796	698883	

PROPORTIONAL PARTS

Diff.	1	2	3	4	5	6	7	8	9
98	9.8	19.6	29.4	39.2	49.0	58.8	68.6	78.4	88.2
96	9.6	19.2	28.8	38.4	48.0	57.6	67.2	76.8	86.4
94	9.4	18.8	28.2	37.6	47.0	56.4	65.8	75.2	84.6
92	9.2	18.4	27.6	36.8	46.0	55.2	64.4	73.6	82.8
90	9.0	18.0	27.0	36.0	45.0	54.0	63.0	72.0	81.0
88	8.8	17.6	26.4	35.2	44.0	52.8	61.6	70.4	79.2

N	0	1	2	3	4	5	6	7	8	9	Diff.
500	**698970**	**699057**	**699144**	**699231**	**699317**	**699404**	**699491**	**699578**	**699664**	**699751**	
1	699838	699924	700011	700098	700184	700271	700358	700444	700531	700617	
2	700704	700790	700877	700963	701050	701136	701222	701309	701395	701482	
3	701568	701654	701741	701827	701913	701999	702086	702172	702258	702344	
4	702431	702517	702603	702689	702775	702861	702947	703033	703119	703205	
5	703291	703377	703463	703549	703635	703721	703807	703893	703979	704065	86
6	704151	704236	704322	704408	704494	704579	704665	704751	704837	704922	
7	705008	705094	705179	705265	705350	705436	705522	705607	705693	705778	
8	705864	705949	706035	706120	706206	706291	706376	706462	706547	706632	
9	706718	706803	706888	706974	707059	707144	707229	707315	707400	707485	
510	**707570**	**707655**	**707740**	**707826**	**707911**	**707996**	**708081**	**708166**	**708251**	**708336**	
1	708421	708506	708591	708676	708761	708846	708931	709015	709100	709185	85
2	709270	709355	709440	709524	709609	709694	709779	709863	709948	710033	
3	710117	710202	710287	710371	710456	710540	710625	710710	710794	710879	
4	710963	711048	711132	711217	711301	711385	711470	711554	711639	711723	
5	711807	711892	711976	712060	712144	712229	712313	712397	712481	712566	
6	712650	712734	712818	712902	712986	713070	713154	713238	713323	713407	
7	713491	713575	713659	713742	713826	713910	713994	714078	714162	714246	84
8	714330	714414	714497	714581	714665	714749	714833	714916	715000	715084	
9	715167	715251	715335	715418	715502	715586	715669	715753	715836	715920	
520	**716003**	**716087**	**716170**	**716254**	**716337**	**716421**	**716504**	**716588**	**716671**	**716754**	
1	716838	716921	717004	717088	717171	717254	717338	717421	717504	717587	
2	717671	717754	717837	717920	718003	718086	718169	718253	718336	718419	83
3	718502	718585	718668	718751	718834	718917	719000	719083	719165	719248	
4	719331	719414	719497	719580	719663	719745	719828	719911	719994	720077	
5	720159	720242	720325	720407	720490	720573	720655	720738	720821	720903	
6	720986	721068	721151	721233	721316	721398	721481	721563	721646	721728	
7	721811	721893	721975	722058	722140	722222	722305	722387	722469	722552	
8	722634	722716	722798	722881	722963	723045	723127	723209	723291	723374	
9	723456	723538	723620	723702	723784	723866	723948	724030	724112	724194	82
530	**724276**	**724358**	**724440**	**724522**	**724604**	**724685**	**724767**	**724849**	**724931**	**725013**	
1	725095	725176	725258	725340	725422	725503	725585	725667	725748	725830	
2	725912	725993	726075	726156	726238	726320	726401	726483	726564	726646	
3	726727	726809	726890	726972	727053	727134	727216	727297	727379	727460	
4	727541	727623	727704	727785	727866	727948	728029	728110	728191	728273	
5	728354	728435	728516	728597	728678	728759	728841	728922	729003	729084	
6	729165	729246	729327	729408	729489	729570	729651	729732	729813	729893	81
7	729974	730055	730136	730217	730298	730378	730459	730540	730621	730702	
8	730782	730863	730944	731024	731105	731186	731266	731347	731428	731508	
9	731589	731669	731750	731830	731911	731991	732072	732152	732233	732313	
540	**732394**	**732474**	**732555**	**732635**	**732715**	**732796**	**732876**	**732956**	**733037**	**733117**	
1	733197	733278	733358	733438	733518	733598	733679	733759	733839	733919	
2	733999	734079	734160	734240	734320	734400	734480	734560	734640	734720	80
3	734800	734880	734960	735040	735120	735200	735279	735359	735439	735519	
4	735599	735679	735759	735838	735918	735998	736078	736157	736237	736317	
5	736397	736476	736556	736635	736715	736795	736874	736954	737034	737113	
6	737193	737272	737352	737431	737511	737590	737670	737749	737829	737908	
7	737987	738067	738146	738225	738305	738384	738463	738543	738622	738701	
8	738781	738860	738939	739018	739097	739177	739256	739335	739414	739493	
9	739572	739651	739731	739810	739889	739968	740047	740126	740205	740284	79
550	**740363**	**740442**	**740521**	**740600**	**740678**	**740757**	**740836**	**740915**	**740994**	**741073**	
1	741152	741230	741309	741388	741467	741546	741624	741703	741782	741860	
2	741939	742018	742096	742175	742254	742332	742411	742489	742568	742647	
3	742725	742804	742882	742961	743039	743118	743196	743275	743353	743431	
4	743510	743588	743667	743745	743823	743902	743980	744058	744136	744215	
5	744293	744371	744449	744528	744606	744684	744762	744840	744919	744997	
6	745075	745153	745231	745309	745387	745465	745543	745621	745699	745777	78
7	745855	745933	746011	746089	746167	746245	746323	746401	746479	746556	
8	746634	746712	746790	746868	746945	747023	747101	747179	747256	747334	
9	747412	747489	747567	747645	747722	747800	747878	747955	748033	748110	

Proportional Parts

Diff.	1	2	3	4	5	6	7	8	9
86	8.6	17.2	25.8	34.4	43.0	51.6	60.2	68.8	77.4
84	8.4	16.8	25.2	33.6	42.0	50.4	58.8	67.2	75.6
82	8.2	16.4	24.6	32.8	41.0	49.2	57.4	65.6	73.8
80	8.0	16.0	24.0	32.0	40.0	48.0	56.0	64.0	72.0
78	7.8	15.6	23.4	31.2	39.0	46.8	54.6	62.4	70.2

N	0	1	2	3	4	5	6	7	8	9	Diff.
560	**748188**	**748266**	**748343**	**748421**	**748498**	**748576**	**748653**	**748731**	**748808**	**748885**	
1	748963	749040	749118	749195	749272	749350	749427	749504	749582	749659	
2	749736	749814	749891	749968	750045	750123	750200	750277	750354	750431	
3	750508	750586	750663	750740	750817	750894	750971	751048	751125	751202	
4	751279	751356	751433	751510	751587	751664	751741	751818	751895	751972	77
5	752048	752125	752202	752279	752356	752433	752509	752586	752663	752740	
6	752816	752893	752970	753047	753123	753200	753277	753353	753430	753506	
7	753583	753660	753736	753813	753889	753966	754042	754119	754195	754272	
8	754348	754425	754501	754578	754654	754730	754807	754883	754960	755036	
9	755112	755189	755265	755341	755417	755494	755570	755646	755722	755799	
570	**755875**	**755951**	**756027**	**756103**	**756180**	**756256**	**756332**	**756408**	**756484**	**756560**	
1	756636	756712	756788	756864	756940	757016	757092	757168	757244	757320	76
2	757396	757472	757548	757624	757700	757775	757851	757927	758003	758079	
3	758155	758230	758306	758382	758458	758533	758609	758685	758761	758836	
4	758912	758988	759063	759139	759214	759290	759366	759441	759517	759592	
5	759668	759743	759819	759894	759970	760045	760121	760196	760272	760347	
6	760422	760498	760573	760649	760724	760799	760875	760950	761025	761101	
7	761176	761251	761326	761402	761477	761552	761627	761702	761778	761853	
8	761928	762003	762078	762153	762228	762303	762378	762453	762529	762604	75
9	762679	762754	762829	762904	762978	763053	763128	763203	763278	763353	
580	**763428**	**763503**	**763578**	**763653**	**763727**	**763802**	**763877**	**763952**	**764027**	**764101**	
1	764176	764251	764326	764400	764475	764550	764624	764699	764774	764848	
2	764923	764998	765072	765147	765221	765296	765370	765445	765520	765594	
3	765669	765743	765818	765892	765966	766041	766115	766190	766264	766338	
4	766413	766487	766562	766636	766710	766785	766859	766933	767007	767082	
5	767156	767230	767304	767379	767453	767527	767601	767675	767749	767823	
6	767898	767972	768046	768120	768194	768268	768342	768416	768490	768564	74
7	768638	768712	768786	768860	768934	769008	769082	769156	769230	769303	
8	769377	769451	769525	769599	769673	769746	769820	769894	769968	770042	
9	770115	770189	770263	770336	770410	770484	770557	770631	770705	770778	
590	**770852**	**770926**	**770999**	**771073**	**771146**	**771220**	**771293**	**771367**	**771440**	**771514**	
1	771587	771661	771734	771808	771881	771955	772028	772102	772175	772248	
2	772322	772395	772468	772542	772615	772688	772762	772835	772908	772981	
3	773055	773128	773201	773274	773348	773421	773494	773567	773640	773713	
4	773786	773860	773933	774006	774079	774152	774225	774298	774371	774444	73
5	774517	774590	774663	774736	774809	774882	774955	775028	775100	775173	
6	775246	775319	775392	775465	775538	775610	775683	775756	775829	775902	
7	775974	776047	776120	776193	776265	776338	776411	776483	776556	776629	
8	776701	776774	776846	776919	776992	777064	777137	777209	777282	777354	
9	777427	777499	777572	777644	777717	777789	777862	777934	778006	778079	
600	**778151**	**778224**	**778296**	**778368**	**778441**	**778513**	**778585**	**778658**	**778730**	**778802**	
1	778874	778947	779019	779091	779163	779236	779308	779380	779452	779524	
2	779596	779669	779741	779813	779885	779957	780029	780101	780173	780245	
3	780317	780389	780461	780533	780605	780677	780749	780821	780893	780965	72
4	781037	781109	781181	781253	781324	781396	781468	781540	781612	781684	
5	781755	781827	781899	781971	782042	782114	782186	782258	782329	782401	
6	782473	782544	782616	782688	782759	782831	782902	782974	783046	783117	
7	783189	783260	783332	783403	783475	783546	783618	783689	783761	783832	
8	783904	783975	784046	784118	784189	784261	784332	784403	784475	784546	
9	784617	784689	784760	784831	784902	784974	785045	785116	785187	785259	
610	**785330**	**785401**	**785472**	**785543**	**785615**	**785686**	**785757**	**785828**	**785899**	**785970**	
1	786041	786112	786183	786254	786325	786396	786467	786538	786609	786680	71
2	786751	786822	786893	786964	787035	787106	787177	787248	787319	787390	
3	787460	787531	787602	787673	787744	787815	787885	787956	788027	788098	
4	788168	788239	788310	788381	788451	788522	788593	788663	788734	788804	
5	788875	788946	789016	789087	789157	789228	789299	789369	789440	789510	
6	789581	789651	789722	789792	789863	789933	790004	790074	790144	790215	
7	790285	790356	790426	790496	790567	790637	790707	790778	790848	790918	
8	790988	791059	791129	791199	791269	791340	791410	791480	791550	791620	
9	791691	791761	791831	791901	791971	792041	792111	792181	792252	792322	

Proportional Parts

Diff.	1	2	3	4	5	6	7	8	9
78	7.8	15.6	23.4	31.2	39.0	46.8	54.6	62.4	70.2
76	7.6	15.2	22.8	30.4	38.0	45.6	53.2	60.8	68.4
74	7.4	14.8	22.2	29.6	37.0	44.4	51.8	59.2	66.6
72	7.2	14.4	21.6	28.8	36.0	43.2	50.4	57.6	64.8
70	7.0	14.0	21.0	28.0	35.0	42.0	49.0	56.0	63.0

N	0	1	2	3	4	5	6	7	8	9	Diff.
620	**792392**	**792462**	**792532**	**792602**	**792672**	**792742**	**792812**	**792882**	**792952**	**793022**	70
1	793092	793162	793231	793301	793371	793441	793511	793581	793651	793721	
2	793790	793860	793930	794000	794070	794139	794209	794279	794349	794418	
3	794488	794558	794627	794697	794767	794836	794906	794976	795045	795115	
4	795185	795254	795324	795393	795463	795532	795602	795672	795741	795811	
5	795880	795949	796019	796088	796158	796227	796297	796366	796436	796505	
6	796574	796644	796713	796782	796852	796921	796990	797060	797129	797198	
7	797268	797337	797406	797475	797545	797614	797683	797752	797821	797890	
8	797960	798029	798098	798167	798236	798305	798374	798443	798513	798582	
9	798651	798720	798789	798858	798927	798996	799065	799134	799203	799272	69
630	**799341**	**799409**	**799478**	**799547**	**799616**	**799685**	**799754**	**799823**	**799892**	**799961**	
1	800029	800098	800167	800236	800305	800373	800442	800511	800580	800648	
2	800717	800786	800854	800923	800992	801061	801129	801198	801266	801335	
3	801404	801472	801541	801609	801678	801747	801815	801884	801952	802021	
4	802089	802158	802226	802295	802363	802432	802500	802568	802637	802705	
5	802774	802842	802910	802979	803047	803116	803184	803252	803321	803389	
6	803457	803525	803594	803662	803730	803798	803867	803935	804003	804071	
7	804139	804208	804276	804344	804412	804480	804548	804616	804685	804753	
8	804821	804889	804957	805025	805093	805161	805229	805297	805365	805433	68
9	805501	805569	805637	805705	805773	805841	805908	805976	806044	806112	
640	**806180**	**806248**	**806316**	**806384**	**806451**	**806519**	**806587**	**806655**	**806723**	**806790**	
1	806858	806926	806994	807061	807129	807197	807264	807332	807400	807467	
2	807535	807603	807670	807738	807806	807873	807941	808008	808076	808143	
3	808211	808279	808346	808414	808481	808549	808616	808684	808751	808818	
4	808886	808953	809021	809088	809156	809223	809290	809358	809425	809492	
5	809560	809627	809694	809762	809829	809896	809964	810031	810098	810165	
6	810233	810300	810367	810434	810501	810569	810636	810703	810770	810837	
7	810904	810971	811039	811106	811173	811240	811307	811374	811441	811508	67
8	811575	811642	811709	811776	811843	811910	811977	812044	812111	812178	
9	812245	812312	812379	812445	812512	812579	812646	812713	812780	812847	
650	**812913**	**812980**	**813047**	**813114**	**813181**	**813247**	**813314**	**813381**	**813448**	**813514**	
1	813581	813648	813714	813781	813848	813914	813981	814048	814114	814181	
2	814248	814314	814381	814447	814514	814581	814647	814714	814780	814847	
3	814913	814980	815046	815113	815179	815246	815312	815378	815445	815511	
4	815578	815644	815711	815777	815843	815910	815976	816042	816109	816175	
5	816241	816308	816374	816440	816506	816573	816639	816705	816771	816838	
6	816904	816970	817036	817102	817169	817235	817301	817367	817433	817499	
7	817565	817631	817698	817764	817830	817896	817962	818028	818094	818160	
8	818226	818292	818358	818424	818490	818556	818622	818688	818754	818820	66
9	818885	818951	819017	819083	819149	819215	819281	819346	819412	819478	
660	**819544**	**819610**	**819676**	**819741**	**819807**	**819873**	**819939**	**820004**	**820070**	**820136**	
1	820201	820267	820333	820399	820464	820530	820595	820661	820727	820792	
2	820858	820924	820989	821055	821120	821186	821251	821317	821382	821448	
3	821514	821579	821645	821710	821775	821841	821906	821972	822037	822103	
4	822168	822233	822299	822364	822430	822495	822560	822626	822691	822756	
5	822822	822887	822952	823018	823083	823148	823213	823279	823344	823409	
6	823474	823539	823605	823670	823735	823800	823865	823930	823996	824061	
7	824126	824191	824256	824321	824386	824451	824516	824581	824646	824711	
8	824776	824841	824906	824971	825036	825101	825166	825231	825296	825361	65
9	825426	825491	825556	825621	825686	825751	825815	825880	825945	826010	
670	**826075**	**826140**	**826204**	**826269**	**826334**	**826399**	**826464**	**826528**	**826593**	**826658**	
1	826723	826787	826852	826917	826981	827046	827111	827175	827240	827305	
2	827369	827434	827499	827563	827628	827692	827757	827821	827886	827951	
3	828015	828080	828144	828209	828273	828338	828402	828467	828531	828595	
4	828660	828724	828789	828853	828918	828982	829046	829111	829175	829239	
5	829304	829368	829432	829497	829561	829625	829690	829754	829818	829882	
6	829947	830011	830075	830139	830204	830268	830332	830396	830460	830525	
7	830589	830653	830717	830781	830845	830909	830973	831037	831102	831166	
8	831230	831294	831358	831422	831486	831550	831614	831678	831742	831806	64
9	831870	831934	831998	832062	832126	832189	832253	832317	832381	832445	

PROPORTIONAL

Diff.	1	2	3	4	5	6	7	8	9
70	7.0	14.0	21.0	28.0	35.0	42.0	49.0	56.0	63.0
68	6.8	13.6	20.4	27.2	34.0	40.8	47.6	54.4	61.2
66	6.6	13.2	19.8	26.4	33.0	39.6	46.2	52.8	59.4
64	6.4	12.8	19.2	25.6	32.0	38.4	44.8	51.2	57.6
62	6.2	12.4	18.6	24.8	31.0	37.2	43.4	49.6	55.8

N				3					8		
680	**832509**	**832573**	**832637**	**832700**	**832764**	**832828**	**832892**	**832956**	**833020**	**833083**	
1	833147	833211	833275	833338	833402	833466	833530	833593	833657	833721	
2	833784	833848	833912	833975	834039	834103	834166	834230	834294	834357	
3	834421	834484	834548	834611	834675	834739	834802	834866	834929	834993	
4	835056	835120	835183	835247	835310	835373	835437	835500	835564	835627	
5	835691	835754	835817	835881	835944	836007	836071	836134	836197	836261	
6	836324	836387	836451	836514	836577	836641	836704	836767	836830	836894	
7	836957	837020	837083	837146	837210	837273	837336	837399	837462	837525	
8	837588	837652	837715	837778	837841	837904	837967	838030	838093	838156	63
9	838219	838282	838345	838408	838471	838534	838597	838660	838723	838786	
690	**838849**	**838912**	**838975**	**839038**	**839101**	**839164**	**839227**	**839289**	**839352**	**839415**	
1	839478	839541	839604	839667	839729	839792	839855	839918	839981	840043	
2	840106	840169	840232	840294	840357	840420	840482	840545	840608	840671	
3	840733	840796	840859	840921	840984	841046	841109	841172	841234	841297	
4	841359	841422	841485	841547	841610	841672	841735	841797	841860	841922	
5	841985	842047	842110	842172	842235	842297	842360	842422	842484	842547	
6	842609	842672	842734	842796	842859	842921	842983	843046	843108	843170	
7	843233	843295	843357	843420	843482	843544	843606	843669	843731	843793	
8	843855	843918	843980	844042	844104	844166	844229	844291	844353	844415	
9	844477	844539	844601	844664	844726	844788	844850	844912	844974	845036	
700	**845098**	**845160**	**845222**	**845284**	**845346**	**845408**	**845470**	**845532**	**845594**	**845656**	62
1	845718	845780	845842	845904	845966	846028	846090	846151	846213	846275	
2	846337	846399	846461	846523	846585	846646	846708	846770	846832	846894	
3	846955	847017	847079	847141	847202	847264	847326	847388	847449	847511	
4	847573	847634	847696	847758	847819	847881	847943	848004	848066	848128	
5	848189	848251	848312	848374	848435	848497	848559	848620	848682	848743	
6	848805	848866	848928	848989	849051	849112	849174	849235	849297	849358	
7	849419	849481	849542	849604	849665	849726	849788	849849	849911	849972	
8	850033	850095	850156	850217	850279	850340	850401	850462	850524	850585	
9	850646	850707	850769	850830	850891	850952	851014	851075	851136	851197	
710	**851258**	**851320**	**851381**	**851442**	**851503**	**851564**	**851625**	**851686**	**851747**	**851809**	
1	851870	851931	851992	852053	852114	852175	852236	852297	852358	852419	61
2	852480	852541	852602	852663	852724	852785	852846	852907	852968	853029	
3	853090	853150	853211	853272	853333	853394	853455	853516	853577	853637	
4	853698	853759	853820	853881	853941	854002	854063	854124	854185	854245	
5	854306	854367	854428	854488	854549	854610	854670	854731	854792	854852	
6	854913	854974	855034	855095	855156	855216	855277	855337	855398	855459	
7	855519	855580	855640	855701	855761	855822	855882	855943	856003	856064	
8	856124	856185	856245	856306	856366	856427	856487	856548	856608	856668	
9	856729	856789	856850	856910	856970	857031	857091	857152	857212	857272	
720	**857332**	**857393**	**857453**	**857513**	**857574**	**857634**	**857694**	**857755**	**857815**	**857875**	
1	857935	857995	858056	858116	858176	858236	858297	858357	858417	858477	
2	858537	858597	858657	858718	858778	858838	858898	858958	859018	859078	
3	859138	859198	859258	859318	859379	859439	859499	859559	859619	859679	60
4	859739	859799	859859	859918	859978	860038	860098	860158	860218	860278	
5	860338	860398	860458	860518	860578	860637	860697	860757	860817	860877	
6	860937	860996	861056	861116	861176	861236	861295	861355	861415	861475	
7	861534	861594	861654	861714	861773	861833	861893	861952	862012	862072	
8	862131	862191	862251	862310	862370	862430	862489	862549	862608	862668	
9	862728	862787	862847	862906	862966	863025	863085	863144	863204	863263	
730	**863323**	**863382**	**863442**	**863501**	**863561**	**863620**	**863680**	**863739**	**863799**	**863858**	
1	863917	863977	864036	864096	864155	864214	864274	864333	864392	864452	
2	864511	864570	864630	864689	864748	864808	864867	864926	864985	865045	
3	865104	865163	865222	865282	865341	865400	865459	865519	865578	865637	
4	865696	865755	865814	865874	865933	865992	866051	866110	866169	866228	
5	866287	866346	866405	866465	866524	866583	866642	866701	866760	866819	
6	866878	866937	866996	867055	867114	867173	867232	867291	867350	867409	59
7	867467	867526	867585	867644	867703	867762	867821	867880	867939	867998	
8	868056	868115	868174	868233	868292	868350	868409	868468	868527	868586	
9	868644	868703	868762	868821	868879	868938	868997	869056	869114	869173	

Proportional Parts

Diff.	1	2	3	4	5	6	7	8	9
64	6.4	12.8	19.2	25.6	32.0	38.4	44.8	51.2	57.6
62	6.2	12.4	18.6	24.8	31.0	37.2	43.4	49.6	55.8
60	6.0	12.0	18.0	24.0	30.0	36.0	42.0	48.0	54.0
58	5.8	11.6	17.4	23.2	29.0	34.8	40.6	46.4	52.2

N											Diff.
740		**869290**	**869349**	**869408**	**869466**	**869525**	**869584**	**869642**	**869701**	**869760**	
1	869818	869877	869935	869994	870053	870111	870170	870228	870287	870345	
2	870404	870462	870521	870579	870638	870696	870755	870813	870872	870930	
3	870989	871047	871106	871164	871223	871281	871339	871398	871456	871515	
4	871573	871631	871690	871748	871806	871865	871923	871981	872040	872098	
5	872156	872215	872273	872331	872389	872448	872506	872564	872622	872681	
6	872739	872797	872855	872913	872972	873030	873088	873146	873204	873262	
7	873321	873379	873437	873495	873553	873611	873669	873727	873785	873844	
8	873902	873960	874018	874076	874134	874192	874250	874308	874366	874424	58
9	874482	874540	874598	874656	874714	874772	874830	874888	874945	875003	
750	**875061**	**875119**	**875177**	**875235**	**875293**	**875351**	**875409**	**875466**	**875524**	**875582**	
1	875640	875698	875756	875813	875871	875929	875987	876045	876102	876160	
2	876218	876276	876333	876391	876449	876507	876564	876622	876680	876737	
3	876795	876853	876910	876968	877026	877083	877141	877199	877256	877314	
4	877371	877429	877487	877544	877602	877659	877717	877774	877832	877889	
5	877947	878004	878062	878119	878177	878234	878292	878349	878407	878464	
6	878522	878579	878637	878694	878752	878809	878866	878924	878981	879039	
7	879096	879153	879211	879268	879325	879383	879440	879497	879555	879612	
8	879669	879726	879784	879841	879898	879956	880013	880070	880127	880185	
9	880242	880299	880356	880413	880471	880528	880585	880642	880699	880756	
760	**880814**	**880871**	**880928**	**880985**	**881042**	**881099**	**881156**	**881213**	**881271**	**881328**	
1	881385	881442	881499	881556	881613	881670	881727	881784	881841	881898	
2	881955	882012	882069	882126	882183	882240	882297	882354	882411	882468	57
3	882525	882581	882638	882695	882752	882809	882866	882923	882980	883037	
4	883093	883150	883207	883264	883321	883377	883434	883491	883548	883605	
5	883661	883718	883775	883832	883888	883945	884002	884059	884115	884172	
6	884229	884285	884342	884399	884455	884512	884569	884625	884682	884739	
7	884795	884852	884909	884965	885022	885078	885135	885192	885248	885305	
8	885361	885418	885474	885531	885587	885644	885700	885757	885813	885870	
9	885926	885983	886039	886096	886152	886209	886265	886321	886378	886434	
770	**886491**	**886547**	**886604**	**886660**	**886716**	**886773**			**886942**	**886998**	
1	887054	887111	887167	887223	887280	887336	887392	887449	887505	887561	
2	887617	887674	887730	887786	887842	887898	887955	888011	888067	888123	
3	888179	888236	888292	888348	888404	888460	888516	888573	888629	888685	
4	888741	888797	888853	888909	888965	889021	889077	889134	889190	889246	
5	889302	889358	889414	889470	889526	889582	889638	889694	889750	889806	56
6	889862	889918	889974	890030	890086	890141	890197	890253	890309	890365	
7	890421	890477	890533	890589	890645	890700	890756	890812	890868	890924	
8	890980	891035	891091	891147	891203	891259	891314	891370	891426	891482	
9	891537	891593	891649	891705	891760	891816	891872	891928	891983	892039	
780	**892095**	**892150**	**892206**	**892262**	**892317**	**892373**	**892429**	**892484**	**892540**	**892595**	
1	892651	892707	892762	892818	892873	892929	892985	893040	893096	893151	
2	893207	893262	893318	893373	893429	893484	893540	893595	893651	893706	
3	893762	893817	893873	893928	893984	894039	894094	894150	894205	894261	
4	894316	894371	894427	894482	894538	894593	894648	894704	894759	894814	
5	894870	894925	894980	895036	895091	895146	895201	895257	895312	895367	
6	895423	895478	895533	895588	895644	895699	895754	895809	895864	895920	
7	895975	896030	896085	896140	896195	896251	896306	896361	896416	896471	
8	896526	896581	896636	896692	896747	896802	896857	896912	896967	897022	
9	897077	897132	897187	897242	897297	897352	897407	897462	897517	897572	55
790	**897627**	**897682**	**897737**	**897792**	**897847**	**897902**	**897957**	**898012**	**898067**	**898122**	
1	898176	898231	898286	898341	898396	898451	898506	898561	898615	898670	
2	898725	898780	898835	898890	898944	898999	899054	899109	899164	899218	
3	899273	899328	899383	899437	899492	899547	899602	899656	899711	899766	
4	899821	899875	899930	899985	900039	900094	900149	900203	900258	900312	
5	900367	900422	900476	900531	900586	900640	900695	900749	900804	900859	
6	900913	900968	901022	901077	901131	901186	901240	901295	901349	901404	
7	901458	901513	901567	901622	901676	901731	901785	901840	901894	901948	
8	902003	902057	902112	902166	902221	902275	902329	902384	902438	902492	
9	902547	902601	902655	902710	902764	902818	902873	902927	902981	903036	

Proportional Parts

Diff.	1	2	3	4	5	6	7	8	9
60	6.0	12.0	18.0	24.0	30.0	36.0	42.0	48.0	54.0
58	5.8	11.6	17.4	23.2	29.0	34.8	40.6	46.4	52.2
56	5.6	11.2	16.8	22.4	28.0	33.6	39.2	44.8	50.4
54	5.4	10.8	16.2	21.6	27.0	32.4	37.8	43.2	48.6

N	0	1	2	3	4	5	6	7	8	9	Diff.
800	**903090**	**903144**	**903199**	**903253**	**903307**	**903361**	**903416**	**903470**	**903524**	**903578**	
1	903633	903687	903741	903795	903849	903904	903958	904012	904066	904120	
2	904174	904229	904283	904337	904391	904445	904499	904553	904607	904661	
3	904716	904770	904824	904878	904932	904986	905040	905094	905148	905202	
4	905256	905310	905364	905418	905472	905526	905580	905634	905688	905742	54
5	905796	905850	905904	905958	906012	906066	906119	906173	906227	906281	
6	906335	906389	906443	906497	906551	906604	906658	906712	906766	906820	
7	906874	906927	906981	907035	907089	907143	907196	907250	907304	907358	
8	907411	907465	907519	907573	907626	907680	907734	907787	907841	907895	
9	907949	908002	908056	908110	908163	908217	908270	908324	908378	908431	
810	**908485**	**908539**	**908592**	**908646**	**908699**	**908753**	**908807**	**908860**	**908914**	**908967**	
1	909021	909074	909128	909181	909235	909289	909342	909396	909449	909503	
2	909556	909610	909663	909716	909770	909823	909877	909930	909984	910037	
3	910091	910144	910197	910251	910304	910358	910411	910464	910518	910571	
4	910624	910678	910731	910784	910838	910891	910944	910998	911051	911104	
5	911158	911211	911264	911317	911371	911424	911477	911530	911584	911637	
6	911690	911743	911797	911850	911903	911956	912009	912063	912116	912169	
7	912222	912275	912328	912381	912435	912488	912541	912594	912647	912700	
8	912753	912806	912859	912913	912966	913019	913072	913125	913178	913231	
9	913284	913337	913390	913443	913496	913549	913602	913655	913708	913761	53
820	**913814**	**913867**	**913920**	**913973**	**914026**	**914079**	**914132**	**914184**	**914237**	**914290**	
1	914343	914396	914449	914502	914555	914608	914660	914713	914766	914819	
2	914872	914925	914977	915030	915083	915136	915189	915241	915294	915347	
3	915400	915453	915505	915558	915611	915664	915716	915769	915822	915875	
4	915927	915980	916033	916085	916138	916191	916243	916296	916349	916401	
5	916454	916507	916559	916612	916664	916717	916770	916822	916875	916927	
6	916980	917033	917085	917138	917190	917243	917295	917348	917400	917453	
7	917506	917558	917611	917663	917716	917768	917820	917873	917925	917978	
8	918030	918083	918135	918188	918240	918293	918345	918397	918450	918502	
9	918555	918607	918659	918712	918764	918816	918869	918921	918973	919026	
830	**919078**	**919130**	**919183**	**919235**	**919287**	**919340**	**919392**	**919444**	**919496**	**919549**	
1	919601	919653	919706	919758	919810	919862	919914	919967	920019	920071	
2	920123	920176	920228	920280	920332	920384	920436	920489	920541	920593	
3	920645	920697	920749	920801	920853	920906	920958	921010	921062	921114	52
4	921166	921218	921270	921322	921374	921426	921478	921530	921582	921634	
5	921686	921738	921790	921842	921894	921946	921998	922050	922102	922154	
6	922206	922258	922310	922362	922414	922466	922518	922570	922622	922674	
7	922725	922777	922829	922881	922933	922985	923037	923089	923140	923192	
8	923244	923296	923348	923399	923451	923503	923555	923607	923658	923710	
9	923762	923814	923865	923917	923969	924021	924072	924124	924176	924228	
840	**924279**	**924331**	**924383**	**924434**	**924486**	**924538**	**924589**	**924641**	**924693**	**924744**	
1	924796	924848	924899	924951	925003	925054	925106	925157	925209	925261	
2	925312	925364	925415	925467	925518	925570	925621	925673	925725	925776	
3	925828	925879	925931	925982	926034	926085	926137	926188	926240	926291	
4	926342	926394	926445	926497	926548	926600	926651	926702	926754	926805	
5	926857	926908	926959	927011	927062	927114	927165	927216	927268	927319	
6	927370	927422	927473	927524	927576	927627	927678	927730	927781	927832	
7	927883	927935	927986	928037	928088	928140	928191	928242	928293	928345	
8	928396	928447	928498	928549	928601	928652	928703	928754	928805	928857	
9	928908	928959	929010	929061	929112	929163	929215	929266	929317	929368	
850	**929419**	**929470**	**929521**	**929572**	**929623**	**929674**	**929725**	**929776**	**929827**	**929879**	
1	929930	929981	930032	930083	930134	930185	930236	930287	930338	930389	51
2	930440	930491	930542	930592	930643	930694	930745	930796	930847	930898	
3	930949	931000	931051	931102	931153	931204	931254	931305	931356	931407	
4	931458	931509	931560	931610	931661	931712	931763	931814	931865	931915	
5	931966	932017	932068	932118	932169	932220	932271	932322	932372	932423	
6	932474	932524	932575	932626	932677	932727	932778	932829	932879	932930	
7	932981	933031	933082	933133	933183	933234	933285	933335	933386	933437	
8	933487	933538	933589	933639	933690	933740	933791	933841	933892	933943	
9	933993	934044	934094	934145	934195	934246	934296	934347	934397	934448	

PROPORTIONAL PARTS

Diff.	1	2	3	4	5	6	7	8	9
56	5.6	11.2	16.8	22.4	28.0	33.6	39.2	44.8	50.4
54	5.4	10.8	16.2	21.6	27.0	32.4	37.8	43.2	48.6
52	5.2	10.4	15.6	20.8	26.0	31.2	36.4	41.6	46.8

N	0	1	2	3	4	5	6	7	8	9	Diff.
860	**934498**	**934549**	**934599**	**934650**	**934700**	**934751**	**934801**	**934852**	**934902**	**934953**	
1	935003	935054	935104	935154	935205	935255	935306	935356	935406	935457	
2	935507	935558	935608	935658	935709	935759	935809	935860	935910	935960	
3	936011	936061	936111	936162	936212	936262	936313	936363	936413	936463	
4	936514	936564	936614	936665	936715	936765	936815	936865	936916	936966	
5	937016	937066	937116	937167	937217	937267	937317	937367	937418	937468	
6	937518	937568	937618	937668	937718	937769	937819	937869	937919	937969	
7	938019	938069	938119	938169	938219	938269	938320	938370	938420	938470	50
8	938520	938570	938620	938670	938720	938770	938820	938870	938920	938970	
9	939020	939070	939120	939170	939220	939270	939320	939369	939419	939469	
870	**939519**	**939569**	**939619**	**939669**	**939719**	**939769**	**939819**	**939869**	**939918**	**939968**	
1	940018	940068	940118	940168	940218	940267	940317	940367	940417	940467	
2	940516	940566	940616	940666	940716	940765	940815	940865	940915	940964	
3	941014	941064	941114	941163	941213	941263	941313	941362	941412	941462	
4	941511	941561	941611	941660	941710	941760	941809	941859	941909	941958	
5	942008	942058	942107	942157	942207	942256	942306	942355	942405	942455	
6	942504	942554	942603	942653	942702	942752	942801	942851	942901	942950	
7	943000	943049	943099	943148	943198	943247	943297	943346	943396	943445	
8	943495	943544	943593	943643	943692	943742	943791	943841	943890	943939	
9	943989	944038	944088	944137	944186	944236	944285	944335	944384	944433	
880	**944483**	**944532**	**944581**	**944631**	**944680**	**944729**	**944779**	**944828**	**944877**	**944927**	
1	944976	945025	945074	945124	945173	945222	945272	945321	945370	945419	
2	945469	945518	945567	945616	945665	945715	945764	945813	945862	945912	
3	945961	946010	946059	946108	946157	946207	946256	946305	946354	946403	
4	946452	946501	946551	946600	946649	946698	946747	946796	946845	946894	
5	946943	946992	947041	947090	947140	947189	947238	947287	947336	947385	
6	947434	947483	947532	947581	947630	947679	947728	947777	947826	947875	49
7	947924	947973	948022	948070	948119	948168	948217	948266	948315	948364	
8	948413	948462	948511	948560	948608	948657	948706	948755	948804	948853	
9	948902	948951	948999	949048	949097	949146	949195	949244	949292	949341	
890	**949390**	**949439**	**949488**	**949536**	**949585**	**949634**	**949683**	**949731**	**949780**	**949829**	
1	949878	949926	949975	950024	950073	950121	950170	950219	950267	950316	
2	950365	950414	950462	950511	950560	950608	950657	950706	950754	950803	
3	950851	950900	950949	950997	951046	951095	951143	951192	951240	951289	
4	951338	951386	951435	951483	951532	951580	951629	951677	951726	951775	
5	951823	951872	951920	951969	952017	952066	952114	952163	952211	952260	
6	952308	952356	952405	952453	952502	952550	952599	952647	952696	952744	
7	952792	952841	952889	952938	952986	953034	953083	953131	953180	953228	
8	953276	953325	953373	953421	953470	953518	953566	953615	953663	953711	
9	953760	953808	953856	953905	953953	954001	954049	954098	954146	954194	
900	**954243**	**954291**	**954339**	**954387**	**954435**	**954484**	**954532**	**954580**	**954628**	**954677**	
1	954725	954773	954821	954869	954918	954966	955014	955062	955110	955158	
2	955207	955255	955303	955351	955399	955447	955495	955543	955592	955640	
3	955688	955736	955784	955832	955880	955928	955976	956024	956072	956120	
4	956168	956216	956265	956313	956361	956409	956457	956505	956553	956601	
5	956649	956697	956745	956793	956840	956888	956936	956984	957032	957080	48
6	957128	957176	957224	957272	957320	957368	957416	957464	957512	957559	
7	957607	957655	957703	957751	957799	957847	957894	957942	957990	958038	
8	958086	958134	958181	958229	958277	958325	958373	958421	958468	958516	
9	958564	958612	958659	958707	958755	958803	958850	958898	958946	958994	
910	**959041**	**959089**	**959137**	**959185**	**959232**	**959280**	**959328**	**959375**	**959423**	**959471**	
1	959518	959566	959614	959661	959709	959757	959804	959852	959900	959947	
2	959995	960042	960090	960138	960185	960233	960280	960328	960376	960423	
3	960471	960518	960566	960613	960661	960709	960756	960804	960851	960899	
4	960946	960994	961041	961089	961136	961184	961231	961279	961326	961374	
5	961421	961469	961516	961563	961611	961658	961706	961753	961801	961848	
6	961895	961943	961990	962038	962085	962132	962180	962227	962275	962322	
7	962369	962417	962464	962511	962559	962606	962653	962701	962748	962795	
8	962843	962890	962937	962985	963032	963079	963126	963174	963221	963268	
9	963316	963363	963410	963457	963504	963552	963599	963646	963693	963741	

PROPORTIONAL PARTS

Diff.	1	2	3	4	5	6	7	8	9
52	5.2	10.4	15.6	20.8	26.0	31.2	36.4	41.6	46.8
50	5.0	10.0	15.0	20.0	25.0	30.0	35.0	40.0	45.0
48	4.8	9.6	14.4	19.2	24.0	28.8	33.6	38.4	43.2

N	0	1	2	3	4	5	6	7	8	9	Diff.
920	**963788**	**963835**	**963882**	**963929**	**963977**	**964024**	**964071**	**964118**	**964165**	**964212**	
1	964260	964307	964354	964401	964448	964495	964542	964590	964637	964684	
2	964731	964778	964825	964872	964919	964966	965013	965061	965108	965155	
3	965202	965249	965296	965343	965390	965437	965484	965531	965578	965625	
4	965672	965719	965766	965813	965860	965907	965954	966001	966048	966095	47
5	966142	966189	966236	966283	966329	966376	966423	966470	966517	966564	
6	966611	966658	966705	966752	966799	966845	966892	966939	966986	967033	
7	967080	967127	967173	967220	967267	967314	967361	967408	967454	967501	
8	967548	967595	967642	967688	967735	967782	967829	967875	967922	967969	
9	968016	968062	968109	968156	968203	968249	968296	968343	968390	968436	
930	**968483**	**968530**	**968576**	**968623**	**968670**	**968716**	**968763**	**968810**	**968856**	**968903**	
1	968950	968996	969043	969090	969136	969183	969229	969276	969323	969369	
2	969416	969463	969509	969556	969602	969649	969695	969742	969789	969835	
3	969882	969928	969975	970021	970068	970114	970161	970207	970254	970300	
4	970347	970393	970440	970486	970533	970579	970626	970672	970719	970765	
5	970812	970858	970904	970951	970997	971044	971090	971137	971183	971229	
6	971276	971322	971369	971415	971461	971508	971554	971601	971647	971693	
7	971740	971786	971832	971879	971925	971971	972018	972064	972110	972157	
8	972203	972249	972295	972342	972388	972434	972481	972527	972573	972619	
9	972666	972712	972758	972804	972851	972897	972943	972989	973035	973082	
940	**973128**	**973174**	**973220**	**973266**	**973313**	**973359**	**973405**	**973451**	**973497**	**973543**	
1	973590	973636	973682	973728	973774	973820	973866	973913	973959	974005	
2	974051	974097	974143	974189	974235	974281	974327	974374	974420	974466	
3	974512	974558	974604	974650	974696	974742	974788	974834	974880	974926	
4	974972	975018	975064	975110	975156	975202	975248	975294	975340	975386	46
5	975432	975478	975524	975570	975616	975662	975707	975753	975799	975845	
6	975891	975937	975983	976029	976075	976121	976167	976212	976258	976304	
7	976350	976396	976442	976488	976533	976579	976625	976671	976717	976763	
8	976808	976854	976900	976946	976992	977037	977083	977129	977175	977220	
9	977266	977312	977358	977403	977449	977495	977541	977586	977632	977678	
950	**977724**	**977769**	**977815**	**977861**	**977906**	**977952**	**977998**	**978043**	**978089**	**978135**	
1	978181	978226	978272	978317	978363	978409	978454	978500	978546	978591	
2	978637	978683	978728	978774	978819	978865	978911	978956	979002	979047	
3	979093	979138	979184	979230	979275	979321	979366	979412	979457	979503	
4	979548	979594	979639	979685	979730	979776	979821	979867	979912	979958	
5	980003	980049	980094	980140	980185	980231	980276	980322	980367	980412	
6	980458	980503	980549	980594	980640	980685	980730	980776	980821	980867	
7	980912	980957	981003	981048	981093	981139	981184	981229	981275	981320	
8	981366	981411	981456	981501	981547	981592	981637	981683	981728	981773	
9	981819	981864	981909	981954	982000	982045	982090	982135	982181	982226	
960	**982271**	**982316**	**982362**	**982407**	**982452**	**982497**	**982543**	**982588**	**982633**	**982678**	
1	982723	982769	982814	982859	982904	982949	982994	983040	983085	983130	
2	983175	983220	983265	983310	983356	983401	983446	983491	983536	983581	
3	983626	983671	983716	983762	983807	983852	983897	983942	983987	984032	
4	984077	984122	984167	984212	984257	984302	984347	984392	984437	984482	
5	984527	984572	984617	984662	984707	984752	984797	984842	984887	984932	45
6	984977	985022	985067	985112	985157	985202	985247	985292	985337	985382	
7	985426	985471	985516	985561	985606	985651	985696	985741	985786	985830	
8	985875	985920	985965	986010	986055	986100	986144	986189	986234	986279	
9	986324	986369	986413	986458	986503	986548	986593	986637	986682	986727	
970	**986772**	**986817**	**986861**	**986906**	**986951**	**986996**	**987040**	**987085**	**987130**	**987175**	
1	987219	987264	987309	987353	987398	987443	987488	987532	987577	987622	
2	987666	987711	987756	987800	987845	987890	987934	987979	988024	988068	
3	988113	988157	988202	988247	988291	988336	988381	988425	988470	988514	
4	988559	988604	988648	988693	988737	988782	988826	988871	988916	988960	
5	989005	989049	989094	989138	989183	989227	989272	989316	989361	989405	
6	989450	989494	989539	989583	989628	989672	989717	989761	989806	989850	
7	989895	989939	989983	990028	990072	990117	990161	990206	990250	990294	
8	990339	990383	990428	990472	990516	990561	990605	990650	990694	990738	
9	990783	990827	990871	990916	990960	991004	991049	991093	991137	991182	

PROPORTIONAL PARTS

Diff.	1	2	3	4	5	6	7	8	9
48	4.8	9.6	14.4	19.2	24.0	28.8	33.6	38.4	43.2
46	4.6	9.2	13.8	18.4	23.0	27.6	32.2	36.8	41.4
44	4.4	8.8	13.2	17.6	22.0	26.4	30.8	35.2	39.6
42	4.2	8.4	12.6	16.8	21.0	25.2	29.4	33.6	37.8

N	0	1	2	3	4	5	6	7	8	9	Diff.
980	**991226**	**991270**	**991315**	**991359**	**991403**	**991448**	**991492**	**991536**	**991580**	**991625**	
1	991669	991713	991758	991802	991846	991890	991935	991979	992023	992067	
2	992111	992156	992200	992244	992288	992333	992377	992421	992465	992509	
3	992554	992598	992642	992686	992730	992774	992819	992863	992907	992951	
4	992995	993039	993083	993127	993172	993216	993260	993304	993348	993392	
5	993436	993480	993524	993568	993613	993657	993701	993745	993789	993833	
6	993877	993921	993965	994009	994053	994097	994141	994185	994229	994273	
7	994317	994361	994405	994449	994493	994537	994581	994625	994669	994713	44
8	994757	994801	994845	994889	994933	994977	995021	995065	995108	995152	
9	995196	995240	995284	995328	995372	995416	995460	995504	995547	995591	
990	**995635**	**995679**	**995723**	**995767**	**995811**	**995854**	**995898**	**995942**	**995986**	**996030**	
1	996074	996117	996161	996205	996249	996293	996337	996380	996424	996468	
2	996512	996555	996599	996643	996687	996731	996774	996818	996862	996906	
3	996949	996993	997037	997080	997124	997168	997212	997255	997299	997343	
4	997386	997430	997474	997517	997561	997605	997648	997692	997736	997779	
5	997823	997867	997910	997954	997998	998041	998085	998129	998172	998216	
6	998259	998303	998347	998390	998434	998477	998521	998564	998608	998652	
7	998695	998739	998782	998826	998869	998913	998956	999000	999043	999087	
8	999131	999174	999218	999261	999305	999348	999392	999435	999479	999522	
9	999565	999609	999652	999696	999739	999783	999826	999870	999913	999957	
1000	**000000**	**000043**	**000087**	**000130**	**000174**	**000217**	**000260**	**000304**	**000347**	**000391**	43

Table 3.—Natural (Napierian or Hyperbolic) Logarithms of Numbers

Table gives natural logarithms of numbers from 1.0 to 9.99 directly. To find logarithms of numbers outside that range add or subtract natural logarithm of the powers of 10 as follows:

$\log_e 10 = 2.302585$ $\qquad \log_e 10^4 = 9.210340$ $\qquad \log_e 10^7 = 16.118096$

$\log_e 10^2 = 4.605170$ $\qquad \log_e 10^5 = 11.512925$ $\qquad \log_e 10^8 = 18.420681$

$\log_e 10^3 = 6.907755$ $\qquad \log_e 10^6 = 13.815511$ $\qquad \log_e 10^9 = 20.703266$

Example.—$\log_e 679. = \log_e 6.79 + \log_e 10^2 = 1.9155 + 4.6052 = 5.5207$

$\log_e .0679 = \log_e 6.79 - \log_e 10^2 = 1.9155 - 4.6052 = -2.6897$

The common logarithm is the natural logarithm multiplied by the modulus of $\log_{10}$; $\log_{10} N = 0.434294 \log_e N$.

N	0	1	2	3	4	5	6	7	8	9
1.0	**0.0000**	**0.0100**	**0.0198**	**0.0296**	**0.0392**	**0.0488**	**0.0583**	**0.0677**	**0.0770**	**0.0862**
1.1	0.0953	0.1044	0.1133	0.1222	0.1310	0.1398	0.1484	0.1570	0.1655	0.1740
1.2	0.1823	0.1906	0.1989	0.2070	0.2151	0.2231	0.2311	0.2390	0.2469	0.2546
1.3	0.2624	0.2700	0.2776	0.2852	0.2927	0.3001	0.3075	0.3148	0.3221	0.3293
1.4	0.3365	0.3436	0.3507	0.3577	0.3646	0.3716	0.3784	0.3853	0.3920	0.3988
1.5	0.4055	0.4121	0.4187	0.4253	0.4318	0.4383	0.4447	0.4511	0.4574	0.4637
1.6	0.4700	0.4762	0.4824	0.4886	0.4947	0.5008	0.5068	0.5128	0.5188	0.5247
1.7	0.5306	0.5365	0.5423	0.5481	0.5539	0.5596	0.5653	0.5710	0.5766	0.5822
1.8	0.5878	0.5933	0.5988	0.6043	0.6098	0.6152	0.6206	0.6259	0.6313	0.6366
1.9	0.6419	0.6471	0.6523	0.6575	0.6627	0.6678	0.6729	0.6780	0.6831	0.6881
2.0	**0.6931**	**0.6981**	**0.7031**	**0.7080**	**0.7129**	**0.7178**	**0.7227**	**0.7275**	**0.7324**	**0.7372**
2.1	0.7419	0.7467	0.7514	0.7561	0.7608	0.7655	0.7701	0.7747	0.7793	0.7839
2.2	0.7885	0.7930	0.7975	0.8020	0.8065	0.8109	0.8154	0.8198	0.8242	0.8286
2.3	0.8329	0.8372	0.8416	0.8459	0.8502	0.8544	0.8587	0.8629	0.8671	0.8713
2.4	0.8755	0.8796	0.8838	0.8879	0.8920	0.8961	0.9002	0.9042	0.9083	0.9123
2.5	0.9163	0.9203	0.9243	0.9282	0.9322	0.9361	0.9400	0.9439	0.9478	0.9517
2.6	0.9555	0.9594	0.9632	0.9670	0.9708	0.9746	0.9783	0.9821	0.9858	0.9895
2.7	0.9933	0.9969	1.0006	1.0043	1.0080	1.0116	1.0152	1.0188	1.0225	1.0260
2.8	1.0296	1.0332	1.0367	1.0403	1.0438	1.0473	1.0508	1.0543	1.0578	1.0613
2.9	1.0647	1.0682	1.0716	1.0750	1.0784	1.0818	1.0852	1.0886	1.0919	1.0953
3.0	**1.0986**	**1.1019**	**1.1053**	**1.1086**	**1.1119**	**1.1151**	**1.1184**	**1.1217**	**1.1249**	**1.1282**
3.1	1.1314	1.1346	1.1378	1.1410	1.1442	1.1474	1.1506	1.1537	1.1569	1.1600
3.2	1.1632	1.1663	1.1694	1.1725	1.1756	1.1787	1.1817	1.1848	1.1878	1.1909
3.3	1.1939	1.1969	1.2000	1.2030	1.2060	1.2090	1.2119	1.2149	1.2179	1.2208
3.4	1.2238	1.2267	1.2296	1.2326	1.2355	1.2384	1.2413	1.2442	1.2470	1.2499
3.5	1.2528	1.2556	1.2585	1.2613	1.2641	1.2669	1.2698	1.2726	1.2754	1.2782
3.6	1.2809	1.2837	1.2865	1.2892	1.2920	1.2947	1.2975	1.3002	1.3029	1.3056
3.7	1.3083	1.3110	1.3137	1.3164	1.3191	1.3218	1.3244	1.3271	1.3297	1.3324
3.8	1.3350	1.3376	1.3403	1.3429	1.3455	1.3481	1.3507	1.3533	1.3558	1.3584
3.9	1.3610	1.3635	1.3661	1.3686	1.3712	1.3737	1.3762	1.3788	1.3813	1.3838

(Table continued on following page)

Table 3.—Natural (Napierian or Hyperbolic) Logarithms of Numbers (*Continued*)

N	0	1	2	3	4	5	6	7	8	9
4.0	**1.3863**	**1.3888**	**1.3913**	**1.3938**	**1.3962**	**1.3987**	**1.4012**	**1.4036**	**1.4061**	**1.4085**
4.1	1.4110	1.4134	1.4159	1.4183	1.4207	1.4231	1.4255	1.4279	1.4303	1.4327
4.2	1.4351	1.4375	1.4398	1.4422	1.4446	1.4469	1.4493	1.4516	1.4540	1.4563
4.3	1.4586	1.4609	1.4633	1.4656	1.4679	1.4702	1.4725	1.4748	1.4770	1.4793
4.4	1.4816	1.4839	1.4861	1.4884	1.4907	1.4929	1.4951	1.4974	1.4996	1.5019
4.5	1.5041	1.5063	1.5085	1.5107	1.5129	1.5151	1.5173	1.5195	1.5217	1.5239
4.6	1.5261	1.5282	1.5304	1.5326	1.5347	1.5369	1.5390	1.5412	1.5433	1.5454
4.7	1.5476	1.5497	1.5518	1.5539	1.5560	1.5581	1.5602	1.5623	1.5644	1.5665
4.8	1.5686	1.5707	1.5728	1.5748	1.5769	1.5790	1.5810	1.5831	1.5851	1.5872
4.9	1.5892	1.5913	1.5933	1.5953	1.5974	1.5994	1.6014	1.6034	1.6054	1.6074
5.0	**1.6094**	**1.6114**	**1.6134**	**1.6154**	**1.6174**	**1.6194**	**1.6214**	**1.6233**	**1.6253**	**1.6273**
5.1	1.6292	1.6312	1.6332	1.6351	1.6371	1.6390	1.6409	1.6429	1.6448	1.6467
5.2	1.6487	1.6506	1.6525	1.6544	1.6563	1.6582	1.6601	1.6620	1.6639	1.6658
5.3	1.6677	1.6696	1.6715	1.6734	1.6752	1.6771	1.6790	1.6808	1.6827	1.6845
5.4	1.6864	1.6882	1.6901	1.6919	1.6938	1.6956	1.6974	1.6993	1.7011	1.7029
5.5	1.7047	1.7066	1.7084	1.7102	1.7120	1.7138	1.7156	1.7174	1.7192	1.7210
5.6	1.7228	1.7246	1.7263	1.7281	1.7299	1.7317	1.7334	1.7352	1.7370	1.7387
5.7	1.7405	1.7422	1.7440	1.7457	1.7475	1.7492	1.7509	1.7527	1.7544	1.7561
5.8	1.7579	1.7596	1.7613	1.7630	1.7647	1.7664	1.7681	1.7699	1.7716	1.7733
5.9	1.7750	1.7766	1.7783	1.7800	1.7817	1.7834	1.7851	1.7867	1.7884	1.7901
6.0	**1.7918**	**1.7934**	**1.7951**	**1.7967**	**1.7984**	**1.8001**	**1.8017**	**1.8034**	**1.8050**	**1.8066**
6.1	1.8083	1.8099	1.8116	1.8132	1.8148	1.8165	1.8181	1.8197	1.8213	1.8229
6.2	1.8245	1.8262	1.8278	1.8294	1.8310	1.8326	1.8342	1.8358	1.8374	1.8390
6.3	1.8405	1.8421	1.8437	1.8453	1.8469	1.8485	1.8500	1.8516	1.8532	1.8547
6.4	1.8563	1.8579	1.8594	1.8610	1.8625	1.8641	1.8656	1.8672	1.8687	1.8703
6.5	1.8718	1.8733	1.8749	1.8764	1.8779	1.8795	1.8810	1.8825	1.8840	1.8856
6.6	1.8871	1.8886	1.8901	1.8916	1.8931	1.8946	1.8961	1.8976	1.8991	1.9006
6.7	1.9021	1.9036	1.9051	1.9066	1.9081	1.9095	1.9110	1.9125	1.9140	1.9155
6.8	1.9169	1.9184	1.9199	1.9213	1.9228	1.9242	1.9257	1.9272	1.9286	1.9301
6.9	1.9315	1.9330	1.9344	1.9359	1.9373	1.9387	1.9402	1.9416	1.9430	1.9445
7.0	**1.9459**	**1.9473**	**1.9488**	**1.9502**	**1.9515**	**1.9530**	**1.9544**	**1.9559**	**1.9573**	**1.9587**
7.1	1.9601	1.9615	1.9629	1.9643	1.9657	1.9671	1.9685	1.9699	1.9713	1.9727
7.2	1.9741	1.9755	1.9769	1.9782	1.9796	1.9810	1.9824	1.9838	1.9851	1.9865
7.3	1.9879	1.9892	1.9906	1.9920	1.9933	1.9947	1.9961	1.9974	1.9988	2.0001
7.4	2.0015	2.0028	2.0042	2.0055	2.0069	2.0082	2.0096	2.0109	2.0122	2.0136
7.5	2.0149	2.0162	2.0176	2.0189	2.0202	2.0215	2.0229	2.0242	2.0255	2.0268
7.6	2.0281	2.0295	2.0308	2.0321	2.0334	2.0347	2.0360	2.0373	2.0386	2.0399
7.7	2.0412	2.0425	2.0438	2.0451	2.0464	2.0477	2.0490	2.0503	2.0516	2.0528
7.8	2.0541	2.0554	2.0567	2.0580	2.0592	2.0605	2.0618	2.0631	2.0643	2.0656
7.9	2.0669	2.0681	2.0694	2.0707	2.0719	2.0732	2.0744	2.0757	2.0769	2.0782
8.0	**2.0794**	**2.0807**	**2.0819**	**2.0832**	**2.0844**	**2.0857**	**2.0869**	**2.0882**	**2.0894**	**2.0906**
8.1	2.0919	2.0931	2.0943	2.0956	2.0968	2.0980	2.0992	2.1005	2.1017	2.1029
8.2	2.1041	2.1054	2.1066	2.1078	2.1090	2.1102	2.1114	2.1126	2.1138	2.1150
8.3	2.1163	2.1175	2.1187	2.1199	2.1211	2.1223	2.1235	2.1247	2.1258	2.1270
8.4	2.1282	2.1294	2.1306	2.1318	2.1330	2.1342	2.1353	2.1365	2.1377	2.1389
8.5	2.1401	2.1412	2.1424	2.1436	2.1448	2.1459	2.1471	2.1483	2.1494	2.1506
8.6	2.1518	2.1529	2.1541	2.1552	2.1564	2.1576	2.1587	2.1599	2.1610	2.1622
8.7	2.1633	2.1645	2.1656	2.1668	2.1679	2.1691	2.1702	2.1713	2.1725	2.1736
8.8	2.1748	2.1759	2.1770	2.1782	2.1793	2.1804	2.1815	2.1827	2.1838	2.1849
8.9	2.1861	2.1872	2.1883	2.1894	2.1905	2.1917	2.1928	2.1939	2.1950	2.1961
9.0	**2.1972**	**2.1983**	**2.1994**	**2.2006**	**2.2017**	**2.2028**	**2.2039**	**2.2050**	**2.2061**	**2.2072**
9.1	2.2083	2.2094	2.2105	2.2116	2.2127	2.2138	2.2148	2.2159	2.2170	2.2181
9.2	2.2192	2.2203	2.2214	2.2225	2.2235	2.2246	2.2257	2.2268	2.2279	2.2289
9.3	2.2300	2.2311	2.2322	2.2332	2.2343	2.2354	2.2364	2.2375	2.2386	2.2396
9.4	2.2407	2.2418	2.2428	2.2439	2.2450	2.2460	2.2471	2.2481	2.2492	2.2502
9.5	2.2513	2.2523	2.2534	2.2544	2.2555	2.2565	2.2576	2.2586	2.2597	2.2607
9.6	2.2618	2.2628	2.2638	2.2649	2.2659	2.2670	2.2680	2.2690	2.2701	2.2711
9.7	2.2721	2.2732	2.2742	2.2752	2.2762	2.2773	2.2783	2.2793	2.2803	2.2814
9.8	2.2824	2.2834	2.2844	2.2854	2.2865	2.2875	2.2885	2.2895	2.2905	2.2915
9.9	2.2925	2.2935	2.2946	2.2956	2.2966	2.2976	2.2986	2.2996	2.3006	2.3016

Table 4.—Properties of Numbers

Decimal Equivalents, Squares, Cubes, Three-halves Powers, Square Roots, Cube Roots, Fifth Roots, Reciprocals, Circumference and Area of Circles

Number, N		N^2	N^3	$\sqrt{N}$	$\sqrt[3]{N}$	$N^{3/2}$	$\sqrt[5]{N}$	$\frac{1}{N}$	Circle (N= Diam.)	
Fraction	Decimal								Circum.	Area
1/64	.015625	0.000244	$.381\times 10^{-5}$	.1250	.2500	.00195	.4353	64.0	0.04909	.00019
1/32	.03125	.000977	$.305\times 10^{-4}$	.1768	.3150	.00552	.5000	32.0	.0981[illegible]	.00077
3/64	.046875	.002197	$.103\times 10^{-3}$	.2165	.3606	.01015	.5422	21.3333	.14726	.00173
1/16	.0625	.003906	$.244\times 10^{-3}$	.2500	.3969	.01563	.5744	16.0	.1963[illegible]	.00307
5/64	.078125	.006104	$.477\times 10^{-3}$	.2795	.4275	.02184	.6006	12.80	.24544	.00479
3/32	.09375	.008789	$.824\times 10^{-3}$	.3062	.4543	.02871	.6229	10.6667	.29452	.00690
	.10	.010	.00100	.3162	.4642	.03162	.6310	10.0	.31416	.00785
7/64	.109375	.01196	.001308	.3307	.4782	.03617	.6424	9.1429	.34361	.00939
1/8	.125	.01563	.001953	.3536	.5000	.04419	.6598	8.0	.39270	.01227
9/64	.140625	.01978	.002782	.3750	.5200	.05273	.6755	7.1111	.44179	.01554
5/32	.15625	.02441	.003814	.3953	.5386	.06176	.6899	6.40	.49087	.01917
11/64	.171875	.02954	.005077	.4146	.5560	.07126	.7031	5.8182	.53990	.02320
3/16	.1875	.03516	.006592	.4330	.5724	.08119	.7155	5.3333	.5590[illegible]	.02761
	.20	.040	.0080	.4472	.5848	.08944	.7248	5.0	.62832	.03142
13/64	.203125	.04126	.008381	.4507	.5878	.09155	.7270	4.9231	.63814	.03241
7/32	.21875	.04785	.01047	.4677	.6025	.10231	.7379	4.5714	.68722	.03758
15/64	.234375	.05493	.01287	.4841	.6166	.11347	.7481	4.2667	.73631	.04314
1/4	.250	.0625	.01563	.5000	.6300	.12500	.7579	4.0	.78540	.04909
17/64	.265625	.07056	.01874	.5154	.6428	.13690	.7671	3.7647	.83448	.05542
9/32	.28125	.07910	.02225	.5303	.6552	.14916	.7759	3.5556	.88357	.06213
19/64	.296875	.08813	.02616	.5449	.6671	.16176	.7844	3.3684	.93266	.06922
	.30	.090	.0270	.5477	.6694	.16432	.7860	3.3333	.94240	.07069
5/16	.3125	.09766	.03052	.5590	.6786	.17469	.7925	3.2000	.9817[illegible]	.07670
21/64	.328125	.10767	.03533	.5728	.6897	.18796	.8002	3.0476	1.0308	.08456
11/32	.34375	.11816	.04062	.5863	.7005	.20154	.8077	2.9091	1.0799	.09281
23/64	.359375	.12915	.04641	.5995	.7110	.21544	.8149	2.7826	1.1290	.10143
3/8	.375	.14063	.05273	.6124	.7211	.22964	.8219	2.6667	1.1781	.11045
25/64	.390625	.15259	.05961	.6250	.7310	.24414	.8286	2.5600	1.2272	.11984
	.40	.16	.0640	.6325	.7368	.25298	.8326	2.50	1.2566	.12566
13/32	.40625	.16504	.06705	.6374	.7406	.25894	.8351	2.4615	1.2763	.12962
27/64	.421875	.17798	.07508	.6495	.7500	.27402	.8415	2.3704	1.3254	.13979
7/16	.4375	.19141	.08374	.6614	.7592	.28938	.8476	2.2857	1.3744	.15033
29/64	.453125	.20532	.09304	.6732	.7681	.30502	.8536	2.2069	1.4235	.16126
15/32	.46875	.21973	.10300	.6847	.7768	.32093	.8594	2.1333	1.4726	.17257
31/64	.484375	.23462	.11364	.6960	.7854	.33711	.8650	2.0645	1.5217	.18427
1/2	.50	.2500	.12500	.7071	.7937	.35355	.8706	2.0	1.5708	.19635
33/64	.515625	.26587	.13709	.7181	.8019	.37025	.8759	1.9394	1.6199	.20881
17/32	.53125	.28223	.14993	.7289	.8099	.38721	.8812	1.8824	1.6690	.22166
35/64	.546875	.29907	.16355	.7395	.8178	.40442	.8863	1.8286	1.7181	.23489
9/16	.5625	.31641	.17798	.7500	.8255	.42188	.8913	1.7778	1.7671	.24850
37/64	.578125	.33423	.19323	.7604	.8331	.43957	.8962	1.7297	1.8162	.26250
19/32	.59375	.35254	.20932	.7706	.8405	.45751	.9010	1.6842	1.8653	.27688
	.60	.3600	.21600	.7746	.8434	.46476	.9029	1.6667	1.8850	.28274
39/64	.609375	.37134	.22628	.7806	.8478	.47569	.9057	1.6410	1.9144	.29165
5/8	.625	.39063	.24414	.7906	.8550	.49410	.9103	1.6000	1.9635	.30680
41/64	.640625	.41040	.26291	.8004	.8621	.51275	.9148	1.5610	2.0126	.32233
21/32	.65625	.43066	.28262	.8101	.8690	.53162	.9192	1.5238	2.0617	.33824
43/64	.671875	.45142	.30330	.8197	.8759	.55072	.9235	1.4884	2.1108	.35454
11/16	.6875	.47266	.32495	.8297	.8826	.57005	.9278	1.4545	2.1598	.37122
	.70	.4900	.34300	.8367	.8879	.58566	.9312	1.4286	2.1991	.38485
45/64	.703125	.49438	.34761	.8385	.8892	.58959	.9320	1.4222	2.2089	.38829
23/32	.71875	.51660	.37131	.8478	.8958	.60935	.9361	1.3913	2.2580	.40574
47/64	.734375	.53931	.39605	.8570	.9022	.62933	.9401	1.3617	2.3071	.42357
3/4	.750	.56250	.42188	.8660	.9086	.64952	.9441	1.3333	2.3562	.44179
49/64	.765625	.58618	.44879	.8750	.9148	.66992	.9480	1.3061	2.4053	.46038
25/32	.78125	.61035	.47684	.8839	.9210	.69053	.9518	1.2800	2.4544	.47937
51/64	.796875	.63501	.50602	.8927	.9271	.71135	.9556	1.2549	2.5035	.49874
	.80	.6400	.51200	.8944	.9283	.71554	.9564	1.2500	2.5133	.50265
13/16	.8125	.66016	.53638	.9014	.9331	.73238	.9593	1.2308	2.5525	.51849
53/64	.828125	.68579	.56792	.9100	.9391	.75361	.9630	1.2075	2.6016	.53862
27/32	.84375	.71191	.60067	.9186	.9449	.77503	.9666	1.1852	2.6507	.55914
55/64	.859375	.73853	.63467	.9270	.9507	.79666	.9702	1.1636	2.6998	.58004
7/8	.875	.76563	.66992	.9354	.9565	.81849	.9737	1.1429	2.7489	.60132
57/64	.890625	.79321	.70645	.9437	.9621	.84051	.9771	1.1228	2.7980	.62299
	.90	.81000	.72900	.9487	.9655	.85435	.9792	1.1111	2.8274	.63617
29/32	.90625	.82129	.74429	.9520	.9677	.86272	.9805	1.1034	2.8471	.64504
59/64	.921875	.84985	.78346	.9601	.9733	.88513	.9839	1.0847	2.8962	.66747
15/16	.9375	.87891	.82398	.9683	.9787	.90773	.9872	1.0667	2.9452	.69029
61/64	.953125	.90845	.86587	.9763	.9841	.93053	.9905	1.0492	2.9943	.71349
31/32	.96875	.93848	.90915	.9843	.9895	.95349	.9937	1.0323	3.0434	.73708
63/64	.984375	.96899	.95385	.9922	.9948	.97666	.9969	1.0159	3.0925	.76104

Table 4.—Properties of Numbers—*Continued*

N	N^2	N^3	$\sqrt{N}$	$\sqrt[3]{N}$	$N^{3/2}$	$\sqrt[5]{N}$	$\frac{1}{N}$	Circle (N = Diam.)	
								Circum.	Area
1.	**1.0000**	**1.0000**	**1.0000**	1.0000	**1.0000**	**1.0000**	**1.0000000**	**3.1416**	**0.7854**
1.125	1.2656	1.4238	1.0606	1.0400	1.1932	1.0238	.8888888	3.5343	.9940
1.25	1.5625	1.9531	1.1180	1.0772	1.3975	1.0456	.80000000	3.9270	1.2272
1.375	1.8906	2.5996	1.1726	1.1120	1.6123	1.0658	.72727272	4.3197	1.4849
1.5	2.25	3.3750	1.2247	1.1447	1.8371	1.0845	.66666666	4.7124	1.7671
1.625	2.6406	4.2910	1.2748	1.1757	2.0715	1.1020	.61538462	5.1051	2.0739
1.75	3.0625	5.3594	1.3229	1.2051	2.3150	1.1186	.57142857	5.4978	2.4053
1.875	3.5156	6.5918	1.3693	1.2331	2.5675	1.1340	.53333333	5.8905	2.7612
2.	**4.0000**	**8.0000**	**1.4142**	**1.2599**	**2.8284**	**1.1487**	**.50000000**	**6.2832**	**3.1416**
2.125	4.5156	9.5957	1.4577	1.2856	3.0977	1.1627	.47058823	6.6759	3.5466
2.25	5.0625	11.3906	1.5000	1.3104	3.3750	1.1761	.44444444	7.0686	3.9761
2.375	5.6406	13.3965	1.5411	1.3342	3.6601	1.1889	.42105263	7.4613	4.4301
2.5	6.2500	15.6250	1.5811	1.3572	3.9529	1.2011	.40000000	7.8540	4.9087
2.625	6.8906	18.0879	1.6202	1.3795	4.2530	1.2129	.38095231	8.2467	5.4119
2.75	7.5625	20.7969	1.6583	1.4011	4.5604	1.2242	.36363636	8.6394	5.9396
2.875	8.2656	23.7637	1.6956	1.4219	4.8748	1.2352	.34782609	9.0321	6.4918
3.	**9.0000**	**27.0000**	**1.7321**	**1.4422**	**5.1962**	**1.2457**	**.33333333**	**9.4248**	**7.0686**
3.125	9.7656	30.5176	1.7678	1.4620	5.5243	1.2559	.32000000	9.8175	7.6699
3.25	10.5625	34.3281	1.8028	1.4813	5.8590	1.2658	.30769231	10.2102	8.2958
3.375	11.3906	38.4434	1.8371	1.5000	6.2003	1.2754	.29629629	10.6029	8.9462
3.5	12.2500	42.8750	1.8708	1.5183	6.5479	1.2847	.28571429	10.9956	9.6211
3.625	13.1406	47.6348	1.9039	1.5362	6.9018	1.2938	.27586207	11.3883	10.3206
3.75	14.0625	52.7344	1.9365	1.5536	7.2619	1.3026	.26666666	11.7810	11.0447
3.875	15.0156	58.1856	1.9685	1.5707	7.6279	1.3112	.25806452	12.1737	11.7932
4.	**16.0000**	**64.0000**	**2.0000**	**1.5874**	**8.0000**	**1.3195**	**.25000000**	**12.5664**	**12.5664**
4.125	17.0156	70.1895	2.0310	1.6038	8.3779	1.3277	.24242424	12.9591	13.3640
4.25	18.0625	76.7656	2.0616	1.6198	8.7616	1.3356	.23529412	13.3518	14.1863
4.375	19.1406	83.7402	2.0916	1.6355	9.1510	1.3434	.22857143	13.7445	15.0330
4.5	20.2500	91.1250	2.1213	1.6510	9.5460	1.3510	.22222222	14.1372	15.9043
4.625	21.3906	98.9317	2.1506	1.6661	9.9465	1.3584	.21621622	14.5299	16.8001
4.75	22.5625	107.1719	2.1795	1.6810	10.3524	1.3656	.21052632	14.9226	17.7205
4.875	23.7656	115.8574	2.2079	1.6956	10.7637	1.3728	.20512821	15.3153	18.6655
5.	**25.0000**	**125.0000**	**2.2361**	**1.7100**	**11.1803**	**1.3799**	**.20000000**	**15.7080**	**19.6350**
5.125	26.2656	134.6113	2.2638	1.7241	11.6022	1.3866	.19512195	16.1006	20.6289
5.25	27.5625	144.7031	2.2913	1.7380	12.0293	1.3933	.19047619	16.4933	21.6475
5.375	28.8906	155.2871	2.3184	1.7517	12.4614	1.3998	.18604651	16.8860	22.6906
5.5	30.2500	166.3750	2.3452	1.7652	12.8987	1.4063	.18181818	17.2787	23.7583
5.625	31.6406	177.9785	2.3727	1.7784	13.3409	1.4126	.17777777	17.6714	24.8505
5.75	33.0625	190.1094	2.3979	1.7915	13.7880	1.4188	.17391304	18.0641	25.9672
5.875	34.5156	202.7793	2.4238	1.8044	14.2400	1.4250	.17021277	18.4568	27.1085
6.	**36.0000**	**216.0000**	**2.4495**	**1.8171**	**14.6969**	**1.4310**	**.16666666**	**18.8495**	**28.2743**
6.125	37.5156	229.7832	2.4749	1.8297	15.1586	1.4369	.16326531	19.2422	29.4647
6.25	39.0625	244.1406	2.5000	1.8420	15.6250	1.4427	.16000000	19.6349	30.6796
6.375	40.6406	259.0840	2.5249	1.8542	16.0961	1.4484	.15686275	20.0276	31.9190
6.5	42.2500	274.6250	2.5495	1.8663	16.5718	1.4542	.15384615	20.4203	33.1831
6.625	43.8906	290.7754	2.5739	1.8781	17.0522	1.4596	.15094339	20.8130	34.4716
6.75	45.5625	307.5469	2.5981	1.8899	17.5370	1.4651	.14814815	21.2057	35.7847
6.875	47.2656	324.9512	2.6220	1.9015	18.0264	1.4705	.14545454	21.5984	37.1223
7.	**49.0000**	**343.0000**	**2.6458**	**1.9129**	**18.5203**	**1.4758**	**.14285714**	**21.9911**	**38.4845**
7.125	50.7656	361.7051	2.6693	1.9243	19.0186	1.4810	.14035088	22.3838	39.8712
7.25	52.5625	381.0781	2.6926	1.9354	19.5212	1.4862	.13793103	22.7765	41.2825
7.375	54.3906	401.1309	2.7157	1.9465	20.0283	1.4913	.13559322	23.1692	42.7183
7.5	56.2500	421.8750	2.7386	1.9574	20.5396	1.4963	.13333333	23.5619	44.1786
7.625	58.1406	443.3223	2.7613	1.9683	21.0552	1.5012	.13114754	23.9546	45.6635
7.75	60.0625	465.4844	2.7839	1.9789	21.5751	1.5061	.12903226	24.3473	47.1730
7.875	62.0156	488.3731	2.8063	1.9895	22.0992	1.5110	.12698413	24.7400	48.7069
8.	**64.0000**	**512.0000**	**2.8284**	**2.0000**	**22.6274**	**1.5157**	**.12500000**	**25.1327**	**50.2655**
8.125	66.0156	536.3770	2.8504	2.0104	23.1598	1.5204	.12307692	25.5254	51.8485
8.25	68.0625	561.5156	2.8723	2.0206	23.6963	1.5251	.12121212	25.9181	53.4562
8.375	70.1406	587.4278	2.8940	2.0308	24.2369	1.5297	.11940298	26.3108	55.0883
8.5	72.2500	614.1250	2.9155	2.0408	24.7816	1.5342	.11764706	26.7035	56.7450
8.625	74.3906	641.6192	2.9368	2.0508	25.3301	1.5387	.11594203	27.0962	58.4262
8.75	76.5625	669.9219	2.9580	2.0606	25.8828	1.5431	.11428571	27.4889	60.1320
8.875	78.7656	699.0450	2.9791	2.0704	26.4394	1.5475	.11267605	27.8816	61.8623
9.	**81.0000**	**729.0000**	**3.0000**	**2.0801**	**27.0000**	**1.5518**	**.11111111**	**28.2743**	**63.6172**
9.125	83.2656	759.7989	3.0207	2.0897	27.5645	1.5561	.10958904	28.6670	65.3966
9.25	85.5625	791.4531	3.0414	2.0992	28.1328	1.5604	.10810811	29.0597	67.2006
9.375	87.8906	823.9746	3.0619	2.1086	28.7050	1.5646	.10666666	29.4524	69.0291
9.5	90.2500	857.3750	3.0822	2.1179	29.2810	1.5687	.10526316	29.8451	70.8822
9.625	92.6406	891.6660	3.1024	2.1272	29.8608	1.5728	.10389610	30.2378	72.7597
9.75	95.0625	926.8594	3.1225	2.1363	30.4444	1.5769	.10256410	30.6305	74.6619
9.875	97.5156	962.9668	3.1425	2.1454	31.0317	1.5809	.10126582	31.0232	76.5886

Table 4.—Properties of Numbers—*Continued*

N	N^2	N^3	$\sqrt{N}$	$\sqrt[3]{N}$	$N^{3/2}$	$\sqrt[5]{N}$	$\frac{1}{N}$	Circle (N = Diam.)	
								Circum.	Area
10	**100**	**1000**	**3.1623**	**2.1544**	**31.623**	**1.5849**	**.10000000**	**31.4159**	**78.5398**
11	121	1331	3.3166	2.2240	36.483	1.6154	.09090909	34.5575	95.0332
12	144	1728	3.4641	2.2894	41.569	1.6438	.08333333	37.6991	113.0973
13	169	2197	3.6056	2.3513	46.873	1.6703	.07692308	40.8407	132.7323
14	196	2744	3.7417	2.4101	52.384	1.6953	.07142857	43.9823	153.9380
15	225	3375	3.8730	2.4662	58.095	1.7188	.06666667	47.1239	176.7146
16	256	4096	4.0000	2.5198	64.000	1.7411	.06250000	50.2654	201.0619
17	289	4913	4.1231	2.5713	70.093	1.7623	.05882353	53.4070	226.9801
18	324	5832	4.2426	2.6207	76.367	1.7826	.05555556	56.5486	254.4690
19	361	6859	4.3589	2.6684	82.819	1.8020	.05263158	59.6902	283.5287
20	**400**	**8000**	**4.4721**	**2.7144**	**89.442**	**1.8206**	**.05000000**	**62.8318**	**314.1593**
21	441	9261	4.5826	2.7589	96.235	1.8384	.04761905	65.9734	346.3606
22	484	10648	4.6904	2.8020	103.19	1.8556	.04545455	69.1150	380.1327
23	529	12167	4.7958	2.8439	110.30	1.8722	.04347826	72.2566	415.4756
24	576	13824	4.8990	2.8845	117.58	1.8882	.04166667	75.3982	452.3893
25	625	15625	5.0000	2.9240	125.00	1.9037	.04000000	78.5398	490.8739
26	676	17576	5.0990	2.9625	132.57	1.9186	.03846154	81.6813	530.9292
27	729	19683	5.1962	3.0000	140.30	1.9332	.03703704	84.8229	572.5553
28	784	21952	5.2915	3.0366	148.16	1.9473	.03571429	87.9645	615.7522
29	841	24389	5.3852	3.0723	156.17	1.9610	.03448276	91.1061	660.5198
30	**900**	**27000**	**5.4772**	**3.1072**	**164.32**	**1.9744**	**.03333333**	**94.2477**	**706.8583**
31	961	29791	5.5678	3.1414	172.60	1.9873	.03225806	97.3893	754.7676
32	1024	32768	5.6569	3.1748	181.02	2.0000	.03125000	100.5309	804.2477
33	1089	35937	5.7446	3.2075	189.57	2.0123	.03030303	103.6725	855.2986
34	1156	39304	5.8310	3.2396	198.25	2.0244	.02941176	106.8141	907.9203
35	1225	42875	5.9161	3.2711	207.06	2.0362	.02857143	109.9557	962.1127
36	1296	46656	6.0000	3.3019	216.00	2.0477	.02777778	113.0972	1017.8760
37	1369	50653	6.0828	3.3322	225.06	2.0589	.02702703	116.2388	1075.2101
38	1444	54872	6.1644	3.3620	234.25	2.0699	.02631579	119.3804	1134.1149
39	1521	59319	6.2450	3.3912	243.56	2.0807	.02564103	122.5220	1194.5906
40	**1600**	**64000**	**6.3246**	**3.4200**	**252.98**	**2.0913**	**.02500000**	**125.6636**	**1256.6371**
41	1681	68921	6.4031	3.4482	262.53	2.1016	.02439024	128.8052	1320.2543
42	1764	74088	6.4807	3.4760	272.19	2.1118	.02380952	131.9468	1385.4424
43	1849	79507	6.5574	3.5034	281.97	2.1218	.02325581	135.0884	1452.2012
44	1936	85184	6.6332	3.5303	291.86	2.1315	.02272727	138.2300	1520.5308
45	2025	91125	6.7082	3.5569	301.87	2.1411	.02222222	141.3716	1590.4313
46	2116	97336	6.7823	3.5830	311.99	2.1506	.02173913	144.5131	1661.9025
47	2209	103823	6.8557	3.6088	322.22	2.1598	.02127660	147.6547	1734.9445
48	2304	110592	6.9282	3.6342	332.55	2.1689	.02083333	150.7963	1809.5574
49	2401	117649	7.0000	3.6593	343.00	2.1779	.02040816	153.9379	1885.7410
50	**2500**	**125000**	**7.0711**	**3.6840**	**353.55**	**2.1867**	**.02000000**	**157.0795**	**1963.500**
51	2601	132651	7.1414	3.7084	364.21	2.1954	.01960784	160.2211	2042.820
52	2704	140608	7.2111	3.7325	374.98	2.2039	.01923077	163.3627	2123.716
53	2809	148877	7.2801	3.7563	385.85	2.2124	.01886792	166.5043	2206.183
54	2916	157464	7.3485	3.7798	396.82	2.2206	.01851852	169.6459	2290.221
55	3025	166375	7.4162	3.8030	407.89	2.2288	.01818182	172.7875	2375.829
56	3136	175616	7.4833	3.8259	419.07	2.2369	.01785714	175.9290	2463.008
57	3249	185193	7.5498	3.8485	430.35	2.2448	.01754386	179.0706	2551.758
58	3364	195112	7.6158	3.8709	441.72	2.2526	.01724138	182.2122	2642.079
59	3481	205379	7.6811	3.8930	453.19	2.2603	.01694915	185.3538	2733.970
60	**3600**	**216000**	**7.7460**	**3.9149**	**464.76**	**2.2679**	**.01666667**	**188.4954**	**2827.433**
61	3721	226981	7.8102	3.9365	476.43	2.2755	.01639344	191.6370	2922.466
62	3844	238328	7.8740	3.9579	488.19	2.2829	.01612903	194.7786	3019.070
63	3969	250047	7.9373	3.9791	500.05	2.2902	.01587302	197.9202	3117.245
64	4096	262144	8.0000	4.0000	512.00	2.2974	.01562500	201.0618	3216.990
65	4225	274625	8.0623	4.0207	524.05	2.3045	.01538462	204.2034	3318.307
66	4356	287496	8.1240	4.0412	536.19	2.3116	.01515152	207.3449	3421.194
67	4489	300763	8.1854	4.0615	548.42	2.3186	.01492537	210.4865	3525.652
68	4624	314432	8.2462	4.0817	560.74	2.3254	.01470588	213.6281	3631.680
69	4761	328509	8.3066	4.1016	573.16	2.3322	.01449275	216.7697	3739.280
70	**4900**	**343000**	**8.3666**	**4.1213**	**585.66**	**2.3389**	**.01428571**	**219.9113**	**3848.450**
71	5041	357911	8.4261	4.1408	598.26	2.3456	.01408451	223.0529	3959.191
72	5184	373248	8.4853	4.1602	610.94	2.3522	.01388889	226.1945	4071.503
73	5329	389017	8.5440	4.1793	623.71	2.3587	.01369863	229.3361	4185.386
74	5476	405224	8.6023	4.1983	636.57	2.3651	.01351351	232.4777	4300.839
75	5625	421875	8.6603	4.2172	649.52	2.3714	.01333333	235.6193	4417.864
76	5776	438976	8.7178	4.2358	662.55	2.3777	.01315789	238.7608	4536.459
77	5929	456533	8.7750	4.2543	675.68	2.3840	.01298701	241.9024	4656.625
78	6084	474552	8.8318	4.2727	688.88	2.3901	.01282051	245.0440	4778.361
79	6241	493039	8.8882	4.2908	702.17	2.3962	.01265823	248.1856	4901.669

Table 4.—Properties of Numbers—*Continued*

N	N^2	N^3	$\sqrt{N}$	$\sqrt[3]{N}$	$N^{3/2}$	$\sqrt[5]{N}$	$\frac{1}{N}$	Circle (N = Diam.)	
								Circum.	Area
80	**6400**	**512000**	**8.9443**	**4.3089**	**715.54**	**2.4022**	**.01250000**	**251.327**	**5026.547**
81	6561	531441	9.0000	4.3267	729.00	2.4082	.01234568	254.469	5152.998
82	6724	551368	9.0554	4.3445	742.54	2.4141	.01219512	257.610	5281.016
83	6889	571787	9.1104	4.3621	756.17	2.4200	.01204819	260.752	5410.607
84	7056	592704	9.1652	4.3795	769.88	2.4258	.01190476	263.894	5541.770
85	7225	614125	9.2195	4.3968	783.66	2.4315	.01176471	267.035	5674.501
86	7396	636056	9.2736	4.4140	797.53	2.4372	.01162791	270.177	5808.805
87	7569	658503	9.3274	4.4310	811.49	2.4429	.01149425	273.318	5944.679
88	7744	681472	9.3808	4.4480	825.52	2.4485	.01136364	276.460	6082.124
89	7921	704969	9.4340	4.4647	839.63	2.4540	.01123596	279.602	6221.138
90	**8100**	**729000**	**9.4868**	**4.4814**	**853.82**	**2.4595**	**.01111111**	**282.743**	**6361.725**
91	8281	753571	9.5394	4.4979	868.09	2.4650	.01098901	285.885	6503.882
92	8464	778688	9.5917	4.5144	882.44	2.4705	.01086957	289.026	6647.610
93	8649	804357	9.6437	4.5307	896.86	2.4758	.01075269	292.168	6792.909
94	8836	830584	9.6954	4.5468	911.36	2.4810	.01063830	295.309	6939.778
95	9025	857375	9.7468	4.5629	925.95	2.4863	.01052632	298.451	7088.219
96	9216	884736	9.7980	4.5789	940.61	2.4915	.01041667	301.593	7238.230
97	9409	912673	9.8489	4.5947	955.34	2.4966	.01030928	304.734	7389.812
98	9604	941192	9.8995	4.6104	970.15	2.5018	.01020408	307.876	7542.962
99	9801	970299	9.9499	4.6261	985.04	2.5069	.01010101	311.017	7697.688
100	**10000**	**1000000**	**10.0000**	**4.6416**	**1000.0**	**2.5119**	**.01000000**	**314.159**	**7853.982**
101	10201	1030301	10.0499	4.6570	1015.0	2.5169	.00990099	317.301	8011.85
102	10404	1061208	10.0995	4.6723	1030.1	2.5219	.00980392	320.442	8171.28
103	10609	1092727	10.1489	4.6875	1045.3	2.5268	.00970874	323.584	8332.29
104	10816	1124864	10.1980	4.7027	1060.6	2.5317	.00961538	326.725	8494.87
105	11025	1157625	10.2470	4.7177	1075.9	2.5365	.00952381	329.867	8659.01
106	11236	1191016	10.2956	4.7326	1091.3	2.5413	.00943396	333.009	8824.73
107	11449	1225043	10.3441	4.7475	1106.8	2.5461	.00934579	336.150	8992.02
108	11664	1259712	10.3923	4.7622	1122.4	2.5509	.00925926	339.292	9160.88
109	11881	1295029	10.4403	4.7769	1138.0	2.5556	.00917431	342.433	9331.32
110	**12100**	**1331000**	**10.4881**	**4.7914**	**1153.7**	**2.5602**	**.00909091**	**345.575**	**9503.32**
111	12321	1367631	10.5357	4.8059	1169.5	2.5649	.00900901	348.716	9676.89
112	12544	1404928	10.5830	4.8203	1185.3	2.5695	.00892857	351.858	9852.03
113	12769	1442897	10.6301	4.8346	1201.2	2.5740	.00884956	355.000	10028.75
114	12996	1481544	10.6771	4.8488	1217.2	2.5786	.00877193	358.141	10207.03
115	13225	1520875	10.7238	4.8629	1233.2	2.5831	.00869565	361.283	10386.89
116	13456	1560896	10.7703	4.8770	1249.4	2.5876	.00862069	364.424	10568.32
117	13689	1601613	10.8167	4.8910	1265.5	2.5920	.00854701	367.566	10751.31
118	13924	1643032	10.8628	4.9049	1281.8	2.5964	.00847458	370.708	10935.88
119	14161	1685159	10.9087	4.9187	1298.1	2.6008	.00840336	373.849	11122.02
120	**14400**	**1728000**	**10.9545**	**4.9324**	**1314.5**	**2.6052**	**.00833333**	**376.991**	**11309.73**
121	14641	1771561	11.0000	4.9461	1331.0	2.6095	.00826446	380.132	11499.01
122	14884	1815848	11.0454	4.9597	1347.5	2.6138	.00819672	383.274	11689.86
123	15129	1860867	11.0905	4.9732	1364.1	2.6181	.00813008	386.416	11882.29
124	15376	1906624	11.1355	4.9866	1380.8	2.6223	.00806452	389.557	12076.28
125	15625	1953125	11.1803	5.0000	1397.5	2.6265	.00800000	392.699	12271.84
126	15876	2000376	11.2250	5.0133	1414.4	2.6307	.00793651	395.840	12468.98
127	16129	2048383	11.2694	5.0265	1431.2	2.6349	.00787402	398.982	12667.68
128	16384	2097152	11.3137	5.0397	1448.2	2.6390	.00781250	402.124	12867.96
129	16641	2146689	11.3578	5.0528	1465.2	2.6431	.00775194	405.265	13069.81
130	**16900**	**2197000**	**11.4018**	**5.0658**	**1482.2**	**2.6472**	**.00769231**	**408.407**	**13273.23**
131	17161	2248091	11.4455	5.0788	1499.4	2.6513	.00763359	411.548	13478.22
132	17424	2299968	11.4891	5.0916	1516.6	2.6553	.00757576	414.690	13684.77
133	17689	2352637	11.5326	5.1045	1533.8	2.6593	.00751880	417.831	13892.91
134	17956	2406104	11.5758	5.1172	1551.2	2.6633	.00746269	420.973	14102.61
135	18225	2460375	11.6190	5.1299	1568.6	2.6673	.00740741	424.115	14313.88
136	18496	2515456	11.6619	5.1426	1586.0	2.6712	.00735294	427.256	14526.72
137	18769	2571353	11.7047	5.1551	1603.6	2.6751	.00729927	430.398	14741.14
138	19044	2628072	11.7473	5.1676	1621.1	2.6790	.00724638	433.539	14957.12
139	19321	2685619	11.7898	5.1801	1638.8	2.6829	.00719424	436.681	15174.67
140	**19600**	**2744000**	**11.8322**	**5.1925**	**1656.5**	**2.6867**	**.00714286**	**439.823**	**15393.80**
141	19881	2803221	11.8743	5.2048	1674.3	2.6906	.00709220	442.964	15614.50
142	20164	2863288	11.9164	5.2171	1692.1	2.6944	.00704225	446.106	15836.77
143	20449	2924207	11.9583	5.2293	1710.0	2.6981	.00699301	449.247	16060.60
144	20736	2985984	12.0000	5.2415	1728.0	2.7019	.00694444	452.389	16286.01
145	21025	3048625	12.0416	5.2536	1746.0	2.7057	.00689655	455.531	16512.99
146	21316	3112136	12.0830	5.2656	1764.1	2.7094	.00684932	458.672	16741.54
147	21609	3176523	12.1244	5.2776	1782.2	2.7131	.00680272	461.814	16971.67
148	21904	3241792	12.1655	5.2896	1800.5	2.7168	.00675676	464.955	17203.36
149	22201	3307949	12.2066	5.3015	1818.8	2.7204	.00671141	468.097	17436.62

Table 4.—Properties of Numbers—*Continued*

N	N^2	N^3	$\sqrt{N}$	$\sqrt[3]{N}$	$N^{3/2}$	$\sqrt[5]{N}$	$\frac{1}{N}$	Circle (N = Diam.)	
								Circum.	Area
150	**22500**	**3375000**	**12.2474**	**5.3133**	**1837.1**	**2.7241**	**.00666667**	**471.239**	**17671.46**
151	22801	3442951	12.2882	5.3251	1855.5	2.7277	.00662252	474.380	17907.86
152	23104	3511808	12.3288	5.3368	1874.0	2.7314	.00657895	477.522	18145.84
153	23409	3581577	12.3693	5.3485	1892.5	2.7349	.00653595	480.663	18385.38
154	23716	3652264	12.4097	5.3601	1911.1	2.7385	.00649351	483.805	18626.50
155	24025	3723875	12.4499	5.3717	1929.7	2.7420	.00645161	486.946	18869.19
156	24336	3796416	12.4900	5.3832	1948.4	2.7455	.00641026	490.088	19113.45
157	24649	3869893	12.5300	5.3947	1967.2	2.7490	.00636943	493.230	19359.28
158	24964	3944312	12.5698	5.4061	1986.0	2.7525	.00632911	496.371	19606.68
159	25281	4019679	12.6095	5.4175	2004.9	2.7560	.00628931	499.513	19855.65
160	**25600**	**4096000**	**12.6491**	**5.4288**	**2023.9**	**2.7595**	**.00625000**	**502.654**	**20106.19**
161	25921	4173281	12.6886	5.4401	2042.9	2.7629	.00621118	505.796	20358.30
162	26244	4251528	12.7279	5.4514	2061.9	2.7663	.00617284	508.938	20611.99
163	26569	4330747	12.7671	5.4626	2081.0	2.7697	.00613497	512.079	20867.24
164	26896	4410944	12.8062	5.4737	2100.2	2.7731	.00609756	515.221	21124.06
165	27225	4492125	12.8452	5.4848	2119.5	2.7765	.00606061	518.362	21382.46
166	27556	4574296	12.8841	5.4959	2138.8	2.7799	.00602410	521.504	21642.43
167	27889	4657463	12.9228	5.5069	2158.1	2.7832	.00598802	524.646	21903.96
168	28224	4741632	12.9615	5.5178	2177.5	2.7865	.00595238	527.787	22167.07
169	28561	4826809	13.0000	5.5288	2197.0	2.7898	.00591716	530.929	22431.75
170	**28900**	**4913000**	**13.0384**	**5.5397**	**2216.5**	**2.7931**	**.00588235**	**534.070**	**22698.00**
171	29241	5000211	13.0767	5.5505	2236.1	2.7964	.00584795	537.212	22965.82
172	29584	5088448	13.1149	5.5613	2255.8	2.7997	.00581395	540.353	23235.21
173	29929	5177717	13.1529	5.5721	2275.5	2.8029	.00578035	543.495	23506.18
174	30276	5268024	13.1909	5.5828	2295.2	2.8061	.00574713	546.637	23778.71
175	30625	5359375	13.2288	5.5934	2315.0	2.8094	.00571429	549.778	24052.81
176	30976	5451776	13.2665	5.6041	2334.9	2.8126	.00568182	552.920	24328.49
177	31329	5545233	13.3041	5.6147	2354.8	2.8158	.00564972	556.061	24605.73
178	31684	5639752	13.3417	5.6252	2374.8	2.8189	.00561798	559.203	24884.55
179	32041	5735339	13.3791	5.6357	2394.9	2.8221	.00558659	562.345	25164.94
180	**32400**	**5832000**	**13.4164**	**5.6462**	**2415.0**	**2.8252**	**.00555556**	**565.486**	**25446.90**
181	32761	5929741	13.4536	5.6567	2435.1	2.8284	.00552486	568.628	25730.42
182	33124	6028568	13.4907	5.6671	2455.3	2.8315	.00549451	571.769	26015.52
183	33489	6128487	13.5277	5.6774	2475.6	2.8346	.00546448	574.911	26302.19
184	33856	6229504	13.5647	5.6877	2495.9	2.8377	.00543478	578.053	26590.43
185	34225	6331625	13.6015	5.6980	2516.3	2.8408	.00540541	581.194	26880.25
186	34596	6434856	13.6382	5.7083	2536.7	2.8438	.00537634	584.336	27171.63
187	34969	6539203	13.6748	5.7185	2557.2	2.8469	.00534759	587.477	27464.58
188	35344	6644672	13.7113	5.7287	2577.7	2.8499	.00531915	590.619	27759.11
189	35721	6751269	13.7477	5.7388	2598.3	2.8529	.00529101	593.761	28055.20
190	**36100**	**6859000**	**13.7840**	**5.7489**	**2619.0**	**2.8560**	**.00526316**	**596.902**	**28352.87**
191	36481	6967871	13.8203	5.7590	2639.7	2.8590	.00523560	600.044	28652.10
192	36864	7077888	13.8564	5.7690	2660.4	2.8619	.00520833	603.185	28952.91
193	37249	7189057	13.8924	5.7790	2681.2	2.8649	.00518135	606.327	29255.29
194	37636	7301384	13.9284	5.7890	2702.1	2.8679	.00515464	609.468	29559.24
195	38025	7414875	13.9642	5.7989	2723.0	2.8708	.00512821	612.610	29864.76
196	38416	7529536	14.0000	5.8088	2744.0	2.8738	.00510204	615.752	30171.85
197	38809	7645373	14.0357	5.8186	2765.0	2.8767	.00507614	618.893	30480.51
198	39204	7762392	14.0712	5.8285	2786.1	2.8796	.00505051	622.035	30790.74
199	39601	7880599	14.1067	5.8383	2807.2	2.8825	.00502513	625.176	31102.55
200	**40000**	**8000000**	**14.1421**	**5.8480**	**2828.4**	**2.8854**	**.00500000**	**628.318**	**31415.93**
201	40401	8120601	14.1774	5.8578	2849.7	2.8883	.00497512	631.460	31730.87
202	40804	8242408	14.2127	5.8675	2871.0	2.8911	.00495050	634.601	32047.39
203	41209	8365427	14.2478	5.8771	2892.3	2.8940	.00492611	637.743	32365.47
204	41616	8489664	14.2829	5.8868	2913.7	2.8968	.00490196	640.884	32685.13
205	42025	8615125	14.3178	5.8964	2935.2	2.8997	.00487805	644.026	33006.36
206	42436	8741816	14.3527	5.9059	2956.7	2.9025	.00485437	647.168	33329.16
207	42849	8869743	14.3875	5.9155	2978.2	2.9053	.00483092	650.309	33653.53
208	43264	8998912	14.4222	5.9250	2999.8	2.9081	.00480769	653.451	33979.47
209	43681	9129329	14.4568	5.9345	3021.5	2.9109	.00478469	656.592	34306.98
210	**44100**	**9261000**	**14.4914**	**5.9439**	**3043.2**	**2.9137**	**.00476190**	**659.734**	**34636.06**
211	44521	9393931	14.5258	5.9533	3065.0	2.9165	.00473934	662.875	34966.71
212	44944	9528128	14.5602	5.9627	3086.8	2.9192	.00471698	666.017	35298.94
213	45369	9663597	14.5945	5.9721	3108.7	2.9220	.00469484	669.159	35632.73
214	45796	9800344	14.6287	5.9814	3130.6	2.9247	.00467290	672.300	35968.09
215	46225	9938375	14.6629	5.9907	3152.5	2.9274	.00465116	675.442	36305.03
216	46656	10077696	14.6969	6.0000	3174.5	2.9302	.00462963	678.583	36643.54
217	47089	10218313	14.7309	6.0092	3196.6	2.9329	.00460829	681.725	36983.61
218	47524	10360232	14.7648	6.0185	3218.7	2.9356	.00458716	684.867	37325.26
219	47961	10503459	14.7986	6.0277	3240.9	2.9383	.00456621	688.008	37668.48

Table 4.—Properties of Numbers—*Continued*

							$\frac{1}{N}$	Circle (N = Diam.) Circum.	Area
220	**48400**	**10648000**	**14.8324**	**6.0368**	**3263.1**	**2.9409**	**00454545**	**691.150**	**38013.27**
221	48841	10793861	14.8661	6.0459	3285.4	2.9436	.00452489	694.291	38359.63
222	49284	10941048	14.8997	6.0550	3307.7	2.9463	.00450450	697.433	38707.56
223	49729	11089567	14.9332	6.0641	3330.1	2.9489	.00448430	700.575	39057.07
224	50176	11239424	14.9666	6.0732	3352.5	2.9516	.00446429	703.716	39408.14
225	50625	11390625	15.0000	6.0822	3375.0	2.9542	.00444444	706.858	39760.78
226	51076	11543176	15.0333	6.0912	3397.5	2.9568	.00442478	709.999	40115.00
227	51529	11697083	15.0665	6.1002	3420.1	2.9594	.00440529	713.141	40470.78
228	51984	11852352	15.0997	6.1091	3442.7	2.9620	.00438596	716.283	40828.14
229	52441	12008989	15.1327	6.1180	3465.4	2.9646	.00436681	719.424	41187.07
230	**52900**	**12167000**	**15.1658**	**6.1269**	**3488.1**	**2.9672**	**.00434783**	**722.566**	**41547.56**
231	53361	12326391	15.1987	6.1358	3510.9	2.9698	.00432900	725.707	41909.63
232	53824	12487168	15.2315	6.1446	3533.7	2.9723	.00431034	728.849	42273.27
233	54289	12649337	15.2643	6.1534	3556.6	2.9749	.00429185	731.990	42638.48
234	54756	12812904	15.2971	6.1622	3579.5	2.9774	.00427350	735.132	43005.26
235	55225	12977875	15.3297	6.1710	3602.5	2.9800	.00425532	738.274	43373.61
236	55696	13144256	15.3623	6.1797	3625.5	2.9825	.00423729	741.415	43743.54
237	56169	13312053	15.3948	6.1885	3648.6	2.9850	.00421941	744.557	44115.03
238	56644	13481272	15.4272	6.1972	3671.7	2.9875	.00420168	747.698	44488.09
239	57121	13651919	15.4596	6.2058	3694.8	9900	.00418410	750.840	44862.73
240	**57600**	**13824000**	**15.4919**	**6.2145**	**3718.0**	**2.9925**	**.00416667**	**753.982**	**45238.93**
241	58081	13997521	15.5242	6.2231	3741.3	2.9950	.00414938	757.123	45616.71
242	58564	14172488	15.5563	6.2317	3764.6	2.9975	.00413223	760.265	45996.06
243	59049	14348907	15.5885	6.2403	3788.0	3.0000	.00411523	763.406	46376.98
244	59536	14526784	15.6205	6.2488	3811.4	3.0025	.00409836	766.548	46759.47
245	60025	14706125	15.6525	6.2573	3834.9	3.0049	.00408163	769.690	47143.52
246	60516	14886936	15.6844	6.2658	3858.4	3.0074	.00406504	772.831	47529.16
247	61009	15069223	15.7162	6.2743	3881.9	3.0098	.00404858	775.973	47916.36
248	61504	15252992	15.7480	6.2828	3905.5	3.0122	.00403226	779.114	48305.13
249	62001	15438249	15.7797	6.2912	3929.2	3.0147	00401606	782.256	48695.47
250	**62500**	**15625000**	**15.8114**	**6.2996**	**3952.9**	**3.0171**	**00400000**	**785.398**	**49087.39**
251	63001	15813251	15.8430	6.3080	3976.6	3.0195	.00398406	788.539	49480.87
252	63504	16003008	15.8745	6.3164	4000.4	3.0219	.00396825	791.681	49875.92
253	64009	16194277	15.9060	6.3247	4024.2	3.0243	.00395257	794.822	50272.55
254	64516	16387064	15.9374	6.3330	4048.1	3.0267	.00393701	797.964	50670.75
255	65025	16581375	15.9687	6.3413	4072.0	3.0291	.00392157	801.105	51070.52
256	65536	16777216	16.0000	6.3496	4096.0	3.0314	.00390625	804.247	51471.85
257	66049	16974593	16.0312	6.3579	4120.0	3.0338	.00389105	807.389	51874.76
258	66564	17173512	16.0624	6.3661	4144.1	3.0362	00387597	810.530	52279.24
259	67081	17373979	16.0935	6.3743	4168.2	3.0385	00386100	813.672	52685.29
260	**67600**	**17576000**	**16.1245**	**6.3825**	**4192.4**	**3.0418**	**00384615**	**816.813**	**53092.92**
261	68121	17779581	16.1555	6.3907	4216.6	3.0432	00383142	819.955	53502.11
262	68644	17984728	16.1864	6.3988	4240.8	3.0455	.00381679	823.097	53912.87
263	69169	18191447	16.2173	6.4070	4265.1	3.0478	.00380228	826.238	54325.21
264	69696	18399744	16.2481	6.4151	4289.5	3.0501	.00378788	829.380	54739.11
265	70225	18609625	16.2788	6.4232	4313.9	3.0524	.00377358	832.521	55154.59
266	70756	18821096	16.3095	6.4312	4338.3	3.0547	.00375940	835.663	55571.63
267	71289	19034163	16.3401	6.4393	4362.8	3.0570	00374532	838.805	55990.25
268	71824	19248832	16.3707	6.4473	4387.3	3.0593	00373134	841.946	56410.44
269	72361	19465109	16.4012	6.4553	4411.9	3.0616	00371747	845.088	56832.20
270	**72900**	**19683000**	**16.4317**	**6.4633**	**4436.5**	**3.0639**	**.00370370**	**848.229**	**57255.53**
271	73441	19902511	16.4621	6.4713	4461.2	3.0662	00369004	851.371	57680.43
272	73984	20123648	16.4924	6.4792	4485.9	3.0684	.00367647	854.512	58106.90
273	74529	20346417	16.5227	6.4872	4510.7	3.0707	.00366300	857.654	58534.94
274	75076	20570824	16.5529	6.4951	4535.5	3.0729	00364964	860.796	58964.55
275	75625	20796875	16.5831	6.5030	4560.4	3.0752	.00363636	863.937	59395.74
276	76176	21024576	16.6132	6.5108	4585.3	3.0774	.00362319	867.079	59828.49
277	76729	21253933	16.6433	6.5187	4610.2	3.0796	00361011	870.220	60262.82
278	77284	21484952	16.6733	6.5265	4635.2	3.0818	.00359712	873.362	60698.71
279	77841	21717639	16.7033	6.5343	4660.2	3.0840	.00358423	876.504	61136.18
280	**78400**	**21952000**	**16.7332**	**6.5421**	**4685.3**	**3.0863**	**.00357143**	**879.645**	**61575.22**
281	78961	22188041	16.7631	6.5499	4710.4	3.0885	.00355872	882.787	62015.82
282	79524	22425768	16.7929	6.5577	4735.6	3.0907	.00354610	885.928	62458.00
283	80089	22665187	16.8226	6.5654	4760.8	3.0928	.00353357	889.070	62901.75
284	80656	22906304	16.8523	6.5731	4786.0	3.0950	.00352113	892.212	63347.07
285	81225	23149125	16.8819	6.5808	4811.3	3.0972	.00350877	895.353	63793.97
286	81796	23393656	16.9115	6.5885	4836.7	3.0994	.00349650	898.495	64242.43
287	82369	23639903	16.9411	6.5962	4862.1	3.1015	.00348432	901.636	64692.46
288	82944	23887872	16.9706	6.6039	4887.5	3.1037	.00347222	904.778	65144.07
289	83521	24137569	17.0000	6.6115	4913.0	3.1058	.00346021	907.920	65597.24

Table 4.—Properties of Numbers—*Continued*

N	N^2	N^3	$\sqrt{N}$	$\sqrt[3]{N}$	$N^{3/2}$	$\sqrt[5]{N}$	$\frac{1}{N}$	Circle (N = Diam.)	
								Circum.	Area
290	**84100**	**24389000**	**17.0294**	**6.6191**	**4938.5**	**3.1080**	**.00344828**	**911.061**	**66051.99**
291	84681	24642171	17.0587	6.6267	4964.1	3.1101	.00343643	914.203	66508.30
292	85264	24897088	17.0880	6.6343	4989.7	3.1123	.00342466	917.344	66966.19
293	85849	25153757	17.1172	6.6419	5015.4	3.1144	.00341297	920.486	67425.65
294	86436	25412184	17.1464	6.6494	5041.1	3.1165	.00340136	923.627	67886.68
295	87025	25672375	17.1756	6.6569	5066.8	3.1186	.00338983	926.769	68349.28
296	87616	25934336	17.2047	6.6644	5092.6	3.1207	.00337838	929.911	68813.45
297	88209	26198073	17.2337	6.6719	5118.4	3.1228	.00336700	933.052	69279.19
298	88804	26463592	17.2627	6.6794	5144.3	3.1249	.00335570	936.194	69746.50
299	89401	26730899	17.2916	6.6869	5170.2	3.1270	.00334448	939.335	70215.38
300	**90000**	**27000000**	**17.3205**	**6.6943**	**5196.2**	**3.1291**	**.00333333**	**942.477**	**70685.83**
301	90601	27270901	17.3494	6.7018	5222.2	3.1312	.00332226	945.619	71157.86
302	91204	27543608	17.3781	6.7092	5248.2	3.1333	.00331126	948.760	71631.45
303	91809	27818127	17.4069	6.7166	5274.3	3.1354	.00330033	951.902	72106.62
304	92416	28094464	17.4356	6.7240	5300.4	3.1374	.00328947	955.043	72583.36
305	93025	28372625	17.4642	6.7313	5326.6	3.1395	.00327869	958.185	73061.66
306	93636	28652616	17.4929	6.7387	5352.8	3.1416	.00326797	961.327	73541.54
307	94249	28934443	17.5214	6.7460	5379.1	3.1436	.00325733	964.468	74022.99
308	94864	29218112	17.5499	6.7533	5405.4	3.1456	.00324675	967.610	74506.01
309	95481	29503629	17.5784	6.7606	5431.7	3.1477	.00323625	970.751	74990.60
310	**96100**	**29791000**	**17.6068**	**6.7679**	**5458.1**	**3.1497**	**.00322581**	**973.893**	**75476.76**
311	96721	30080231	17.6352	6.7752	5484.5	3.1518	.00321543	977.034	75964.50
312	97344	30371328	17.6635	6.7824	5511.0	3.1538	.00320513	980.176	76453.80
313	97969	30664297	17.6918	6.7897	5537.5	3.1558	.00319489	983.318	76944.67
314	98596	30959144	17.7200	6.7969	5564.1	3.1578	.00318471	986.459	77437.12
315	99225	31255875	17.7482	6.8041	5590.7	3.1598	.00317460	989.601	77931.13
316	99856	31554496	17.7764	6.8113	5617.3	3.1618	.00316456	992.742	78426.72
317	100489	31855013	17.8045	6.8185	5644.0	3.1638	.00315457	995.884	78923.88
318	101124	32157432	17.8326	6.8256	5670.7	3.1658	.00314465	999.026	79422.60
319	101761	32461759	17.8606	6.8328	5697.5	3.1678	.00313480	1002.167	79922.90
320	**102400**	**32768000**	**17.8885**	**6.8399**	**5724.3**	**3.1698**	**.00312500**	**1005.309**	**80424.77**
321	103041	33076161	17.9165	6.8470	5751.2	3.1718	.00311526	1008.450	80928.21
322	103684	33386248	17.9444	6.8541	5778.1	3.1737	.00310559	1011.592	81433.22
323	104329	33698267	17.9722	6.8612	5805.0	3.1757	.00309598	1014.734	81939.80
324	104976	34012224	18.0000	6.8683	5832.0	3.1777	.00308642	1017.875	82447.96
325	105625	34328125	18.0278	6.8753	5859.0	3.1796	.00307692	1021.017	82957.68
326	106276	34645976	18.0555	6.8824	5886.1	3.1816	.00306748	1024.158	83468.97
327	106929	34965783	18.0831	6.8894	5913.2	3.1835	.00305810	1027.300	83981.84
328	107584	35287552	18.1108	6.8964	5940.3	3.1855	.00304878	1030.442	84496.28
329	108241	35611289	18.1384	6.9034	5967.5	3.1874	.00303951	1033.583	85012.28
330	**108900**	**35937000**	**18.1659**	**6.9104**	**5994.7**	**3.1894**	**.00303030**	**1036.725**	**85529.86**
331	109561	36264691	18.1934	6.9174	6022.0	3.1913	.00302115	1039.866	86049.01
332	110224	36594368	18.2209	6.9244	6049.3	3.1932	.00301205	1043.008	86569.73
333	110889	36926037	18.2483	6.9313	6076.7	3.1951	.00300300	1046.149	87092.02
334	111556	37259704	18.2757	6.9382	6104.1	3.1970	.00299401	1049.291	87615.88
335	112225	37595375	18.3030	6.9451	6131.5	3.1989	.00298507	1052.433	88141.31
336	112896	37933056	18.3303	6.9521	6159.0	3.2009	.00297619	1055.574	88668.31
337	113569	38272753	18.3576	6.9589	6186.5	3.2028	.00296736	1058.716	89196.88
338	114244	38614472	18.3848	6.9658	6214.1	3.2047	.00295858	1061.857	89727.03
339	114921	38958219	18.4120	6.9727	6241.7	3.2066	.00294985	1064.999	90258.74
340	**115600**	**39304000**	**18.4391**	**6.9795**	**6269.3**	**3.2085**	**.00294118**	**1068.141**	**90792.03**
341	116281	39651821	18.4662	6.9864	6297.0	3.2103	.00293255	1071.282	91326.88
342	116964	40001688	18.4932	6.9932	6324.7	3.2122	.00292398	1074.424	91863.31
343	117649	40353607	18.5203	7.0000	6352.4	3.2141	.00291545	1077.565	92401.31
344	118336	40707584	18.5472	7.0068	6380.2	3.2160	.00290698	1080.707	92940.88
345	119025	41063625	18.5742	7.0136	6408.1	3.2178	.00289855	1083.849	93482.02
346	119716	41421736	18.6011	7.0203	6436.0	3.2197	.00289017	1086.990	94024.73
347	120409	41781923	18.6279	7.0271	6463.9	3.2216	.00288184	1090.132	94569.01
348	121104	42144192	18.6548	7.0338	6491.9	3.2234	.00287356	1093.273	95114.86
349	121801	42508549	18.6815	7.0406	6519.9	3.2253	.00286533	1096.415	95662.28
350	**122500**	**42875000**	**18.7083**	**7.0473**	**6547.9**	**3.2271**	**.00285714**	**1099.557**	**96211.28**
351	123201	43243551	18.7350	7.0540	6576.0	3.2289	.00284900	1102.698	96761.84
352	123904	43614208	18.7617	7.0607	6604.1	3.2308	.00284091	1105.840	97313.97
353	124609	43986977	18.7883	7.0674	6632.3	3.2326	.00283286	1108.981	97867.68
354	125316	44361864	18.8149	7.0740	6660.5	3.2345	.00282486	1112.123	98422.96
355	126025	44738875	18.8414	7.0807	6688.7	3.2363	.00281690	1115.264	98979.80
356	126736	45118016	18.8680	7.0873	6717.0	3.2381	.00280899	1118.406	99538.22
357	127449	45499293	18.8944	7.0940	6745.3	3.2399	.00280112	1121.548	100098.21
358	128164	45882712	18.9209	7.1006	6773.7	3.2417	.00279330	1124.689	100659.77
359	128881	46268279	18.9473	7.1072	6802.1	3.2435	.00278552	1127.831	101222.90

Table 4.—Properties of Numbers—*Continued*

N	N^2	N^3	$\sqrt{N}$	$\sqrt[3]{N}$	$N^{3/2}$	$\sqrt[5]{N}$	$\frac{1}{N}$	(Circle N = Diam.) Circum.	Area
360	**129600**	**46656000**	**18.9737**	**7.1138**	**6830.5**	**3.2453**	**.00277778**	**1130.972**	**101787.60**
361	130321	47045881	19.0000	7.1204	6859.0	3.2471	.00277008	1134.114	102353.87
362	131044	47437928	19.0263	7.1269	6887.5	3.2489	.00276243	1137.256	102921.72
363	131769	47832147	19.0526	7.1335	6916.1	3.2507	.00275482	1140.397	103491.13
364	132496	48228544	19.0788	7.1400	6944.7	3.2525	.00274725	1143.539	104062.12
365	133225	48627125	19.1050	7.1466	6973.3	3.2543	.00273973	1146.680	104634.67
366	133956	49027896	19.1311	7.1531	7002.0	3.2561	.00273224	1149.822	105208.80
367	134689	49430863	19.1572	7.1596	7030.7	3.2579	.00272480	1152.964	105784.49
368	135424	49836032	19.1833	7.1661	7059.5	3.2597	.00271739	1156.105	106361.76
369	136161	50243409	19.2094	7.1726	7088.3	3.2614	.00271003	1159.247	106940.60
370	**136900**	**50653000**	**19.2354**	**7.1791**	**7117.1**	**3.2632**	**.00270270**	**1162.388**	**107521.01**
371	137641	51064811	19.2614	7.1855	7146.0	3.2650	.00269542	1165.530	108102.99
372	138384	51478848	19.2873	7.1920	7174.9	3.2668	.00268817	1168.671	108686.54
373	139129	51895117	19.3132	7.1984	7203.9	3.2685	.00268097	1171.813	109271.66
374	139876	52313624	19.3391	7.2048	7232.8	3.2702	.00267380	1174.955	109858.35
375	140625	52734375	19.3649	7.2112	7261.8	3.2719	.00266667	1178.096	110446.62
376	141376	53157376	19.3907	7.2177	7290.9	3.2737	.00265957	1181.238	111036.45
377	142129	53582633	19.4165	7.2240	7320.0	3.2754	.00265252	1184.379	111627.86
378	142884	54010152	19.4422	7.2304	7349.2	3.2772	.00264550	1187.521	112220.83
379	143641	54439939	19.4679	7.2368	7378.4	3.2789	.00263852	1190.663	112815.38
380	**144400**	**54872000**	**19.4936**	**7.2432**	**7407.6**	**3.2807**	**.00263158**	**1193.804**	**113411.49**
381	145161	55306341	19.5192	7.2495	7436.8	3.2824	.00262467	1196.946	114009.18
382	145924	55742968	19.5448	7.2558	7466.1	3.2841	.00261780	1200.087	114608.44
383	146689	56181887	19.5704	7.2622	7495.4	3.2858	.00261097	1203.229	115209.27
384	147456	56623104	19.5959	7.2685	7524.8	3.2875	.00260417	1206.371	115811.67
385	148225	57066625	19.6214	7.2748	7554.2	3.2892	.00259740	1209.512	116415.64
386	148996	57512456	19.6469	7.2811	7583.7	3.2909	.00259067	1212.654	117021.18
387	149769	57960603	19.6723	7.2874	7613.2	3.2926	.00258398	1215.795	117628.30
388	150544	58411072	19.6977	7.2936	7642.7	3.2943	.00257732	1218.937	118236.98
389	151321	58863869	19.7231	7.2999	7672.3	3.2960	.00257069	1222.079	118847.24
390	**152100**	**59319000**	**19.7484**	**7.3061**	**7701.9**	**3.2977**	**.00256410**	**1225.220**	**119459.06**
391	152881	59776471	19.7737	7.3124	7731.5	3.2994	.00255754	1228.362	120072.46
392	153664	60236288	19.7990	7.3186	7761.2	3.3011	.00255102	1231.503	120687.42
393	154449	60698457	19.8242	7.3248	7790.9	3.3028	.00254453	1234.645	121303.96
394	155236	61162984	19.8494	7.3310	7820.7	3.3045	.00253807	1237.786	121922.07
395	156025	61629875	19.8746	7.3372	7850.5	3.3061	.00253165	1240.928	122541.75
396	156816	62099136	19.8997	7.3434	7880.3	3.3078	.00252525	1244.070	123163.00
397	157609	62570773	19.9249	7.3496	7910.2	3.3095	.00251889	1247.211	123785.82
398	158404	63044792	19.9499	7.3558	7940.1	3.3111	.00251256	1250.353	124410.21
399	159201	63521199	19.9750	7.3619	7970.0	3.3128	.00250627	1253.494	125036.17
400	**160000**	**64000000**	**20.0000**	**7.3681**	**8000.0**	**3.3145**	**.00250000**	**1256.636**	**125663.71**
401	160801	64481201	20.0250	7.3742	8030.0	3.3161	.00249377	1259.778	126292.81
402	161604	64964808	20.0499	7.3803	8061.1	3.3178	.00248756	1262.919	126923.48
403	162409	65450827	20.0749	7.3864	8090.2	3.3194	.00248139	1266.061	127555.73
404	163216	65939264	20.0998	7.3925	8120.3	3.3211	.00247525	1269.202	128189.55
405	164025	66430125	20.1246	7.3986	8150.5	3.3227	.00246914	1272.344	128824.93
406	164836	66923416	20.1494	7.4047	8180.7	3.3243	.00246305	1275.486	129461.89
407	165649	67419143	20.1742	7.4108	8210.9	3.3260	.00245700	1278.627	130100.42
408	166464	67917312	20.1990	7.4169	8241.2	3.3276	.00245098	1281.769	130740.52
409	167281	68417929	20.2237	7.4229	8271.5	3.3292	.00244499	1284.910	131382.19
410	**168100**	**68921000**	**20.2485**	**7.4290**	**8301.9**	**3.3308**	**.00243902**	**1288.052**	**132025.43**
411	168921	69426531	20.2731	7.4350	8332.3	3.3325	.00243309	1291.193	132670.24
412	169744	69934528	20.2978	7.4410	8362.7	3.3341	.00242718	1294.335	133316.63
413	170569	70444997	20.3224	7.4470	8393.2	3.3357	.00242131	1297.477	133964.58
414	171396	70957944	20.3470	7.4530	8423.7	3.3373	.00241546	1300.618	134614.10
415	172225	71473375	20.3715	7.4590	8454.2	3.3390	.00240964	1303.760	135265.20
416	173056	71991296	20.3961	7.4650	8484.8	3.3406	.00240385	1306.901	135917.86
417	173889	72511713	20.4206	7.4710	8515.4	3.3422	.00239808	1310.043	136572.10
418	174724	73034632	20.4450	7.4770	8546.0	3.3438	.00239234	1313.185	137227.91
419	175561	73560059	20.4695	7.4829	8576.7	3.3454	.00238663	1316.326	137885.29
420	**176400**	**74088000**	**20.4939**	**7.4889**	**8607.4**	**3.3470**	**.00238095**	**1319.468**	**138544.24**
421	177241	74618461	20.5183	7.4948	8638.2	3.3485	.00237530	1322.609	139204.76
422	178084	75151448	20.5426	7.5007	8669.0	3.3501	.00236967	1325.751	139866.85
423	178929	75686967	20.5670	7.5067	8699.8	3.3517	.00236407	1328.893	140530.51
424	179776	76225024	20.5913	7.5126	8730.7	3.3533	.00235849	1332.034	141195.74
425	180625	76765625	20.6155	7.5185	8761.6	3.3559	.00235294	1335.176	141862.54
426	181476	77308776	20.6398	7.5244	8792.5	3.3564	.00234742	1338.317	142530.92
427	182329	77854483	20.6640	7.5302	8823.5	3.3580	.00234192	1341.459	143200.86
428	183184	78402752	20.6882	7.5361	8854.5	3.3596	.00233645	1344.601	143872.38
429	184041	78953589	20.7123	7.5420	8885.6	3.3612	.00233100	1347.742	144545.46

Table 4.—Properties of Numbers—*Continued*

N	N^2	N^3	$\sqrt{N}$	$\sqrt[3]{N}$	$N^{3/2}$	$\sqrt[5]{N}$	$\frac{1}{N}$	Circle (N = Diam.)	
								Circum.	Area
430	**184900**	**79507000**	**20.7364**	**7.5478**	**8916.7**	**3.3627**	**.00232558**	**1350.884**	**145220.12**
431	185761	80062991	20.7605	7.5537	8947.8	3.3643	.00232019	1354.025	145896.35
432	186624	80621568	20.7846	7.5595	8979.0	3.3659	.00231481	1357.167	146574.15
433	187489	81182737	20.8087	7.5654	9010.1	3.3674	.00230947	1360.308	147253.52
434	188356	81746504	20.8327	7.5712	9041.4	3.3690	.00230415	1363.450	147934.46
435	189225	82312875	20.8567	7.5770	9072.7	3.3705	.00229885	1366.592	148616.97
436	190096	82881856	20.8806	7.5828	9104.0	3.3720	.00229358	1369.733	149301.05
437	190969	83453453	20.9045	7.5886	9135.3	3.3736	.00228833	1372.875	149986.70
438	191844	84027672	20.9284	7.5944	9166.7	3.3752	.00228311	1376.016	150673.92
439	192721	84604519	20.9523	7.6001	9198.1	3.3767	.00227790	1379.158	151362.72
440	**193600**	**85184000**	**20.9762**	**7.6059**	**9229.5**	**3.3783**	**.00227273**	**1382.300**	**152053.08**
441	194481	85766121	21.0000	7.6117	9261.0	3.3798	.00226757	1385.441	152745.02
442	195364	86350888	21.0238	7.6174	9292.5	3.3813	.00226244	1388.583	153438.53
443	196249	86938307	21.0476	7.6232	9324.1	3.3828	.00225734	1391.724	154133.60
444	197136	87528384	21.0713	7.6289	9355.7	3.3844	.00225225	1394.866	154830.25
445	198025	88121125	21.0950	7.6346	9387.3	3.3859	.00224719	1398.008	155528.47
446	198916	88716536	21.1187	7.6403	9419.0	3.3874	.00224215	1401.149	156228.26
447	199809	89314623	21.1424	7.6460	9450.7	3.3889	.00223714	1404.291	156929.62
448	200704	89915392	21.1660	7.6517	9482.4	3.3904	.00223214	1407.432	157632.55
449	201601	90518849	21.1896	7.6574	9514.2	3.3919	.00222717	1410.574	158337.05
450	**202500**	**91125000**	**21.2132**	**7.6631**	**9546.0**	**3.3935**	**.00222222**	**1413.710**	**159043.13**
451	203401	91733851	21.2368	7.6688	9577.8	3.3950	.00221729	1416.857	159750.77
452	204304	92345408	21.2603	7.6744	9609.6	3.3965	.00221239	1419.999	160459.99
453	205209	92959677	21.2838	7.6801	9641.5	3.3980	.00220751	1423.140	161170.77
454	206116	93576664	21.3073	7.6857	9673.5	3.3995	.00220264	1426.282	161883.13
455	207025	94196375	21.3307	7.6914	9705.5	3.4010	.00219780	1429.423	162597.05
456	207936	94818816	21.3542	7.6970	9737.5	3.4025	.00219298	1432.565	163312.55
457	208849	95443993	21.3776	7.7026	9769.5	3.4039	.00218818	1435.707	164029.62
458	209764	96071912	21.4009	7.7082	9801.6	3.4054	.00218341	1438.849	164748.26
459	210681	96702579	21.4243	7.7138	9833.8	3.4069	.00217865	1441.990	165468.47
460	**211600**	**97336000**	**21.4476**	**7.7194**	**9865.9**	**3.4084**	**.00217391**	**1445.131**	**166190.25**
461	212521	97972181	21.4709	7.7250	9898.1	3.4199	.00216920	1448.273	166913.60
462	213444	98611128	21.4942	7.7306	9930.3	3.4113	.00216450	1451.415	167638.52
463	214369	99252847	21.5174	7.7362	9962.6	3.4128	.00215983	1454.556	168365.02
464	215296	99897344	21.5407	7.7418	9994.8	3.4143	.00215517	1457.698	169093.08
465	216225	100544625	21.5639	7.7473	10027.	3.4158	.00215054	1460.839	169822.72
466	217156	101194696	21.5870	7.7529	10060.	3.4173	.00214592	1463.981	170553.92
467	218089	101847563	21.6102	7.7584	10092.	3.4187	.00214133	1467.123	171286.70
468	219024	102503232	21.6333	7.7639	10124.	3.4202	.00213675	1470.264	172021.05
469	219961	103161709	21.6564	7.7695	10157.	3.4217	.00213220	1473.406	172756.96
470	**220900**	**103823000**	**21.6795**	**7.7750**	**10189.**	**3.4231**	**.00212766**	**1476.547**	**173494.45**
471	221841	104487111	21.7025	7.7805	10222.	3.4246	.00212314	1479.689	174233.51
472	222784	105154048	21.7256	7.7860	10255.	3.4260	.00211864	1482.830	174974.14
473	223729	105823817	21.7486	7.7915	10287.	3.4275	.00211416	1485.972	175716.34
474	224676	106496424	21.7715	7.7970	10320.	3.4289	.00210970	1489.114	176460.12
475	225625	107171875	21.7945	7.8025	10352.	3.4304	.00210526	1492.255	177205.46
476	226576	107850176	21.8174	7.8079	10385.	3.4318	.00210084	1495.397	177952.37
477	227529	108531333	21.8403	7.8134	10418.	3.4332	.00209644	1498.538	178700.86
478	228484	109215352	21.8632	7.8188	10450.	3.4347	.00209205	1501.680	179450.91
479	229441	109902239	21.8861	7.8243	10483.	3.4361	.00208768	1504.822	180202.54
480	**230400**	**110592000**	**21.9089**	**7.8297**	**10516.**	**3.4375**	**.00208333**	**1507.963**	**180955.74**
481	231361	111284641	21.9317	7.8352	10549.	3.4390	.00207900	1511.105	181710.50
482	232324	111980168	21.9545	7.8406	10582.	3.4404	.00207469	1514.246	182466.84
483	233289	112678587	21.9773	7.8460	10615.	3.4418	.00207039	1517.388	183224.75
484	234256	113379904	22.0000	7.8514	10648.	3.4433	.00206612	1520.530	183984.23
485	235225	114084125	22.0227	7.8568	10681.	3.4447	.00206186	1523.671	184745.28
486	236196	114791256	22.0454	7.8622	10714.	3.4461	.00205761	1526.813	185507.90
487	237169	115501303	22.0681	7.8676	10747.	3.4475	.00205339	1529.954	186272.10
488	238144	116214272	22.0907	7.8730	10780.	3.4489	.00204918	1533.096	187037.86
489	239121	116930169	22.1133	7.8784	10813.	3.4504	.00204499	1536.238	187805.19
490	**240100**	**117649000**	**22.1359**	**7.8837**	**10847.**	**3.4518**	**.00204082**	**1539.379**	**188574.10**
491	241081	118370771	22.1585	7.8891	10880.	3.4532	.00203666	1542.521	189344.57
492	242064	119095488	22.1811	7.8944	10913.	3.4546	.00203252	1545.662	190116.62
493	243049	119823157	22.2036	7.8998	10946.	3.4560	.00202840	1548.804	190890.24
494	244036	120553784	22.2261	7.9051	10980.	3.4574	.00202429	1551.945	191665.43
495	245025	121287375	22.2486	7.9105	11013.	3.4588	.00202020	1555.087	192442.18
496	246016	122023936	22.2711	7.9158	11046.	3.4602	.00201613	1558.229	193220.51
497	247009	122763473	22.2935	7.9211	11080.	3.4616	.00201207	1561.370	194000.41
498	248004	123505992	22.3159	7.9264	11113.	3.4630	.00200803	1564.512	194781.89
499	249001	124251499	22.3383	7.9317	11147.	3.4643	.00200401	1567.653	195564.93

Table 4.—Properties of Numbers—*Continued*

N	N^2	N^3	$\sqrt{N}$	$\sqrt[3]{N}$	$N^{3/2}$	$\sqrt[5]{N}$	$\frac{1}{N}$	Circle (N = Diam.) Circum.	Area
500	**250000**	**125000000**	**22.3607**	**7.9370**	**11180**	**3.4657**	**.00200000**	**1570.795**	**196349.54**
501	251001	125751501	22.3830	7.9423	11214	3.4671	.00199601	1573.937	197135.72
502	252004	126506008	22.4054	7.9476	11247	3.4685	.00199203	1577.078	197923.48
503	253009	127263527	22.4277	7.9528	11281	3.4699	.00198807	1580.220	198712.80
504	254016	128024064	22.4499	7.9581	11315	3.4713	.00198413	1583.361	199503.70
505	255025	128787625	22.4722	7.9634	11348	3.4726	.00198020	1586.503	200296.17
506	256036	129554216	22.4944	7.9686	11382	3.4740	.00197628	1589.645	201090.20
507	257049	130323843	22.5167	7.9739	11416	3.4754	.00197239	1592.786	201885.81
508	258064	131096512	22.5389	7.9791	11450	3.4768	.00196850	1595.928	202682.99
509	259081	131872229	22.5610	7.9843	11484	3.4781	.00196464	1599.069	203481.74
510	**260100**	**132651000**	**22.5832**	**7.9896**	**11517**	**3.4795**	**.00196078**	**1602.211**	**204282.06**
511	261121	133432831	22.6053	7.9948	11551	3.4808	.00195695	1605.352	205083.95
512	262144	134217728	22.6274	8.0000	11585	3.4822	.00195313	1608.494	205887.42
513	263169	135005697	22.6495	8.0052	11619	3.4836	.00194932	1611.636	206692.45
514	264196	135796744	22.6716	8.0104	11653	3.4849	.00194553	1614.777	207499.05
515	265225	136590875	22.6936	8.0156	11687	3.4863	.00194175	1617.919	208307.23
516	266256	137388096	22.7156	8.0208	11721	3.4876	.00193798	1621.060	209116.97
517	267289	138188413	22.7376	8.0260	11755	3.4890	.00193424	1624.202	209928.29
518	268324	138991832	22.7596	8.0311	11789	3.4904	.00193050	1627.344	210741.16
519	269361	139798359	22.7816	8.0363	11824	3.4917	.00192678	1630.485	211555.63
520	**270400**	**140608000**	**22.8035**	**8.0415**	**11858**	**3.4930**	**.00192308**	**1633.627**	**212371.66**
521	271441	141420761	22.8254	8.0466	11892	3.4944	.00191939	1636.768	213189.26
522	272484	142236648	22.8473	8.0517	11926	3.4957	.00191571	1639.910	214008.43
523	273529	143055667	22.8692	8.0569	11960	3.4970	.00191205	1643.052	214829.17
524	274576	143877824	22.8910	8.0620	11995	3.4984	.00190840	1646.193	215651.49
525	275625	144703125	22.9129	8.0671	12029	3.4997	.00190476	1649.335	216475.37
526	276676	145531576	22.9347	8.0723	12064	3.5010	.00190114	1652.476	217300.82
527	277729	146363183	22.9565	8.0774	12098	3.5024	.00189753	1655.618	218127.85
528	278784	147197952	22.9783	8.0825	12133	3.5037	.00189394	1658.760	218956.44
529	279841	148035889	23.0000	8.0876	12167	3.5050	.00189036	1661.901	219786.61
530	**280900**	**148877000**	**23.0217**	**8.0927**	**12202**	**3.5064**	**.00188679**	**1665.043**	**220618.34**
531	281961	149721291	23.0434	8.0978	12236	3.5077	.00188324	1668.184	221451.65
532	283024	150568768	23.0651	8.1028	12271	3.5090	.00187970	1671.326	222286.53
533	284089	151419437	23.0868	8.1079	12305	3.5103	.00187617	1674.467	223122.98
534	285156	152273304	23.1084	8.1130	12340	3.5116	.00187266	1677.609	223961.00
535	286225	153130375	23.1301	8.1180	12375	3.5130	.00186916	1680.751	224800.59
536	287296	153990656	23.1517	8.1231	12410	3.5143	.00186567	1683.892	225641.75
537	288369	154854153	23.1733	8.1281	12444	3.5156	.00186220	1687.034	226484.48
538	289444	155720872	23.1948	8.1332	12479	3.5169	.00185874	1690.175	227328.79
539	290521	156590819	23.2164	8.1382	12514	3.5182	.00185529	1693.317	228174.66
540	**291600**	**157464000**	**23.2379**	**8.1433**	**12549**	**3.5195**	**.00185185**	**1696.459**	**229022.10**
541	292681	158340421	23.2594	8.1483	12583	3.5208	.00184843	1699.600	229871.12
542	293764	159220088	23.2809	8.1533	12618	3.5221	.00184502	1702.742	230721.71
543	294849	160103007	23.3024	8.1583	12653	3.5234	.00184162	1705.883	231573.86
544	295936	160989184	23.3238	8.1633	12688	3.5247	.00183824	1709.025	232427.59
545	297025	161878625	23.3452	8.1683	12723	3.5260	.00183486	1712.167	233282.89
546	298116	162771336	23.3666	8.1733	12758	3.5273	.00183150	1715.308	234139.76
547	299209	163667323	23.3880	8.1783	12793	3.5286	.00182815	1718.450	234998.20
548	300304	164566592	23.4094	8.1833	12828	3.5299	.00182482	1721.591	235858.21
549	301401	165469149	23.4307	8.1882	12863	3.5311	.00182149	1724.733	236719.79
550	**302500**	**166375000**	**23.4521**	**8.1932**	**12899**	**3.5324**	**.00181818**	**1727.875**	**237582.94**
551	303601	167284151	23.4734	8.1982	12934	3.5337	.00181488	1731.016	238447.67
552	304704	168196608	23.4947	8.2031	12969	3.5350	.00181159	1734.158	239313.96
553	305809	169112377	23.5160	8.2081	13004	3.5363	.00180832	1737.299	240181.83
554	306916	170031464	23.5372	8.2130	13040	3.5376	.00180505	1740.441	241051.26
555	308025	170953875	23.5584	8.2180	13075	3.5388	.00180180	1743.582	241922.27
556	309136	171879616	23.5797	8.2229	13110	3.5401	.00179856	1746.724	242794.85
557	310249	172808693	23.6008	8.2278	13146	3.5414	.00179533	1749.866	243668.99
558	311364	173741112	23.6220	8.2327	13181	3.5426	.00179211	1753.007	244544.71
559	312481	174676879	23.6432	8.2377	13217	3.5439	.00178891	1756.149	245422.00
560	**313600**	**175616000**	**23.6643**	**8.2426**	**13252**	**3.5451**	**.00178571**	**1759.290**	**246300.86**
561	314721	176558481	23.6854	8.2475	13288	3.5464	.00178253	1762.432	247181.30
562	315844	177504328	23.7065	8.2524	13323	3.5477	.00177936	1765.574	248063.30
563	316969	178453547	23.7276	8.2573	13359	3.5490	.00177620	1768.715	248946.87
564	318096	179406144	23.7487	8.2621	13394	3.5502	.00177305	1771.857	249832.01
565	319225	180362125	23.7697	8.2670	13430	3.5515	.00176991	1774.998	250718.73
566	320356	181321496	23.7908	8.2719	13466	3.5527	.00176678	1778.140	251607.01
567	321489	182284263	23.8118	8.2768	13501	3.5540	.00176367	1781.282	252496.87
568	322624	183250432	23.8328	8.2816	13537	3.5553	.00176056	1784.423	253388.30
569	323761	184220009	23.8537	8.2865	13573	3.5565	.00175747	1787.565	254281.29

Table 4.—Properties of Numbers—*Continued*

N	N^2	N^3	$\sqrt{N}$	$\sqrt[3]{N}$	$N^{3/2}$	$\sqrt[5]{N}$	$\frac{1}{N}$	Circle (N = Diam.) Circum.	Area
570	**324900**	**185193000**	**23.8747**	**8.2913**	**13609**	**3.5577**	**.00175439**	**1790.706**	**255175.86**
571	326041	186169411	23.8956	8.2962	13644	3.5590	.00175131	1793.848	256072.00
572	327184	187149248	23.9165	8.3010	13680	3.5602	.00174825	1796.989	256969.71
573	328329	188132517	23.9374	8.3059	13716	3.5615	.00174520	1800.131	257868.99
574	329476	189119224	23.9583	8.3107	13752	3.5627	.00174216	1803.273	258769.85
575	330625	190109375	23.9792	8.3155	13788	3.5640	.00173913	1806.414	259672.27
576	331776	191102976	24.0000	8.3203	13824	3.5652	.00173611	1809.556	260576.26
577	332929	192100033	24.0208	8.3251	13860	3.5664	.00173310	1812.697	261481.83
578	334084	193100552	24.0416	8.3300	13896	3.5677	.00173010	1815.839	262388.96
579	335241	194104539	24.0624	8.3348	13932	3.5689	.00172712	1818.981	263297.67
580	**336400**	**195112000**	**24.0832**	**8.3396**	**13968**	**3.5702**	**.00172414**	**1822.122**	**264207.94**
581	337561	196122941	24.1039	8.3443	14004	3.5714	.00172117	1825.264	265119.79
582	338724	197137368	24.1247	8.3491	14040	3.5726	.00171821	1828.405	266033.21
583	339889	198155287	24.1454	8.3539	14077	3.5738	.00171527	1831.547	266948.20
584	341056	199176704	24.1661	8.3587	14113	3.5751	.00171233	1834.689	267864.76
585	342225	200201625	24.1868	8.3634	14149	3.5763	.00170940	1837.830	268782.89
586	343396	201230056	24.2074	8.3682	14186	3.5775	.00170648	1840.972	269702.59
587	344569	202262003	24.2281	8.3730	14222	3.5787	.00170358	1844.113	270623.86
588	345744	203297472	24.2487	8.3777	14258	3.5799	.00170068	1847.255	271546.70
589	346921	204336469	24.2693	8.3825	14295	3.5812	.00169779	1850.397	272471.12
590	**348100**	**205379000**	**24.2899**	**8.3872**	**14331**	**3.5824**	**.00169492**	**1853.538**	**273397.10**
591	349281	206425071	24.3105	8.3919	14368	3.5886	.00169205	1856.680	274324.66
592	350464	207474688	24.3311	8.3967	14404	3.5848	.00168919	1859.821	275253.78
593	351649	208527857	24.3516	8.4014	14440	3.5860	.00168634	1862.963	276184.48
594	352836	209584584	24.3721	8.4061	14477	3.5872	.00168350	1866.104	277116.75
595	354025	210644875	24.3926	8.4108	14514	3.5884	.00168067	1869.246	278050.58
596	355216	211708736	24.4131	8.4155	14550	3.5896	.00167785	1872.388	278985.99
597	356409	212776173	24.4336	8.4202	14587	3.5908	.00167504	1875.529	279922.97
598	357604	213847192	24.4540	8.4249	14624	3.5920	.00167224	1878.671	280861.52
599	358801	214921799	24.4745	8.4296	14660	3.5932	.00166945	1881.812	281801.65
600	**360000**	**216000000**	**24.4949**	**8.4343**	**14697**	**3.5944**	**.00166667**	**1884.954**	**282743.34**
601	361201	217081801	24.5153	8.4390	14734	3.5956	.00166389	1888.096	283686.60
602	362404	218167208	24.5357	8.4437	14770	3.5958	.00166113	1891.237	284631.44
603	363609	219256227	24.5561	8.4484	14807	3.5980	.00165837	1894.379	285577.84
604	364816	220348864	24.5764	8.4530	14844	3.5992	.00165563	1897.520	286525.82
605	366025	221445125	24.5967	8.4577	14881	3.6004	.00165289	1900.662	287475.36
606	367236	222545016	24.6171	8.4623	14918	3.6016	.00165017	1903.804	288426.48
607	368449	223648543	24.6374	8.4670	14955	3.6028	.00164745	1906.945	289379.17
608	369664	224755712	24.6577	8.4716	14992	3.6040	.00164474	1910.087	290333.43
609	370881	225866529	24.6779	8.4763	15029	3.6052	.00164204	1913.228	291289.26
610	**372100**	**226981000**	**24.6982**	**8.4809**	**15066**	**3.6063**	**.00163934**	**1916.370**	**292246.66**
611	373321	228099131	24.7184	8.4856	15103	3.6075	.00163666	1919.511	293205.63
612	374544	229220928	24.7386	8.4902	15140	3.6087	.00163399	1922.653	294166.17
613	375769	230346397	24.7588	8.4948	15177	3.6099	.00163132	1925.795	295128.28
614	376996	231475544	24.7790	8.4994	15214	3.6111	.00162866	1928.936	296091.97
615	378225	232608375	24.7992	8.5040	15252	3.6122	.00162602	1932.078	297057.22
616	379456	233744896	24.8193	8.5086	15289	3.6134	.00162338	1935.219	298024.05
617	380689	234885113	24.8395	8.5132	15326	3.6146	.00162075	1938.361	298992.44
618	381924	236029032	24.8596	8.5178	15363	3.6158	.00161812	1941.503	299962.41
619	383161	237176659	24.8797	8.5224	15400	3.6169	.00161551	1944.644	300933.95
620	**384400**	**238328000**	**24.8998**	**8.5270**	**15437**	**3.6181**	**.00161290**	**1947.786**	**301907.05**
621	385641	239483061	24.9199	8.5316	15475	3.6192	.00161031	1950.927	302881.73
622	386884	240641848	24.9399	8.5362	15513	3.6204	.00160772	1954.069	303857.98
623	388129	241804367	24.9600	8.5408	15550	3.6216	.00160514	1957.211	304835.80
624	389376	242970624	24.9800	8.5453	15588	3.6227	.00160256	1960.352	305815.20
625	390625	244140625	25.0000	8.5499	15625	3.6239	.00160000	1963.494	306796.16
626	391876	245314376	25.0200	8.5544	15663	3.6250	.00159744	1966.635	307778.69
627	393129	246491883	25.0400	8.5590	15700	3.6262	.00159490	1969.777	308762.79
628	394384	247673152	25.0599	8.5635	15738	3.6274	.00159236	1972.919	309748.47
629	395641	248858189	25.0799	8.5681	15775	3.6285	.00158983	1976.060	310735.71
630	**396900**	**250047000**	**25.0998**	**8.5726**	**15813**	**3.6297**	**.00158730**	**1979.202**	**311724.53**
631	398161	251239591	25.1197	8.5772	15850	3.6309	.00158479	1982.343	312714.92
632	399424	252435968	25.1396	8.5817	15888	3.6320	.00158228	1985.485	313706.88
633	400689	253636137	25.1595	8.5862	15926	3.6331	.00157978	1988.626	314700.40
634	401956	254840104	25.1794	8.5907	15964	3.6343	.00157729	1991.768	315695.50
635	403225	256047875	25.1992	8.5952	16002	3.6354	.00157480	1994.910	316692.17
636	404496	257259456	25.2190	8.5997	16040	3.6366	.00157233	1998.051	317690.42
637	405769	258474853	25.2389	8.6043	16077	3.6377	.00156986	2001.193	318690.23
638	407044	259694072	25.2587	8.6088	16115	3.6389	.00156740	2004.334	319691.61
639	408321	260917119	25.2784	8.6132	16153	3.6400	.00156495	2007.476	320694.56

Table 4.—Properties of Numbers—*Continued*

N					$N^{3/2}$		$\frac{1}{N}$	Circum.	Area
640	**409600**	**262144000**	**25.2982**	**8.6177**	**16191**	**3.6411**	**.00156250**	**2010.618**	**321699.09**
641	410881	263374721	25.3180	8.6222	16229	3.6423	.00156006	2013.759	322705.18
642	412164	264609288	25.3377	8.6267	16267	3.6435	.00155763	2016.901	323712.85
643	413449	65847707	25.3574	8.6312	16305	3.6446	.00155521	2020.042	324722.09
644	414736	7089984	25.3772	8.6357	16343	3.6457	.00155280	2023.184	325732.89
645	416025	8336125	25.3969	8.6401	16381	3.6468	.00155039	2026.326	326745.27
646	417316	9586136	25.4165	8.6446	16419	3.6479	.00154799	2029.467	327759.22
647	418609	70840023	25.4362	8.6490	16457	3.6499	.00154560	2032.609	328774.74
648	419904	2097792	25.4558	8.6535	16495	3.6502	.00154321	2035.750	329791.83
649	421201	3359449	25.4755	8.6579	16534	3.6513	.00154083	2038.892	330810.49
650	**422500**	**4625000**	**25.4951**	**8.6624**	**16572**	**3.6524**	**.00153846**	**2042.034**	**331830.72**
651	423801	5894451	25.5147	8.6668	16610	3.6536	.00153610	2045.175	332852.53
652	425104	7167808	25.5343	8.6713	16648	3.6547	00153374	2048.317	333875.90
653	426409	8445077	25.5539	8.6757	16687	3.6558	.00153139	2051.458	334900.85
654	427716	9726264	25.5734	8.6801	16725	3.6569	.00152905	2054.600	335927.36
655	429025	1011375	25.5930	8.6845	16764	3.6580	.00152672	2057.741	336955.45
656	430336	2300416	25.6125	8.6890	16802	3.6592	.00152439	2060.883	337985.10
657	431649	83593393	25.6320	8.6934	16840	3.6603	.00152207	2064.025	339016.33
658	432964	84890312	25.6515	8.6978	16879	3.6614	.00151976	2067.166	340049.13
659	434281	6191179	25.6710	8.7022	16917	3.6625	.00151745	2070.308	341083.50
660	**435600**	**7496000**	**25.6905**	**8.7066**	**16956**	**3.6636**	**.00151515**	**2073.449**	**342119.44**
661	436921	88804781	25.7099	8.7110	16994	3.6647	00151286	2076.591	343156.95
662	438244	90117528	25.7294	8.7154	17033	3.6658	00151057	2079.733	344196.03
663	439569	91434247	25.7488	8.7198	17071	3.6669	00150830	2082.874	345236.69
664	440896	92754944	25.7682	8.7241	17110	3.6680	.00150602	2086.016	346278.91
665	442225	94079625	25.7876	8.7285	17149	3.6691	.00150376	2089.157	347322.70
666	443556	95408296	25.8070	8.7329	17187	3.6702	.00150150	2092.299	348368.07
667	444889	96740963	25.8263	8.7373	17226	3.6713	.00149925	2095.441	349415.00
668	446224	98077632	25.8457	8.7416	17265	3.6724	.00149701	2098.582	350463.51
669	447561	99418309	25.8650	8.7460	17304	3.6735	.00149477	2101.724	351513.59
670	**448900**	**300763000**	**25.8844**	**8.7503**	**17343**	**3.6746**	**.00149254**	**2104.865**	**352565.24**
671	45024	02111711	25.9037	8.7547	17381	3.6757	.00149031	2108.007	353618.45
672	45158	03464448	25.9230	8.7590	17420	3.6768	.00148810	2111.148	354673.24
673	45292	04821217	25.9422	8.7634	17459	3.6779	.00148588	2114.290	355729.60
674	45427	06182024	25.9615	8.7677	17498	3.6790	.00148368	2117.432	356787.54
675	45562	307546875	25.9808	8.7721	17537	3.6801	.00148148	2120.573	357847.04
676	45697	308915776	26.0000	8.7764	17576	3.6812	.00147929	2123.715	358908.11
677	45832	310288733	26.0192	8.7807	17615	3.6823	.00147710	2126.856	359970.75
678	45968	311665752	26.0384	8.7850	17654	3.6834	.00147493	2129.998	361034.97
679	46104	13046839	26.0576	8.7893	17693	3.6845	.00147275	2133.140	362100.75
680	**462400**	**314432000**	**26.0768**	**8.7937**	**17732**	**3.6856**	**.00147059**	**2136.281**	**363168.11**
681	46376	15821241	26.0960	8.7980	17771	3.6866	.00146843	2139.423	364237.04
682	46512	317214568	26.1151	8.8023	17810	3.6877	.00146628	2142.564	365307.54
683	46648	318611987	26.1343	8.8066	17850	3.6888	.00146413	2145 706	366379.60
684	46785	320013504	26.1534	8.8109	17889	3.6899	.00146199	2148.848	367453.24
685	46922	321419125	26.1725	8.8152	17928	3.6909	.00145985	2151.989	368528.45
686	47059	322828856	26.1916	8.8194	17967	6920	.00145773	2155.131	369605.23
687	47196	324242703	26.2107	8.8237	18007	6931	.00145560	2158.272	370683.59
688	47334	325660672	26.2298	8.8280	18046	6942	.00145349	2161.414	371763.51
689	47472	327082769	26.2488	8.8323	18085	6953	.00145138	2164.556	372845.00
690	**47610**	**328509000**	**26.2679**	**8.8366**	**18125**	**6963**	**.00144928**	**2167.697**	**373928.07**
691	47748	329939371	26.2869	8.8408	18164	6974	.00144718	2170.839	375012.70
692	47886	331373888	26.3059	8.8451	18204	6985	.00144509	2173.980	376098.91
693	48024	332812557	26.3249	8.8493	18243	3.6995	.00144300	2177.122	377186.68
694	48163	334255384	26.3439	8.8536	18283	3.7006	.00144092	2180.263	378276.03
695	48302	335702375	26.3629	8.8578	18322	3.7016	.00143885	2183.405	379366.95
696	48441	337153536	26.3818	8.8621	18362	3.7027	.00143678	2186.547	380459.44
697	48580	338608873	26.4008	8.8663	18401	3.7038	.00143472	2189.688	381553.50
698	48720	340068392	26.4197	8.8706	18441	3.7049	.00143266	2192.830	382649.13
699	48860	341532099	26.4386	8.8748	18480	3.7059	.00143062	2195.971	383746.33
700	**49000**	**343000000**	**26.4575**	**8.8790**	**18520**	**3.7070**	**.00142857**	**2199.113**	**384845.10**
701	49140	344472101	26.4764	8.8833	18560	3.7080	.00142653	2202.255	385945.44
702	49280	345948408	26.4953	8.8875	18600	3.7091	.00142450	2205.396	387047.36
703	49420	347428927	26.5141	8.8917	18640	3.7101	00142248	2208.538	388150.84
704	49561	348913664	26.5330	8.8959	18679	3.7112	00142045	2211.679	389255.90
705	49702	350402625	26.5518	8.9001	18719	3.7123	00141844	2214.821	390362.52
706	49843	351895816	26.5707	8.9043	18759	3.7133	00141643	2217.963	391470.72
707	4998	353393243	26.5895	8.9085	18799	3.7144	00141443	2221.104	392580.49
708	50126	354894912	26.6083	8.9127	18839	3.7154	00141243	2224.246	393691.82
709	50268	6400829	26.6271	8.9169	18879	3.7165	00141044	2227.387	394804.73

Table 4.—Properties of Numbers—*Continued*

N	N^2	N^3	$\sqrt{N}$	$\sqrt[3]{N}$	$N^{3/2}$	$\sqrt[5]{N}$	$\frac{1}{N}$	Circle (N = Diam.) Circum.	Area
710	**504100**	**357911000**	**26.6458**	**8.9211**	**18919**	**3.7175**	**.00140845**	**2230.529**	**395919.21**
711	505521	359425431	26.6646	8.9253	18959	3.7185	.00140647	2233.670	397035.26
712	506944	360944128	26.6833	8.9295	18999	3.7196	.00140449	2236.812	398152.89
713	508369	362467097	26.7021	8.9337	19039	3.7206	.00140252	2239.954	399272.08
714	509796	363994344	26.7208	8.9378	19079	3.7217	.00140056	2243.095	400392.84
715	511225	365525875	26.7395	8.9420	19119	3.7227	.00139860	2246.237	401515.18
716	512656	367061696	26.7582	8.9462	19159	3.7238	.00139665	2249.378	402639.08
717	514089	368601813	26.7769	8.9503	19199	3.7248	.00139470	2252.520	403764.56
718	515524	370146232	26.7955	8.9545	19239	3.7258	.00139276	2255.662	404891.60
719	516961	371694959	26.8142	8.9587	19280	3.7269	.00139082	2258.803	406020.22
720	**518400**	**373248000**	**26.8328**	**8.9628**	**19320**	**3.7279**	**.00138889**	**2261.945**	**407150.41**
721	519841	374805361	26.8514	8.9670	19360	3.7290	.00138696	2265.086	408282.17
722	521284	376367048	26.8701	8.9711	19400	3.7300	.00138504	2268.228	409415.50
723	522729	377933067	26.8887	8.9752	19440	3.7310	.00138313	2271.370	410550.40
724	524176	379503424	26.9072	8.9794	19481	3.7321	.00138122	2274.511	411686.87
725	525625	381078125	26.9258	8.9835	19521	3.7331	.00137931	2277.653	412824.91
726	527076	382657176	26.9444	8.9876	19562	3.7341	.00137741	2280.794	413964.52
727	528529	384240583	26.9629	8.9918	19602	3.7351	.00137552	2283.936	415105.71
728	529984	385828352	26.9815	8.9959	19643	3.7362	.00137363	2287.078	416248.46
729	531441	387420489	27.0000	9.0000	19683	3.7372	.00137174	2290.219	417392.79
730	**532900**	**389017000**	**27.0185**	**9.0041**	**19724**	**3.7382**	**.00136986**	**2293.361**	**418538.68**
731	534361	390617891	27.0370	9.0082	19764	3.7392	.00136799	2296.502	419686.15
732	535824	392223168	27.0555	9.0123	19805	3.7403	.00136612	2299.644	420835.19
733	537289	393832837	27.0740	9.0164	19845	3.7413	.00136426	2302.785	421985.79
734	538756	395446904	27.0924	9.0205	19886	3.7423	.00136240	2305.927	423137.97
735	540225	397065375	27.1109	9.0246	19927	3.7433	.00136054	2309.069	424291.72
736	541696	398688256	27.1293	9.0287	19967	3.7443	.00135870	2312.210	425447.04
737	543169	400315553	27.1477	9.0328	20008	3.7454	.00135685	2315.352	426603.94
738	544644	401947272	27.1662	9.0369	20049	3.7464	.00135501	2318.493	427762.40
739	546121	403583419	27.1846	9.0410	20090	3.7474	.00135318	2321.635	428922.43
740	**547600**	**405224000**	**27.2029**	**9.0450**	**20130**	**3.7484**	**.00135135**	**2324.777**	**430084.03**
741	549081	406869021	27.2213	9.0491	20171	3.7494	.00134953	2327.918	431247.21
742	550564	408518488	27.2397	9.0532	20212	3.7504	.00134771	2331.060	432411.95
743	552049	410172407	27.2580	9.0572	20253	3.7514	.00134590	2334.201	433578.27
744	553536	411830784	27.2764	9.0613	20294	3.7524	.00134409	2337.343	434746.16
745	555025	413493625	27.2947	9.0654	20335	3.7534	.00134228	2340.485	435915.62
746	556516	415160936	27.3130	9.0694	20376	3.7545	.00134048	2343.626	437086.64
747	558009	416832723	27.3313	9.0735	20417	3.7555	.00133869	2346.768	438259.24
748	559504	418508992	27.3496	9.0775	20458	3.7565	.00133690	2349.909	439433.41
749	561001	420189749	27.3679	9.0816	20499	3.7575	.00133511	2353.051	440609.16
750	**562500**	**421875000**	**27.3861**	**9.0856**	**20540**	**3.7585**	**.00133333**	**2356.193**	**441786.47**
751	564001	423564751	27.4044	9.0896	20581	3.7595	.00133156	2359.334	442965.35
752	565504	425259008	27.4226	9.0937	20622	3.7605	.00132979	2362.476	444145.80
753	567009	426957777	27.4408	9.0977	20663	3.7615	.00132802	2365.617	445327.83
754	568516	428661064	27.4591	9.1017	20704	3.7625	.00132626	2368.759	446511.42
755	570025	430368875	27.4773	9.1057	20745	3.7635	.00132450	2371.900	447696.59
756	571536	432081216	27.4955	9.1098	20787	3.7645	.00132275	2375.042	448883.32
757	573049	433798093	27.5136	9.1138	20828	3.7655	.00132100	2378.184	450071.63
758	574564	435519512	27.5318	9.1178	20869	3.7665	.00131926	2381.325	451261.51
759	576081	437245479	27.5500	9.1218	20910	3.7675	.00131752	2384.467	452452.96
760	**577600**	**438976000**	**27.5681**	**9.1258**	**20952**	**3.7685**	**.00131579**	**2387.608**	**453645.98**
761	579121	440711081	27.5862	9.1298	20993	3.7694	.00131406	2390.750	454840.57
762	580644	442450728	27.6043	9.1338	21035	3.7704	.00131234	2393.892	456036.73
763	582169	444194947	27.6225	9.1378	21076	3.7714	.00131062	2397.033	457234.46
764	583696	445943744	27.6405	9.1418	21117	3.7724	.00130890	2400.175	458433.77
765	585225	447697125	27.6586	9.1458	21159	3.7734	.00130719	2403.316	459634.64
766	586756	449455096	27.6767	9.1498	21200	3.7744	.00130548	2406.458	460837.08
767	588289	451217663	27.6948	9.1537	21242	3.7754	.00130378	2409.600	462041.10
768	589824	452984832	27.7128	9.1577	21283	3.7764	.00130208	2412.741	463246.69
769	591361	454756609	27.7308	9.1617	21325	3.7774	.00130039	2415.883	464453.84
770	**592900**	**456533000**	**27.7489**	**9.1657**	**21367**	**3.7784**	**.00129870**	**2419.024**	**465662.57**
771	594441	458314011	27.7669	9.1696	21408	3.7793	.00129702	2422.166	466872.87
772	595984	460099648	27.7849	9.1736	21450	3.7803	.00129534	2425.307	468084.74
773	597529	461889917	27.8029	9.1775	21492	3.7813	.00129366	2428.449	469298.18
774	599076	463684824	27.8209	9.1815	21533	3.7822	.00129199	2431.591	470513.19
775	600625	465484375	27.8388	9.1855	21575	3.7832	.00129032	2434.732	471729.77
776	602176	467288576	27.8568	9.1894	21617	3.7842	.00128866	2437.874	472947.92
777	603729	469097433	27.8747	9.1933	21658	3.7852	.00128700	2441.015	474167.65
778	605284	470910952	27.8927	9.1973	21700	3.7861	.00128535	2444.157	475388.94
779	606841	472729139	27.9106	9.2012	21742	3.7871	.00128370	2447.299	476611.81

Table 4.—Properties of Numbers—*Continued*

N	N^2	N^3	$\sqrt{N}$	$\sqrt[3]{N}$	$N^{3/2}$	$\sqrt[5]{N}$	$\frac{1}{N}$	Circle (N = Diam.) Circum.	Area
780	**608400**	**474552000**	**27.9285**	**9.2052**	**21784**	**3.7881**	**.00128205**	**2450.440**	**477836.24**
781	609961	476379541	27.9464	9.2091	21826	3.7890	.00128041	2453.582	479062.25
782	611524	478211768	27.9643	9.2130	21868	3.7900	.00127877	2456.723	480289.83
783	613089	480048687	27.9821	9.2170	21910	3.7910	.00127714	2459.865	481518.97
784	614656	481890304	28.0000	9.2209	21952	3.7920	.00127551	2463.007	482749.69
785	616225	483736625	28.0179	9.2248	21994	3.7929	.00127389	2466.148	483981.98
786	617796	485587656	28.0357	9.2287	22036	3.7939	.00127226	2469.290	485215.84
787	619369	487443403	28.0535	9.2326	22078	3.7949	.00127065	2472.431	486451.28
788	620944	489303872	28.0713	9.2365	22120	3.7959	.00126904	2475.573	487688.28
789	622521	491169069	28.0891	9.2404	22162	3.7969	.00126743	2478.715	488926.85
790	**624100**	**493039000**	**28.1069**	**9.2443**	**22205**	**3.7978**	**.00126582**	**2481.856**	**490166.99**
791	625681	494913671	28.1247	9.2482	22247	3.7987	.00126422	2484.998	491408.71
792	627264	496793088	28.1425	9.2521	22289	3.7997	.00126263	2488.139	492651.99
793	628849	498677257	28.1603	9.2560	22331	3.8006	.00126103	2491.281	493896.85
794	630436	500566184	28.1780	9.2599	22373	3.8016	.00125945	2494.422	495143.28
795	632025	502459875	28.1957	9.2638	22416	3.8025	.00125786	2497.564	496391.27
796	633616	504358336	28.2135	9.2677	22458	3.8035	.00125628	2500.706	497640.84
797	635209	506261573	28.2312	9.2716	22500	3.8044	.00125471	2503.847	498891.98
798	636804	508169592	28.2489	9.2754	22543	3.8054	.00125313	2506.989	500144.69
799	638401	510082399	28.266	9.2793	22585	3.8064	.00125156	2510.130	501398.97
800	**640000**	**512000000**	**28.2843**	**9.2832**	**22627**	**3.8073**	**.00125000**	**2513.272**	**502654.82**
801	641601		28.3019	9.2870	22670	3.8083	.00124844	2516.414	503912.25
802	643204	[illegible]	28.3196	9.2909	22712	3.8092	.00124688	2519.555	505171.24
803	644809	517781627	28.3373	9.2948	22755	3.8102	.00124533	2522.697	506431.80
804	646416	519718464	28.3549	9.2986	22797	3.8111	.00124378	2525.838	507693.94
805	648025	521660125	28.3725	9.3025	22840	3.8121	.00124224	2528.980	508957.64
806	649636	523606616	28.3901	9.3063	22883	3.8130	.00124069	2532.122	510222.92
807	651249	525557943	28.4077	9.3102	22925	3.8139	.00123916	2535.263	511489.77
808	652864	527514112	28.4253	9.3140	22968	3.8149	.00123762	2538.405	512758.19
809	654481	52947512	28.4429	9.3179	23010	3.8158	.00123609	2541.546	514028.18
810	**656100**	**531441000**	**28.4605**	**9.3217**	**23053**	**3.8168**	**.00123457**	**2544.688**	**515299.74**
811	657721	533411731	28.4781	9.3255	23096	3.8177	.00123305	2547.829	516572.87
812	659344	535387328	28.4956	9.3294	23138	3.8186	[illegible]	[illegible]	517847.57
813	660969	537367797	28.5132	9.3332	23181	3.8196	[illegible]		519123.84
814	662596	53935314.	28.5307	9.3370	23224	3.8205	.00122850	2557.254	520401.68
815	664225	541343375	28.5482	9.3408	23267	3.8215	.00122699	2560.396	521681.10
816	665856	543338496	28.5657	9.3447	23310	3.8224	.00122549	2563.537	522962.08
817	667489	545338513	28.5832	9.3485	23352	3.8234	.00122399	2566.679	524244.63
818	669124	547343432	28.6007	9.3523	23395	3.8243	.00122249	2569.821	525528.76
819	670761	549353259		9.3561	23438	3.8252	.00122100	2572.962	526814.46
820	**672400**	**551368000**	**28.6356**	**9.3599**	**23481**	**3.8262**	**.00121951**	**2576.104**	**528101.73**
821	674041		28.6531	9.3637	23524	3.8271	.00121803	2579.245	529390.56
822	675684	[illegible]	28.6705	9.3675	23567	3.8280	.00121655	2582.387	530680.97
823	677329	557441767	28.6880	9.3713	23610	3.8290	.00121507	2585.529	531972.95
824	678976	559476224	28.7054	9.3751	23653	3.8299	.00121359	2588.670	533266.50
825	680625	561515625	28.7228	9.3789	23696	3.8308	.00121212	2591.812	534561.62
826	682276	563559976	28.7402	9.3827	23740	3.8317	.00121065	2594.953	535858.32
827	683929	565609283	28.7576	9.3865	23783	3.8327	.00120919	2598.095	537156.58
828	685584	567663552	28.7750	9.3902	23826	3.8336	.00120773	2601.237	538456.41
829	687241	569722789	28.7924	9.3940	23869	3.8345	.00120627	2604.378	539757.82
830	**688900**	**571787000**	**28.8097**	**9.3978**	**23912**	**3.8355**	**.00120482**	**2607.520**	**541060.79**
831	690561	573856191	28.8271	9.4016	23955	3.8364	.00120337	2610.661	542365.34
832	692224	575930368	28.8444	9.4053	23999	3.8373	.00120192	2613.803	543671.46
833	693889	578009537	28.8617	9.4091	24042	3.8382	.00120048	2616.944	544979.15
834	695556	580093704	28.8791	9.4129	24085	3.8391	.00119904	2620.086	546288.40
835	697225	582182875	28.8964	9.4166	24128	3.8401	.00119760	2623.228	547599.23
836	698896	584277056	28.9137	9.4204	24172	3.8410	.00119617	2626.369	548911.63
837	700569	586376253	28.9310	9.4241	24215	3.8419	.00119474	2629.511	550225.61
838	702244	588480472	28.9482	9.4279	24259	3.8428	.00119332	2632.652	551541.15
839	703921	590589719	28.9655	9.4316	24302	3.8437	.00119190	2635.794	552858.26
840	**705600**	**592704000**	**28.9828**	**9.4354**	**24346**	**3.8446**	**.00119048**	**2638.936**	**554176.94**
841	707281	594823321	29.0000	9.4391	24389	3.8456	.00118906	2642.077	555497.20
842	708964	596947688	29.0172	9.4429	24432	3.8465	.00118765	2645.219	556819.02
843	710649	599077107	29.0345	9.4466	24476	3.8474	.00118624	2648.360	558142.42
844	712336	601211584	29.0517	9.4503	24520	3.8483	.00118483	2651.502	559467.39
845	714025	603351125	29.0689	9.4541	24563	3.8492	.00118343	2654.644	560793.92
846	715716	605495736	29.0861	9.4578	24607	3.8501	.00118203	2657.785	562122.03
847	717409	607645423	29.1033	9.4615	24650	3.8510	.00118064	2660.927	563451.71
848	719104	609800192	29.1204	9.4652	24694	3.8519	.00117925	2664.068	564782.96
849	720801	611960049	29.1376	9.4690	24738	3.8528	.00117786	2667.210	566115.78

Table 4.—Properties of Numbers—*Continued*

N	N^2	N^3	$\sqrt{N}$	$\sqrt[3]{N}$	$N^{3/2}$	$\sqrt[5]{N}$	$\frac{1}{N}$	Circle (N = Diam.) Circum.	Area
850	**722500**	**614125000**	**29.1548**	**9.4727**	**24782**	**3.8538**	**.00117647**	**2670.352**	**567450.17**
851	724201	616295051	29.1719	9.4764	24825	3.8547	.00117509	2673.493	568786.14
852	725904	618470208	29.1890	9.4801	24869	3.8556	.00117371	2676.635	570123.67
853	727609	620650477	29.2062	9.4838	24913	3.8565	.00117233	2679.776	571462.77
854	729316	622835864	29.2233	9.4875	24957	3.8574	.00117096	2682.918	572803.45
855	731025	625026375	29.2404	9.4912	25000	3.8582	.00116959	2686.059	574145.69
856	732736	627222016	29.2575	9.4949	25044	3.8592	.00116822	2689.201	575489.51
857	734449	629422793	29.2746	9.4986	25088	3.8601	.00116686	2692.343	576834.90
858	736164	631628712	29.2916	9.5023	25132	3.8610	.00116550	2695.484	578181.85
859	737881	633839779	29.3087	9.5060	25176	3.8619	.00116414	2698.626	579530.38
860	**739600**	**636056000**	**29.3258**	**9.5097**	**25220**	**3.8628**	**.00116279**	**2701.767**	**580880.48**
861	741321	638277381	29.3428	9.5134	25264	3.8637	.00116144	2704.909	582232.15
862	743044	640503928	29.3598	9.5171	25308	3.8646	.00116009	2708.051	583585.39
863	744769	642735647	29.3769	9.5207	25352	3.8655	.00115875	2711.192	584940.20
864	746496	644972544	29.3939	9.5244	25396	3.8664	.00115741	2714.334	586296.59
865	748225	647214625	29.4109	9.5281	25440	3.8673	.00115607	2717.475	587654.54
866	749956	649461896	29.4279	9.5317	25485	3.8682	.00115473	2720.617	589014.07
867	751689	651714363	29.4449	9.5354	25529	3.8691	.00115340	2723.759	590375.16
868	753424	653972032	29.4618	9.5391	25573	3.8700	.00115207	2726.900	591737.83
869	755161	656234909	29.4788	9.5427	25617	3.8708	.00115075	2730.042	593102.06
870	**756900**	**658503000**	**29.4958**	**9.5464**	**25661**	**3.8717**	**.00114943**	**2733.183**	**594467.87**
871	758641	660776311	29.5127	9.5501	25706	3.8726	.00114811	2736.325	595835.25
872	760384	663054848	29.5296	9.5537	25750	3.8735	.00114679	2739.466	597204.20
873	762129	665338617	29.5466	9.5574	25794	3.8744	.00114548	2742.608	598574.72
874	763876	667627624	29.5635	9.5610	25839	3.8753	.00114416	2745.750	599946.81
875	765625	669921875	29.5804	9.5647	25883	3.8762	.00114286	2748.891	601320.47
876	767376	672221376	29.5973	9.5683	25927	3.8771	.00114155	2752.033	602695.70
877	769129	674526133	29.6142	9.5719	25972	3.8780	.00114025	2755.174	604072.50
878	770884	676836152	29.6311	9.5756	26016	3.8789	.00113895	2758.316	605450.88
879	772641	679151439	29.6479	9.5792	26061	3.8797	.00113766	2761.458	606830.82
880	**774400**	**681472000**	**29.6648**	**9.5828**	**26105**	**3.8806**	**.00113636**	**2764.599**	**608212.34**
881	776161	683797841	29.6816	9.5865	26150	3.8815	.00113507	2767.741	609595.42
882	777924	686128968	29.6985	9.5901	26194	3.8823	.00113379	2770.882	610980.08
883	779689	688465387	29.7153	9.5937	26239	3.8832	.00113250	2774.024	612366.31
884	781456	690807104	29.7321	9.5973	26283	3.8841	.00113122	2777.166	613754.11
885	783225	693154125	29.7489	9.6010	26328	3.8850	.00112994	2780.307	615143.48
886	784996	695506456	29.7658	9.6046	26373	3.8859	.00112867	2783.449	616534.42
887	786769	697864103	29.7825	9.6082	26417	3.8868	.00112740	2786.590	617926.93
888	788544	700227072	29.7993	9.6118	26462	3.8877	.00112613	2789.732	619321.01
889	790321	702595369	29.8161	9.6154	26507	3.8885	.00112486	2792.874	620716.66
890	**792100**	**704969000**	**29.8329**	**9.6190**	**26551**	**3.8894**	**.00112360**	**2796.015**	**622113.89**
891	793881	707347971	29.8496	9.6226	26596	3.8902	.00112233	2799.157	623512.68
892	795664	709732288	29.8664	9.6262	26641	3.8911	.00112108	2802.298	624913.04
893	797449	712121957	29.8831	9.6298	26686	3.8920	.00111982	2805.440	626314.98
894	799236	714516984	29.8998	9.6334	26730	3.8929	.00111857	2808.581	627718.49
895	801025	716917375	29.9166	9.6370	26775	3.8937	.00111732	2811.723	629123.56
896	802816	719323136	29.9333	9.6406	26820	3.8946	.00111607	2814.865	630530.21
897	804609	721734273	29.9500	9.6442	26865	3.8955	.00111483	2818.006	631938.43
898	806404	724150792	29.9666	9.6477	26910	3.8963	.00111359	2821.148	633348.22
899	808201	726572699	29.9833	9.6513	26955	3.8972	.00111235	2824.289	634759.58
900	**810000**	**729000000**	**30.0000**	**9.6549**	**27000**	**3.8981**	**.00111111**	**2827.431**	**636172.51**
901	811801	731432701	30.0167	9.6585	27045	3.8989	.00110988	2830.573	637587.01
902	813604	733870808	30.0333	9.6620	27090	3.8998	.00110865	2833.714	639003.09
903	815409	736314327	30.0500	9.6656	27135	3.9007	.00110742	2836.856	640420.73
904	817216	738763264	30.0666	9.6692	27180	3.9015	.00110619	2839.997	641839.95
905	819025	741217625	30.0832	9.6727	27225	3.9024	.00110497	2843.139	643260.73
906	820836	743677416	30.0998	9.6763	27270	3.9032	.00110375	2846.281	644683.09
907	822649	746142643	30.1164	9.6799	27316	3.9041	.00110254	2849.422	646107.01
908	824464	748613312	30.1330	9.6834	27361	3.9050	.00110132	2852.564	647532.51
909	826281	751089429	30.1496	9.6870	27406	3.9059	.00110011	2855.705	648959.58
910	**828100**	**753571000**	**30.1662**	**9.6905**	**27451**	**3.9067**	**.00109890**	**2858.847**	**650388.22**
911	829921	756058031	30.1828	9.6941	27497	3.9076	.00109769	2861.988	651818.43
912	831744	758550528	30.1993	9.6976	27542	3.9084	.00109649	2865.130	653250.21
913	833569	761048497	30.2159	9.7012	27587	3.9093	.00109529	2868.272	654683.56
914	835396	763551944	30.2324	9.7047	27632	3.9101	.00109409	2871.413	656118.48
915	837225	766060875	30.2490	9.7082	27678	3.9110	.00109290	2874.555	657554.98
916	839056	768575296	30.2655	9.7118	27723	3.9118	.00109170	2877.696	658993.04
917	840889	771095213	30.2820	9.7153	27769	3.9127	.00109051	2880.838	660432.68
918	842724	773620632	30.2985	9.7188	27814	3.9135	.00108932	2883.980	661873.88
919	844561	776151559	30.3150	9.7224	27859	3.9144	.00108814	2887.121	663316.66

Table 4.—Properties of Numbers—*Continued*

N	N^2	N^3	$\sqrt{N}$	$\sqrt[3]{N}$	$N^{3/2}$	$\sqrt[5]{N}$	$\frac{1}{N}$	Circle (N = Diam.) Circum.	Area
920	**846400**	**778688000**	**30.3315**	**9.7259**	**27905**	**3.9153**	**.00108696**	**2890.263**	**664761.01**
921	848241	781229961	30.3480	9.7294	27950	3.9161	.00108578	2893.404	666206.92
922	850084	783777448	30.3645	9.7329	27996	3.9169	.00108460	2896.546	667654.41
923	851929	786330467	30.3809	9.7364	28042	3.9178	.00108342	2899.688	669103.47
924	853776	788889024	30.3974	9.7400	28087	3.9186	.00108225	2902.829	670554.10
925	855625	791453125	30.4138	9.7435	28133	3.9194	.00108108	2905.971	672006.30
926	857476	794022776	30.4302	9.7470	28179	3.9203	.00107991	2909.112	673460.08
927	859329	796597983	30.4467	9.7505	28224	3.9212	.00107875	2912.254	674915.42
928	861184	799178752	30.4631	9.7540	28270	3.9220	.00107759	2915.396	676372.33
929	863041	801765089	30.4795	9.7575	28315	3.9229	.00107643	2918.537	677830.82
930	**864900**	**804357000**	**30.4959**	**9.7610**	**28361**	**3.9237**	**.00107527**	**2921.679**	**679290.87**
931	866761	806954491	30.5123	9.7645	28407	3.9246	.00107411	2924.820	680752.50
932	868624	809557568	30.5287	9.7680	28453	3.9254	.00107296	2927.962	682215.69
933	870489	812166237	30.5450	9.7715	28499	3.9262	.00107181	2931.103	683680.46
934	872356	814780504	30.5614	9.7750	28544	3.9271	.00107066	2934.245	685146.80
935	874225	817400375	30.5778	9.7785	28590	3.9279	.00106952	2937.387	686614.71
936	876096	820025856	30.5941	9.7819	28636	3.9288	.00106838	2940.528	688084.19
937	877969	822656953	30.6105	9.7854	28682	3.9296	.00106724	2943.670	689555.24
938	879844	825293672	30.6268	9.7889	28728	3.9304	.00106610	2946.811	691027.86
939	881721	827936019	30.6431	9.7924	28774	3.9313	.00106496	2949.953	692502.05
940	**883600**	**830584000**	**30.6594**	**9.7959**	**28820**	**3.9321**	**.00106383**	**2953.095**	**693977.82**
941	885481	833237621	30.6757	9.7993	28866	3.9329	.00106270	2956.236	695455.15
942	887364	835896888	30.6920	9.8028	28912	3.9338	.00106157	2959.378	696934.06
943	889249	838561807	30.7083	9.8063	28958	3.9346	.00106045	2962.519	698414.53
944	891136	841232384	30.7246	9.8097	29004	3.9354	.00105932	2965.661	699896.58
945	893025	843908625	30.7409	9.8132	29050	3.9363	.00105820	2968.803	701380.19
946	894916	846590536	30.7571	9.8167	29096	3.9371	.00105708	2971.944	702865.38
947	896809	849278123	30.7734	9.8201	29142	3.9379	.00105597	2975.086	704352.14
948	898704	851971392	30.7896	9.8236	29189	3.9388	.00105485	2978.227	705840.47
949	900601	854670349	30.8058	9.8270	29235	3.9396	.00105374	2981.369	707330.37
950	**902500**	**857375000**	**30.8221**	**9.8305**	**29281**	**3.9404**	**.00105263**	**2984.511**	**708821.84**
951	904401	860085351	30.8383	9.8339	29327	3.9413	.00105152	2987.652	710314.88
952	906304	862801408	30.8545	9.8374	29374	3.9421	.00105042	2990.794	711809.50
953	908209	865523177	30.8707	9.8408	29420	3.9429	.00104932	2993.935	713305.68
954	910116	868250664	30.8869	9.8443	29466	3.9438	.00104822	2997.077	714803.43
955	912025	870983875	30.9031	9.8477	29513	3.9446	.00104712	3000.218	716302.76
956	913936	873722816	30.9192	9.8511	29559	3.9454	.00104603	3003.360	717803.66
957	915849	876467493	30.9354	9.8546	29605	3.9462	.00104493	3006.502	719306.12
958	917764	879217912	30.9516	9.8580	29652	3.9471	.00104384	3009.643	720810.16
959	919681	881974079	30.9677	9.8614	29698	3.9479	.00104275	3012.785	722315.77
960	**921600**	**884736000**	**30.9839**	**9.8648**	**29745**	**3.9487**	**.00104167**	**3015.926**	**723822.95**
961	923521	887503681	31.0000	9.8683	29791	3.9495	.00104058	3019.068	725331.70
962	925444	890277128	31.0161	9.8717	29838	3.9503	.00103950	3022.210	726842.02
963	927369	893056347	31.0322	9.8751	29884	3.9512	.00103842	3025.351	728353.91
964	929296	895841344	31.0483	9.8785	29931	3.9520	.00103734	3028.493	729867.37
965	931225	898632125	31.0644	9.8819	29977	3.9528	.00103627	3031.634	731382.40
966	933156	901428696	31.0805	9.8854	30024	3.9536	.00103520	3034.776	732899.01
967	935089	904231063	31.0966	9.8888	30070	3.9544	.00103413	3037.918	734417.18
968	937024	907039232	31.1127	9.8922	30117	3.9553	.00103306	3041.059	735936.93
969	938961	909853209	31.1288	9.8956	30164	3.9561	.00103199	3044.201	737458.24
970	**940900**	**912673000**	**31.1448**	**9.8990**	**30210**	**3.9569**	**.00103093**	**3047.342**	**738981.13**
971	942841	915498611	31.1609	9.9024	30257	3.9577	.00102987	3050.484	740505.59
972	944784	918330048	31.1769	9.9058	30304	3.9585	.00102881	3053.625	742031.62
973	946729	921167317	31.1929	9.9092	30351	3.9593	.00102775	3056.767	743559.22
974	948676	924010424	31.2090	9.9126	30398	3.9602	.00102669	3059.909	745088.39
975	950625	926859375	31.2250	9.9160	30444	3.9610	.00102564	3063.050	746619.13
976	952576	929714176	31.2410	9.9194	30491	3.9618	.00102459	3066.192	748151.44
977	954529	932574833	31.2570	9.9227	30538	3.9626	.00102354	3069.333	749685.32
978	956484	935441352	31.2730	9.9261	30585	3.9634	.00102249	3072.475	751220.78
979	958441	938313739	31.2890	9.9295	30632	3.9642	.00102145	3075.617	752757.80
980	**960400**	**941192000**	**31.3050**	**9.9329**	**30679**	**3.9650**	**.00102041**	**3078.758**	**754296.40**
981	962361	944076141	31.3209	9.9363	30726	3.9658	.00101937	3081.900	755836.59
982	964324	946966168	31.3369	9.9396	30773	3.9666	.00101833	3085.041	757378.30
983	966289	949862087	31.3528	9.9430	30820	3.9674	.00101729	3088.183	758921.61
984	968256	952763904	31.3688	9.9464	30867	3.9682	.00101626	3091.325	760466.48
985	970225	955671625	31.3847	9.9497	30914	3.9691	.00101523	3094.466	762012.93
986	972196	958585256	31.4006	9.9531	30961	3.9699	.00101420	3097.608	763560.95
987	974169	961504803	31.4166	9.9565	31008	3.9707	.00101317	3100.749	765110.54
988	976144	964430272	31.4325	9.9598	31055	3.9715	.00101215	3103.891	766661.70
989	978121	967361669	31.4484	9.9632	31102	3.9723	.00101112	3107.033	768214.44

Table 4.—Properties of Numbers—*Continued*

N	N^2	N^3	$\sqrt{N}$	$\sqrt[3]{N}$	$N^{3/2}$	$\sqrt[5]{N}$	$\frac{1}{N}$	Circle ($N = D$) Circum.	Circle ($N = D$) Area
990	**980100**	**970299000**	**31.4643**	**9.9666**	**31150**	**3.9731**	**.00101010**	**3110.174**	**769768.74**
991	982081	973242271	31.4802	9.9699	31197	3.9739	.00100908	3113.316	771324.61
992	984064	976191488	31.4960	9.9733	31244	3.9747	.00100806	3116.457	772882.06
993	986049	979146657	31.5119	9.9766	31291	3.9755	.00100705	3119.599	774441.07
994	988036	982107784	31.5278	9.9800	31339	3.9763	.00100604	3122.740	776001.66
995	990025	985074875	31.5436	9.9833	31386	3.9771	.00100503	3125.882	777563.82
996	992016	988047936	31.5595	9.9866	31433	3.9779	.00100402	3129.024	779127.54
997	994009	991026973	31.5753	9.9900	31480	3.9787	.00100301	3132.165	780692.84
998	996004	994011992	31.5911	9.9933	31528	3.9795	.00100200	3135.307	782259.71
999	998001	997002999	31.6070	9.9967	31575	3.9803	.00100100	3138.448	783828.15
1000	**1000000**	**1000000000**	**31.6228**	**10.0000**	**31623**	**3.9811**	**.00100000**	**3141.593**	**785398.16**

CIRCULAR ARCS, CHORDS, SEGMENTS AND SECTORS

Notation.—A = central angle, degrees; C = chord of arc; c = chord of 1/2 arc; D = diameter; G = area of segment; H = rise of arc or height of segment; L = length of arc; R = radius; S = area of sector.

L (exact) $= 2\pi RA/360 = 0.017453RA$

L (approx.) $= (8c - C)/3 = \{(2c \times 10H)/(60D - 27A)\} + 2c$

$= \{(\sqrt{C^2 + 4H^2} \times 10H^2)/(15C^2 + 33H^2)\} + 2c$

$C = 2\sqrt{c^2 - H^2} = \sqrt{D^2 - (D - 2H)^2} = (8c - 3L) = 2\sqrt{R^2 - (R - H)^2} = 2\sqrt{(D - H) \times H}$

$C/2 = \sqrt{H \times (D - H)}$; $c = 1/2\sqrt{C^2 - 4H^2} = \sqrt{D \times H} = (3L + C)/8$

$D = c^2/H = (1/4\,C^2 + H^2)/H$

$H = c^2/D = 1/2\,(D - \sqrt{D^2 - C^2})$ when $H < R$; $= 1/2\,(D + \sqrt{D^2 - C^2}) = \sqrt{c^2 - 1/4\,C^2}$ when $H > R$

$G = S - 1/2\,C\sqrt{R^2 - 1/4\,C^2}$ when $G <$ semi-circle; $= S + 1/2\,C\sqrt{R^2 - 1/4\,C^2}$ when $S >$ semi-circle

$= S - (C/2)(R - H) = 1/2\,\{(L - C)(D/2) + CH\}$

$S = 1/2\,LR = \pi AR^2/360 = (\pi R^2/360) \times 2[\sin^{-1}\{(C/2)/R\}]$; $R = (C^2 \cdot$

Table 5. Circular Arcs, Chords, and Segments. Radius = 1

Central Angle in Degrees	Arc Length L	Arc Rise H	Chord, Length C	$\frac{H}{C}$	Area of Segment* G	Central Angle in Degrees	Arc Length L	Arc Rise H	Chord, Length C	$\frac{H}{C}$	Area of Segment* G
1	0.0175	0.0000	0.0175	0.0022	0.00000	31	0.5411	0.0364	0.5345	0.0680	0.01301
2	.0349	.0002	.0349	.0044	.00000	32	.5585	.0387	.5513	.0703	.01429
3	.0524	.0003	.0524	.0066	.00001	33	.5760	.0412	.5680	.0725	.01566
4	.0698	.0006	.0698	.0087	.00003	34	.5934	.0437	.5847	.0747	.01711
5	.0873	.0010	.0872	.0109	.00006	35	.6109	.0463	.6014	.0770	.01864
6	.1047	.0014	.1047	.0131	.00010	36	.6283	.0489	.6180	.0792	.02027
7	.1222	.0019	.1221	.0153	.00015	37	.6458	.0517	.6346	.0814	.02198
8	.1396	.0024	.1395	.0175	.00023	38	.6632	.0545	.6511	.0837	.02378
9	.1571	.0031	.1569	.0196	.00032	39	.6807	.0574	.6676	.0859	.02568
10	**.1745**	**.0038**	**.1743**	**.0218**	**.00044**	**40**	**.6981**	**.0603**	**.6840**	**.0882**	**.02767**
11	.1920	.0046	.1917	.0240	.00059	41	.7156	.0633	.7004	.0904	.02976
12	.2094	.0055	.2091	.0262	.00076	42	.7330	.0664	.7167	.0927	.03195
13	.2269	.0064	.2264	.0284	.00097	43	.7505	.0696	.7330	.0949	.03425
14	.2443	.0075	.2437	.0306	.00121	44	.7679	.0728	.7492	.0972	.03664
15	.2618	.0086	.2611	.0328	.00149	45	.7854	.0761	.7654	.0995	.03915
16	.2793	.0097	.2783	.0350	.00181	46	.8029	.0795	.7815	.1017	.04176
17	.2967	.0110	.2956	.0372	.00217	47	.8203	.0829	.7975	.1040	.04448
18	.3142	.0123	.3129	.0394	.00257	48	.8378	.0865	.8135	.1063	.04731
19	.3316	.0137	.3301	.0415	.00302	49	.8552	.0900	.8294	.1086	.05025
20	**.3491**	**.0152**	**.3473**	**.0437**	**.00352**	**50**	**8727**	**.0937**	**.8452**	**.1108**	**.05331**
21	.3665	.0167	.3645	.0459	.00408	51	.8901	.0974	.8610	.1131	.05649
22	.3840	.0184	.3816	.0481	.00468	52	.9076	.1012	.8767	.1154	.05978
23	.4014	.0201	.3987	.0503	.00535	53	.9250	.1051	.8924	.1177	.06319
24	.4189	.0219	.4158	.0526	.00607	54	.9425	.1090	.9080	.1200	.06673
25	.4363	.0237	.4329	.0548	.00686	55	.9599	.1130	.9235	.1223	.07039
26	.4538	.0256	.4499	.0570	.00771	56	.9774	.1171	.9389	.1247	.07417
27	.4712	.0276	.4669	.0592	.00862	57	.9848	.1212	.9543	.1270	.07808
28	.4887	.0297	.4838	.0614	.00961	58	1.0123	.1254	.9696	.1293	.08212
29	.5061	.0319	.5008	.0636	.01067	59	1.0297	.1296	.9848	.1316	.08629
30	**.5236**	**.0341**	**.5176**	**.0658**	**.01180**	**60**	**1.0472**	**.1340**	**1.0000**	**.1340**	**.09059**

*Area of segment of any radius = factor G × square of radius

Table 5.—Circular Arcs, Chords and Segments—*Continued*

Central Angle in Degrees	Arc Length L	Arc Rise H	Chord, Length C	$\frac{H}{C}$	Area of Segment * G	Central Angle in Degrees	Arc Length L	Arc Rise H	Chord, Length C	$\frac{H}{C}$	Area of Segment* G
61	1.0647	0.1384	1.015	0.1363	0.09502	121	2.1118	0.5076	1.741	0.2916	0.62734
62	1.0821	.1428	1.030	.1387	.09958	122	2.1293	.5152	1.749	.2945	.64063
63	1.0996	.1474	1.045	.1410	.10428	123	2.1468	.5228	1.758	.2975	.65404
64	1.1170	.1520	1.060	.1434	.10911	124	2.1642	.5305	1.766	.3004	.66759
65	1.1345	.1566	1.075	.1457	.11408	125	2.1817	.5383	1.774	.3034	.68125
66	1.1519	.1613	1.089	.1481	.11919	126	2.1991	.5460	1.782	.3064	.69505
67	1.1694	.1661	1.104	.1505	.12443	127	2.2166	.5538	1.790	.3094	.70897
68	1.1868	.1710	1.118	.1529	.12982	128	2.2340	.5616	1.798	.3124	.72301
69	1.2043	.1759	1.133	.1553	.13535	129	2.2515	.5695	1.805	.3155	.73716
70	**1.2217**	**.1808**	**1.147**	**.1576**	**.14102**	**130**	**2.2689**	**.5774**	**1.813**	**.3185**	**.75144**
71	1.2392	.1859	1.161	.1601	.14683	131	2.2864	.5853	1.820	.3216	.76584
72	1.2566	.1910	1.176	.1625	.15279	132	2.3038	.5933	1.827	.3247	.78034
73	1.2741	.1961	1.190	.1649	.15889	133	2.3213	.6013	1.834	.3278	.79497
74	1.2915	.2014	1.204	.1673	.16514	134	2.3387	.6093	1.841	.3309	.80970
75	1.3090	.2066	1.218	.1697	.17154	135	2.3562	.6173	1.848	.3341	.82454
76	1.3265	.2120	1.231	.1722	.17808	136	2.3736	.6254	1.854	.3373	.83949
77	1.3439	.2174	1.245	.1746	.18477	137	2.3911	.6335	1.861	.3404	.85455
78	1.3614	.2229	1.259	.1771	.19160	138	2.4086	.6416	1.867	.3436	.86971
79	1.3788	2284	1.272	.1795	.19859	139	2.4260	.6498	1.873	.3469	.88497
80	**1.3963**	**.2340**	**1.286**	**.1820**	**.20573**	**140**	**2.4435**	**.6580**	**1.879**	**.3501**	**.90034**
81	1.4137	.2396	1.299	.1845	.21301	141	2.4609	.6662	1.885	.3534	.91580
82	1.4312	.2453	1.312	.1869	.22045	142	2.4784	.6744	1.891	.3566	.93135
83	1.4486	.2510	1.325	.1894	.22804	143	2.4958	.6827	1.897	.3599	.94700
84	1.4661	.2569	1.338	.1919	.23578	144	2.5133	.6910	1.902	.3633	.96274
85	1.4835	.2627	1.351	.1944	.24367	145	2.5307	.6993	1.907	.3666	.97858
86	1.5010	.2686	1.364	.1970	.25171	146	2.5482	.7076	1.913	.3700	.99449
87	1.5184	.2746	1.377	.1995	.25990	147	2.5656	.7160	1.918	.3734	1.0105
88	1.5359	.2807	1.389	.2020	.26825	148	2.5831	.7244	1.923	.3768	1.0266
89	1.5533	.2867	1.402	.2046	.27675	149	2.6005	.7328	1.927	.3802	1.0428
90	**1.5708**	**.2929**	**1.414**	**.2071**	**.28540**	**150**	**2.6180**	**.7412**	**1.932**	**.3837**	**1.0590**
91	1.5882	.2991	1.427	.2097	.29420	151	2.6354	.7496	1.936	.3871	1.0753
92	1.6057	.3053	1.439	.2122	.30316	152	2.6529	.7581	1.941	.3906	1.0917
93	1.6232	.3116	1.451	.2148	.31226	153	2.6704	.7666	1.945	.3942	1.1082
94	1.6406	.3180	1.463	.2174	.32152	154	2.6878	.7750	1.949	.3977	1.1247
95	1.6581	.3244	1.475	.2200	.33093	155	2.7053	.7836	1.953	.4013	1.1413
96	1.6755	.3309	1.486	.2226	.34050	156	2.7227	.7921	1.956	.4049	1.1580
97	1.6930	.3374	1.498	.2252	.35021	157	2.7402	.8006	1.960	.4085	1.1747
98	1.7104	.3439	1.509	.2279	.36008	158	2.7576	.8092	1.963	.4122	1.1915
99	1.7279	.3506	1.521	.2305	.37009	159	2.7751	.8178	1.967	.4158	1.2084
100	**1.7453**	**.3572**	**1.532**	**.2332**	**.38026**	**160**	**2.7925**	**.8264**	**1.970**	**.4195**	**1.2253**
101	1.7628	.3639	1.543	.2358	.39058	161	2.8100	.8350	1.973	.4233	1.2422
102	1.7802	.3707	1.554	.2385	.40104	162	2.8274	.8436	1.975	.4270	1.2592
103	1.7977	.3775	1.565	.2412	.41166	163	2.8449	.8522	1.978	.4308	1.2763
104	1.8151	.3843	1.576	.2439	.42242	164	2.8623	.8608	1.981	.4346	1.2934
105	1.8326	.3912	1.587	.2466	.43333	165	2.8798	.8695	1.983	.4385	1.3105
106	1.8500	.3982	1.597	.2493	.44439	166	2.8972	.8781	1.985	.4424	1.3277
107	1.8675	.4052	1.608	.2520	.45560	167	2.9147	.8868	1.987	.4463	1.3449
108	1.8850	.4122	1.618	.2548	.46695	168	2.9322	.8955	1.989	.4502	1.3621
109	1.9024	.4193	1.628	.2575	.47844	169	2.9496	.9042	1.991	.4542	1.3794
110	**1.9199**	**.4264**	**1.638**	**.2603**	**.49008**	**170**	**2.9671**	**.9128**	**1.992**	**.4582**	**1.3967**
111	1.9373	.4336	1.648	.2631	.50187	171	2.9845	.9215	1.994	.4622	1.4140
112	1.9548	.4408	1.658	.2659	.51379	172	3.0020	.9302	1.995	.4663	1.4314
113	1.9722	.4481	1.668	.2687	.52586	173	3.0194	.9390	1.996	.4704	1.4488
114	1.9897	.4554	1.677	.2715	.53806	174	3.0369	.9477	1.997	.4745	1.4662
115	2.0071	.4627	1.687	.2743	.55041	175	3.0543	.9564	1.998	.4786	1.4836
116	2.0246	.4701	1.696	.2772	.56289	176	3.0718	.9651	1.9988	.4828	1.5010
117	2.0420	.4775	1.705	.2800	.57551	177	3.0892	.9738	1.9993	.4871	1.5184
118	2.0595	.4850	1.714	.2829	.58827	178	3.1067	.9825	1.9997	.4913	1.5359
119	2.0769	.4925	1.723	.2858	.60116	179	3.1241	.9913	1.9999	.4957	1.5533
120	**2.0944**	**.5000**	**1.732**	**.2887**	**.61419**	**180**	**3.1416**	**1.0000**	**2.000**	**.5000**	**1.5708**

* Area of segment of any radius = factor G × square of radius

Table 6.—Areas of Segments of a Circle

Area $= F \times D^2$ where segment $<$ semicircle; $= \frac{1}{4}\pi D^2 -$ (area of segment of rise h). $H =$ ris or height of segment; $D =$ diam. of circle; $F =$ factor corresponding to H/D. $h = (D - H)$

H/D	F	H/D	F	H/D	F	H/D	F	H/D	F	H/D	F
0.000	0.00000	0.07	0.02417	0.14	0.06683	0.21	0.11990	0.28	0.18002	0.35	0.24498
.001	.00004	.071	.02468	.141	.06753	.211	.12071	.281	.18092	.351	.24593
.002	.00012	.072	.02520	.142	.06822	.212	.12153	.282	.18182	.352	.24689
.003	.00022	.073	.02571	.143	.06892	.213	.12235	.283	.18272	.353	.24784
.004	.00034	.074	.02624	.144	.06963	.214	.12317	.284	.18362	.354	.24880
.005	.00047	.075	.02676	.145	.07033	.215	.12399	.285	.18452	.355	.24976
.006	.00062	.076	.02729	.146	.07103	.216	.12481	.286	.18542	.256	.25071
.007	.00078	.077	.02782	.147	.07174	.217	.12563	.287	.18633	.357	.25167
.008	.00095	.078	.02836	.148	.07245	.218	.12646	.288	.18723	.358	.25263
.009	.00113	.079	.02889	.149	.07316	.219	.12729	.289	.18814	.359	.25359
.01	.00133	.08	.02943	.15	.07387	.22	.12811	.29	.18905	.36	.25455
.011	.00153	.081	.02998	.151	.07459	.221	.12894	.291	.18996	.361	.25551
.012	.00175	.082	.03053	.152	.07531	.222	.12977	.292	.19086	.362	.25647
.013	.00197	.083	.03108	.153	.07603	.223	.13060	.293	.19177	.363	.25743
.014	.00220	.084	.03163	.154	.07675	.224	.13144	.294	.19268	.364	.25839
.015	.00244	.085	.03219	.155	.07747	.225	.13227	.295	.19360	.365	.25936
.016	.00268	.086	.03275	.156	.07819	.226	.13311	.296	.19451	.366	.26032
.017	.00294	.087	.03331	.157	.07892	.227	.13395	.297	.19542	.367	.26128
.018	.00320	.088	.03387	.158	.07965	.228	.13478	.298	.19634	.368	.26225
.019	.00347	.089	.03444	.159	.08038	.229	.13562	.299	.19725	.369	.26321
.02	.00375	.09	.03501	.16	.08111	.23	.13646	.3	.19817	.37	.26418
.021	.00403	.091	.03559	.161	.08185	.231	.13731	.301	.19908	.371	.26514
.022	.00432	.092	.03616	.162	.08258	.232	.13815	.302	.20000	.372	.26611
.023	.00462	.093	.03674	.163	.08332	.233	.13900	.303	.20092	.373	.26708
.024	.00492	.094	.03732	.164	.08406	.234	.13984	.304	.20184	.374	.26805
.025	.00523	.095	.03791	.165	.08480	.235	.14069	.305	.20276	.375	.26901
.026	.00555	.096	.03850	.166	.08554	.236	.14154	.306	.20368	.376	.26998
.027	.00587	.097	.03909	.167	.08629	.237	.14239	.307	.20460	.377	.27095
.028	.00619	.098	.03968	.168	.08704	.238	.14324	.308	.20553	.378	.27192
.029	.00653	.099	.04028	.169	.08779	.239	.14409	.309	.20645	.379	.27289
.03	.00687	.1	.04087	.17	.08854	.24	.14494	.31	.20738	.38	.27386
.031	.00721	.101	.04148	.171	.08929	.241	.14580	.311	.20830	.381	.27483
.032	.00756	.102	.04208	.172	.09004	.242	.14666	.312	.20923	.382	.27580
.033	.00791	.103	.04269	.173	.09080	.243	.14751	.313	.21015	.383	.27678
.034	.00827	.104	.04330	.174	.09155	.244	.14837	.314	.21108	.384	.27775
.035	.00864	.105	.04391	.175	.09231	.245	.14923	.315	.21201	.385	.27872
.036	.00901	.106	.04452	.176	.09307	.246	.15009	.316	.21294	.386	.27969
.037	.00938	.107	.04514	.177	.09384	.247	.15095	.317	.21387	.387	.28067
.038	.00976	.108	.04576	.178	.09460	.248	.15182	.318	.21480	.388	.28164
.039	.01015	.109	.04638	.179	.09537	.249	.15268	.319	.21573	.389	.28262
.04	.01054	.11	.04701	.18	.09613	.25	.15355	.32	.21667	.39	.28359
.041	.01093	.111	.04763	.181	.09690	.251	.15441	.321	.21760	.391	.28457
.042	.01133	.112	.04826	.182	.09767	.252	.15528	.322	.21853	.392	.28554
.043	.01173	.113	.04889	.183	.09845	.253	.15615	.323	.21947	.393	.28652
.044	.01214	.114	.04953	.184	.09922	.254	.15702	.324	.22040	.394	.28750
.045	.01255	.115	.05016	.185	.10000	.255	.15789	.325	.22134	.395	.28848
.046	.01297	.116	.05080	.186	.10077	.256	.15876	.326	.22228	.396	.28945
.047	.01339	.117	.05145	.187	.10155	.257	.15964	.327	.22322	.397	.29043
.048	.01382	.118	.05209	.188	.10233	.258	.16051	.328	.22415	.398	.29141
.049	.01425	.119	.05274	.189	.10312	.259	.16139	.329	.22509	.399	.29239
.05	.01468	.12	.05338	.19	.10390	.26	.16226	.33	.22603	.4	.29337
.051	.01512	.121	.05404	.191	.10469	.261	.16314	.331	.22697	.401	.29435
.052	.01556	.122	.05469	.192	.10547	.262	.16402	.332	.22792	.402	.29533
.053	.01601	.123	.05535	.193	.10626	.263	.16490	.333	.22886	.403	.29631
.054	.01646	.124	.05600	.194	.10705	.264	.16578	.334	.22980	.404	.29729
.055	.01691	.125	.05666	.195	.10784	.265	.16666	.335	.23074	.405	.29827
.056	.01737	.126	.05733	.196	.10864	.266	.16755	.336	.23169	.406	.29926
.057	.01783	.127	.05799	.197	.10943	.267	.16843	.337	.23263	.407	.30024
.058	.01830	.128	.05866	.198	.11023	.268	.16932	.338	.23358	.408	.30122
.059	.01877	.129	.05933	.199	.11102	.269	.17020	.339	.23453	.409	.30220
.06	.01924	.13	.06000	.2	.11182	.27	.17109	.34	.23547	.41	.30319
.061	.01972	.131	.06067	.201	.11262	.271	.17198	.341	.23642	.411	.30417
.062	.02020	.132	.06135	.202	.11343	.272	.17287	.342	.23737	.412	.30516
.063	.02068	.133	.06203	.203	.11423	.273	.17376	.343	.23832	.413	.30614
.064	.02117	.134	.06271	.204	.11504	.274	.17465	.344	.23927	.414	.30712
.065	.02166	.135	.06339	.205	.11584	.275	.17554	.345	.24022	.415	.30811
.066	.02215	.136	.06407	.206	.11665	.276	.17644	.346	.24117	.416	.30910
.067	.02265	.137	.06476	.207	.11746	.277	.17733	.347	.24212	.417	.31008
.068	.02315	.138	.06545	.208	.11827	.278	.17823	.348	.24307	.418	.31107
.069	.02366	.139	.06614	.209	.11908	.279	.17912	.349	.24403	.419	.31205

(Table continued on following page)

Table 6.—Area of Segments—*Continued*

H/D	F	H/D	F	H/D	F	H/D	F	H/D	F	H/D	F
0.42	0.31304	0.435	0.32788	0.45	0.34278	0.465	0.35773	0.48	0.37270	0.495	0.38770
.421	.31403	.436	.32887	.451	.34378	.466	.35873	.481	.37370	.496	.38870
.422	.31502	.437	.32987	.452	.34477	.467	.35972	.482	.37470	.497	.38970
.423	.31600	.438	.33086	.453	.34577	.468	.36072	.483	.37570	.498	.39070
.424	.31699	.439	.33185	.454	.34676	.469	.36172	.484	.37670	.499	.39170
.425	.31798	.44	.33284	.455	.34776	.47	.36272	.485	.37770	.500	.39270
.426	.31897	.441	.33384	.456	.34876	.471	.36372	.486	.37870		
.427	.31996	.442	.33483	.457	.34975	.472	.36471	.487	.37970		
.428	.32095	.443	.33582	.458	.35075	.473	.36571	.488	.38070		
.429	.32194	.444	.33682	.459	.35175	.474	.36671	.489	.38170		
.43	.32293	.445	.33781	.46	.35274	.475	.36771	.49	.38270		
.431	.32392	.446	.33880	.461	.35374	.476	.36871	.491	.38370		
.432	.32491	.447	.33980	.462	.35474	.477	.36971	.492	.38470		
.433	.32590	.448	.34079	.463	.35573	.478	.37071	.493	.38570		
.434	.32689	.449	.34179	.464	.35673	.479	.37171	.494	.38670		

Table 7.—Values of Degrees and Minutes in Radians

Deg	Radians	Deg	Radians	Deg	Radians	Deg	Radians	Deg	Radians	Deg	Radians	Min	Radians	Min	Radians
1	0.01745	31	0.54105	61	1.06465	91	1.58825	121	2.11185	151	2.63545	1	0.00029	31	0.00902
2	.03491	32	.55851	62	1.08210	92	1.60570	122	2.12930	152	2.65290	2	.00058	32	.00931
3	.05236	33	.57596	63	1.09956	93	1.62316	123	2.14676	153	2.67035	3	.00087	33	.00960
4	.06981	34	.59341	64	1.11701	94	1.64061	124	2.16421	154	2.68781	4	.00116	34	.00989
5	.08727	35	.61087	65	1.13446	95	1.65806	125	2.18166	155	2.70526	5	.00145	35	.01018
6	.10472	36	.62832	66	1.15192	96	1.67552	126	2.19912	156	2.72271	6	.00175	36	.01047
7	.12217	37	.64577	67	1.16937	97	1.69297	127	2.21657	157	2.74017	7	.00204	37	.01076
8	.13963	38	.66323	68	1.18682	98	1.71042	128	2.23402	158	2.75762	8	.00233	38	.01105
9	.15708	39	.68068	69	1.20428	99	1.72788	129	2.25148	159	2.77507	9	.00262	39	.01134
10	.17453	40	.69813	70	1.22173	100	1.74533	130	2.26893	160	2.79253	10	.00291	40	.01164
11	.19199	41	.71559	71	1.23918	101	1.76278	131	2.28638	161	2.80998	11	.00320	41	.01193
12	.20944	42	.73304	72	1.25664	102	1.78024	132	2.30384	162	2.82743	12	.00349	42	.01222
13	.22689	43	.75049	73	1.27409	103	1.79769	133	2.32129	163	2.84489	13	.00378	43	.01251
14	.24435	44	.76795	74	1.29154	104	1.81514	134	2.33874	164	2.86234	14	.00407	44	.01280
15	.26180	45	.78540	75	1.30900	105	1.83260	135	2.35620	165	2.87979	15	.00436	45	.01309
16	.27925	46	.80285	76	1.32645	106	1.85005	136	2.37365	166	2.89725	16	.00465	46	.01338
17	.29671	47	.82031	77	1.34390	107	1.86750	137	2.39110	167	2.91470	17	.00495	47	.01367
18	.31416	48	.83776	78	1.36136	108	1.88496	138	2.40856	168	2.93215	18	.00524	48	.01396
19	.33161	49	.85521	79	1.37881	109	1.90241	139	2.42601	169	2.94961	19	.00553	49	.01425
20	.34907	50	.87267	80	1.39626	110	1.91986	140	2.44346	170	2.96706	20	.00582	50	.01454
21	.36652	51	.89012	81	1.41372	111	1.93732	141	2.46092	171	2.98451	21	.00611	51	.01484
22	.38397	52	.90757	82	1.43117	112	1.95477	142	2.47837	172	3.00197	22	.00640	52	.01513
23	.40143	53	.92502	83	1.44862	113	1.97222	143	2.49582	173	3.01942	23	.00669	53	.01542
24	.41888	54	.94248	84	1.46608	114	1.98968	144	2.51328	174	3.03687	24	.00698	54	.01571
25	.43633	55	.95993	85	1.48353	115	2.00713	145	2.53073	175	3.05433	25	.00727	55	.01600
26	.45379	56	.97738	86	1.50098	116	2.02458	146	2.54818	176	3.07178	26	.00756	56	.01629
27	.47124	57	.99484	87	1.51844	117	2.04204	147	2.56564	177	3.08923	27	.00785	57	.01658
28	.48869	58	1.01299	88	1.53589	118	2.05949	148	2.58309	178	3.10669	28	.00814	58	.01687
29	.50615	59	1.02974	89	1.55334	119	2.07694	149	2.60054	179	3.12414	29	.00844	59	.01716
30	.52360	60	1.04720	90	1.57080	120	2.09440	150	2.61800	180	3.14159	30	.00873	60	.01745

Table 8.—Values of Radians in Degrees

Rad.	.00	.01	.02	.03	.04	.05	.06	.07	.08	.09
	Deg.	Deg.	Deg.	Deg.	Deg.	Deg.	Deg.	Deg.	Deg.	Deg.
0.0	0.0000	0.5730	1.1459	1.7189	2.2918	2.8648	3.4377	4.0107	4.5837	5.1566
.1	5.7296	6.3025	6.8755	7.4485	8.0214	8.5944	9.1673	9.7403	10.3132	10.8862
.2	11.4591	12.0321	12.6051	13.1780	13.7510	14.3239	14.8969	15.4699	16.0428	16.6158
.3	17.1887	17.7617	18.3346	18.9076	19.4806	20.0535	20.6265	21.1994	21.7724	22.3454
.4	22.9183	23.4913	24.0642	24.6372	25.2101	25.7831	26.3561	26.9290	27.5020	28.0749
.5	28.6479	29.2208	29.7938	30.3668	30.9397	31.1527	32.0856	32.6586	33.2316	33.8045
.6	34.3775	34.9504	35.5234	36.0963	36.6693	37.2423	37.8152	38.3882	38.9611	39.5341
.7	40.1070	40.6800	41.2530	41.8259	42.3989	42.9718	43.5448	44.1178	44.6907	45.2637
.8	45.8366	46.4096	46.9825	47.5555	48.1285	48.7014	49.2744	49.8473	50.4203	50.9932
.9	51.5662	52.1392	52.7121	53.2851	53.8580	54.4310	55.0039	55.5769	56.1499	56.7228

1 Radian = 57.29578 deg. | 2 Radians = 114.59156 deg. | 3 Radians = 171.88734 deg.

Table 9.—Decimals of a Degree in Minutes and Seconds

Decimal	.00		.01		.02		.03		.04		.05		.06		.07		.08		.09	
	Min.	Sec.	Min.	Sec.	Min.	Sec.	Min.	Sec.	Min.	Sec.	Min.	Sec.	Min.	Sec.	Min.	Sec.	Min.	Sec.	Min.	Sec.
0.0	0	0	0	36	1	12	1	48	2	24	3	0	3	36	4	12	4	48	5	24
.1	6	0	6	36	7	12	7	48	8	24	9	0	9	36	10	12	10	48	11	24
.2	12	0	12	36	13	12	13	48	14	24	15	0	15	36	16	12	16	48	17	24
.3	18	0	18	36	19	12	19	48	20	24	21	0	21	36	22	12	22	48	23	24
.4	24	0	24	36	25	12	25	48	26	24	27	0	27	36	28	12	28	48	29	24
.5	30	0	30	36	31	12	31	48	32	24	33	0	33	36	34	12	24	48	35	24
.6	36	0	36	36	37	12	37	48	38	24	39	0	39	36	40	12	40	48	41	24
.7	42	0	42	36	43	12	43	48	44	24	45	0	45	36	46	12	46	48	47	24
.8	48	0	48	36	49	12	49	48	50	24	51	0	51	36	52	12	52	48	53	24
.9	54	0	54	36	55	12	55	48	56	24	57	0	57	36	58	12	58	48	59	24

Table 10.—Circumferences and Areas of Circles—Diameters in Feet and Inches

Circumferences in feet; areas in square feet

Diam., feet	Diameter, inches 0	1	2	3	4	5	6	7	8	9	10	11
	Circumference											
0	0.0000	0.2618	0.5236	0.7854	1.0472	1.3090	1.5708	1.8326	2.0944	2.3562	2.6180	2.8798
1	3.1416	3.4034	3.6653	3.9270	4.1877	4.4507	4.7124	4.9741	5.2361	5.4978	5.7595	6.0215
2	6.2832	6.5450	6.8069	7.0686	7.3293	7.5923	7.8540	8.1157	8.3777	8.6394	8.9011	9.1631
3	9.4248	9.6866	9.9485	10.210	10.471	10.734	10.996	11.257	11.519	11.781	12.043	12.305
4	12.566	12.828	13.090	13.352	13.613	13.875	14.137	14.399	14.661	14.923	15.184	15.446
5	15.708	15.970	16.232	16.493	16.754	17.017	17.279	17.540	17.802	18.064	18.326	18.588
6	18.850	19.111	19.373	19.635	19.896	20.159	20.420	20.682	20.944	21.206	21.467	21.729
7	21.991	22.253	22.515	22.777	23.037	23.300	23.562	23.824	24.086	24.347	24.609	24.871
8	25.133	25.395	25.656	25.918	26.179	26.442	26.704	26.965	27.227	27.489	27.751	28.013
9	28.274	28.536	28.798	29.060	29.320	29.583	29.845	30.107	30.369	30.631	30.892	31.154
10	31.416	31.678	31.940	32.201	32.462	32.725	32.987	33.248	33.510	33.772	34.034	34.296
11	34.557	34.819	35.081	35.343	35.604	35.867	36.128	36.390	36.652	36.914	37.175	37.437
12	37.699	37.961	38.223	38.484	38.745	39.008	39.270	39.532	39.794	40.055	40.317	40.579
13	40.841	41.102	41.364	41.626	41.887	42.150	42.411	42.673	42.935	43.197	43.459	43.721
14	43.982	44.244	44.506	44.768	45.028	45.291	45.553	45.815	46.077	46.338	46.600	46.862
15	47.124	47.386	47.648	47.909	48.170	48.433	48.695	48.956	49.218	49.480	49.742	50.004
16	50.265	50.527	50.789	51.051	51.312	51.575	51.836	52.098	52.360	52.622	52.883	53.145
17	53.407	53.669	53.931	54.192	54.453	54.716	54.978	55.240	55.502	55.763	56.025	56.287
18	56.549	56.810	57.072	57.334	57.595	57.858	58.119	58.381	58.643	58.905	59.167	59.429
19	59.690	59.952	60.214	60.476	60.736	61.000	61.261	61.523	61.785	62.046	62.308	62.570
20	62.832	63.094	63.356	63.617	63.878	64.141	64.403	64.664	64.926	65.188	65.450	65.712
21	65.973	66.235	66.497	66.759	67.020	67.282	67.544	67.806	68.068	68.330	68.591	68.853
22	69.115	69.377	69.639	69.900	70.161	70.424	70.686	70.947	71.209	71.471	71.733	71.995
23	72.257	72.518	72.780	73.042	73.303	73.566	73.827	74.089	74.351	74.613	74.874	75.136
24	75.398	75.660	75.922	76.184	76.444	76.707	76.969	77.231	77.493	77.754	78.016	78.278
25	78.540	78.802	79.063	79.325	79.586	79.849	80.111	80.372	80.634	80.896	81.158	81.420
26	81.681	81.943	82.205	82.467	82.727	82.990	83.252	83.514	83.776	84.038	84.299	84.561
27	84.823	85.085	85.347	85.608	85.869	86.132	86.394	86.655	86.917	87.179	87.441	87.703
28	87.965	88.226	88.488	88.749	89.011	89.274	89.535	89.797	90.059	90.321	90.582	90.844
29	91.106	91.367	91.630	91.892	92.152	92.415	92.677	92.939	93.201	93.462	93.724	93.986
30	94.248	94.509	94.771	95.033	95.294	95.557	95.818	96.080	96.342	96.604	96.866	97.128

0	0.0000	0.0055	0.0218	0.0491	0.0873	0.1364	0.1963	0.2673	0.3491	0.4418	0.5454	0.6600
1	0.7854	0.9218	1.069	1.227	1.396	1.576	1.767	1.969	2.182	2.405	2.640	2.885
2	3.142	3.409	3.687	3.976	4.276	4.587	4.909	5.241	5.585	5.940	6.305	6.681
3	7.069	7.467	7.876	8.296	8.727	9.168	9.621	10.08	10.56	11.04	11.54	12.05
4	12.57	13.10	13.64	14.19	14.75	15.32	15.90	16.50	17.10	17.72	18.35	18.99
5	19.63	20.29	20.97	21.65	22.34	23.04	23.76	24.48	25.22	25.97	26.73	27.49
6	28.27	29.07	29.87	30.68	31.50	32.34	33.18	34.04	34.91	35.78	36.67	37.57
7	38.48	39.41	40.34	41.28	42.24	43.20	44.18	45.17	46.16	47.17	48.19	49.22
8	50.27	51.32	52.38	53.46	54.54	55.64	56.75	57.86	58.99	60.13	61.28	62.44
9	63.62	64.80	66.00	67.20	68.42	69.64	70.88	72.13	73.39	74.66	75.94	77.24
10	78.54	79.85	81.18	82.52	83.86	85.22	86.59	87.97	89.36	90.76	92.18	93.60
11	95.03	96.48	97.93	99.40	100.9	102.4	103.9	105.4	106.9	108.4	110.0	111.5
12	113.1	114.7	116.3	117.9	119.5	121.1	122.7	124.4	126.0	127.7	129.4	131.0
13	132.7	134.4	136.2	137.9	139.6	141.4	143.1	144.9	146.7	148.5	150.3	152.1
14	153.9	155.8	157.6	159.5	161.4	163.2	165.1	167.0	168.9	170.9	172.8	174.8
15	176.7	178.7	180.7	182.7	184.7	186.7	188.7	190.7	192.8	194.8	195.9	199.0
16	201.1	203.2	205.3	207.4	209.5	211.7	213.8	216.0	218.2	220.4	222.5	224.8
17	227.0	229.2	231.4	233.7	236.0	238.2	240.5	242.8	245.1	247.5	249.8	252.1
18	254.5	256.8	259.2	261.6	264.0	266.4	268.8	271.2	273.7	276.1	278.6	281.0
19	283.5	286.0	288.5	291.0	293.6	296.1	298.7	301.2	303.8	306.4	308.9	311.5
20	314.2	316.8	319.4	322.1	324.7	327.4	330.1	332.7	335.4	338.2	340.9	343.6
21	346.4	349.1	351.9	354.7	357.4	360.2	363.1	365.9	368.7	371.5	374.4	377.2
22	380.1	383.0	386.0	388.8	391.7	394.7	397.6	400.5	403.5	406.5	409.5	412.5
23	415.5	418.5	421.5	424.6	427.6	430.7	433.7	436.8	439.9	443.0	446.1	449.2
24	452.4	455.5	458.7	461.9	465.0	468.2	471.4	474.6	477.9	481.1	484.3	487.6
25	490.9	494.1	497.4	500.7	504.0	507.4	510.7	514.0	517.4	520.8	524.1	527.5
26	530.9	534.3	537.7	541.2	544.6	548.1	551.6	555.0	558.5	562.0	565.5	569.0
27	572.6	576.1	579.6	583.2	586.8	590.3	594.0	597.5	601.2	604.8	608.4	612.1
28	615.8	619.4	623.1	626.8	630.5	634.2	637.9	641.7	645.4	649.2	652.9	656.7
29	660.5	664.3	668.1	672.0	675.8	679.6	683.5	687.3	691.2	695.1	699.0	702.9
30	706.9	710.8	714.7	718.7	722.6	726.6	730.6	734.6	738.6	742.6	746.7	750.7

Table 11.—Natural Trigonometrical Functions

Deg.	Min.	Sine	Covers	Cosec	Tan	Cotan	Secant	Versin	Cosine		
0	0	0.00000	1.00000	Infinite	0.00000	Infinite	1.0000	0.00000	1.00000	**90**	0
	15	.00436	.99564	229.18	.00436	229.18	1.0000	.00001	.99999		45
	30	.00873	.99127	114.59	.00873	114.59	1.0000	.00004	.99996		30
	45	.01309	.98691	76.397	.01309	76.390	1.0001	.00009	.99991		15
1	0	**.01745**	**.98255**	**57.299**	**.01745**	**57.290**	**1.0001**	**.00015**	**.99985**	**89**	0
	15	.02181	.97819	45.840	.02182	45.829	1.0002	.00024	.99976		45
	30	.02618	.97382	38.202	.02618	38.188	1.0003	.00034	.99966		30
	45	.03054	.96946	32.746	.03055	32.730	1.0005	.00047	.99953		15
2	0	**.03490**	**.96510**	**28.654**	**.03492**	**28.636**	**1.0006**	**.00061**	**.99939**	**88**	0
	15	.03926	.96074	25.471	.03929	25.452	1.0008	.00077	.99923		45
	30	.04362	.95638	22.926	.04366	22.904	1.0009	.00095	.99905		30
	45	.04798	.95202	20.843	.04803	20.819	1.0011	.00115	.99885		15
3	0	**.05234**	**.94766**	**19.107**	**.05241**	**19.081**	**1.0014**	**.00137**	**.99863**	**87**	0
	15	.05669	.94331	17.639	.05678	17.611	1.0016	.00161	.99839		45
	30	.06105	.93895	16.380	.06116	16.350	1.0019	.00187	.99813		30
	45	.06540	.93460	15.290	.06554	15.257	1.0021	.00214	.99786		15
4	0	**.06976**	**.93024**	**14.336**	**.06993**	**14.301**	**1.0024**	**.00244**	**.99756**	**86**	0
	15	.07411	.92589	13.494	.07431	13.457	1.0028	.00275	.99725		45
	30	.07846	.92154	12.745	.07870	12.706	1.0031	.00308	.99692		30
	45	.08281	.91719	12.076	.08309	12.035	1.0034	.00343	.99656		15
5	0	**.08716**	**.91284**	**11.474**	**.08749**	**11.430**	**1.0038**	**.00381**	**.99619**	**85**	0
	15	.09150	.90850	10.929	.09189	10.883	1.0042	.00420	.99580		45
	30	.09585	.90415	10.433	.09629	10.385	1.0046	.00460	.99540		30
	45	.10019	.89981	9.9812	.10069	9.9310	1.0051	.00503	.99497		15
6	0	**.10453**	**.89547**	**9.5668**	**.10510**	**9.5144**	**1.0055**	**.00548**	**.99452**	**84**	0
	15	.10887	.89113	9.1855	.10952	9.1309	1.0060	.00594	.99406		45
	30	.11320	.88680	8.8337	.11393	8.7769	1.0065	.00643	.99357		30
	45	.11754	.88246	8.5079	.11836	8.4490	1.0070	.00693	.99307		15
7	0	**.12187**	**.87813**	**8.2055**	**.12278**	**8.1443**	**1.0075**	**.00745**	**.99255**	**83**	0
	15	.12620	.87380	7.9240	.12722	7.8606	1.0081	.00800	.99200		45
	30	.13053	.86947	7.6613	.13165	7.5958	1.0086	.00856	.99144		30
	45	.13485	.86515	7.4156	.13609	7.3479	1.0092	.00913	.99086		15
8	0	**.13917**	**.86083**	**7.1853**	**.14054**	**7.1154**	**1.0098**	**.00973**	**.99027**	**82**	0
	15	.14349	.85651	6.9690	.14499	6.8969	1.0105	.01035	.98965		45
	30	.14781	.85219	6.7655	.14945	6.6912	1.0111	.01098	.98902		30
	45	.15212	.84788	6.5736	.15391	6.4971	1.0118	.01164	.98836		15
9	0	**.15643**	**.84357**	**6.3924**	**.15838**	**6.3138**	**1.0125**	**.01231**	**.98769**	**81**	0
	15	.16074	.83926	6.2211	.16286	6.1402	1.0132	.01300	.98700		45
	30	.16505	.83495	6.0589	.16734	5.9758	1.0139	.01371	.98629		30
	45	.16935	.83065	5.9049	.17183	5.8197	1.0147	.01444	.98556		15
10	0	**.17365**	**.82635**	**5.7588**	**.17633**	**5.6713**	**1.0154**	**.01519**	**.98481**	**80**	0
	15	.17794	.82206	5.6198	.18083	5.5301	1.0162	.01596	.98404		45
	30	.18224	.81776	5.4874	.18534	5.3955	1.0170	.01675	.98325		30
	45	.18652	.81348	5.3612	.18986	5.2672	1.0179	.01755	.98245		15
11	0	**.19081**	**.80919**	**5.2408**	**.19438**	**5.1446**	**1.0187**	**.01837**	**.98163**	**79**	0
	15	.19509	.80491	5.1258	.19891	5.0273	1.0196	.01921	.98079		45
	30	.19937	.80063	5.0158	.20345	4.9152	1.0205	.02008	.97992		30
	45	.20364	.79636	4.9106	.20800	4.8077	1.0214	.02095	.97905		15
12	0	**.20791**	**.79209**	**4.8097**	**.21256**	**4.7046**	**1.0223**	**.02185**	**.97815**	**78**	0
	15	.21218	.78782	4.7130	.21712	4.6057	1.0233	.02277	.97723		45
	30	.21644	.78356	4.6202	.22169	4.5107	1.0243	.02370	.97630		30
	45	.22070	.77930	4.5311	.22628	4.4194	1.0253	.02466	.97534		15
13	0	**.22495**	**.77505**	**4.4454**	**.23087**	**4.3315**	**1.0263**	**.02563**	**.97437**	**77**	0
	15	.22920	.77080	4.3630	.23547	4.2468	1.0273	.02662	.97338		45
	30	.23345	.76655	4.2837	.24008	4.1653	1.0284	.02763	.97237		30
	45	.23769	.76231	4.2072	.24470	4.0867	1.0295	.02866	.97134		15
14	0	**.24192**	**.75808**	**4.1336**	**.24933**	**4.0108**	**1.0306**	**.02970**	**.97030**	**76**	0
	15	.24615	.75385	4.0625	.25397	3.9375	1.0317	.03077	.96923		45
	30	.25038	.74962	3.9939	.25862	3.8667	1.0329	.03185	.96815		30
	45	.25460	.74540	3.9277	.26328	3.7983	1.0341	.03295	.96705		15
15	0	**.25882**	**.74118**	**3.8637**	**.26795**	**3.7320**	**1.0353**	**.03407**	**.96593**	**75**	0
		Cosine	Versin	Secant	Cotan	Tan	Cosec	Covers	Sine	Deg.	Min.

From 75° to 90° read from bottom of table upwards.

Table 11.—Natural Trigonometrical Functions—*Continued*

Deg.	Min.	Sine	Covers	Cosec	Tan	Cotan	Secant	Versin	Cosine		
15	**0**	**0.25882**	**0.74118**	**3.8637**	**0.26795**	**3.7320**	**1.0353**	**0.03407**	**0.96593**	**75**	**0**
	15	.26303	.73697	3.8018	.27263	3.6680	1.0365	.03521	.96479		45
	30	.26724	.73276	3.7420	.27732	3.6059	1.0377	.03637	.96363		30
	45	.27144	.72856	3.6840	.28203	3.5457	1.0390	.03754	.96246		15
16	**0**	**.27564**	**.72436**	**3.6280**	**.28674**	**3.4874**	**1.0403**	**.03874**	**.96126**	**74**	**0**
	15	.27983	.72017	3.5736	.29147	3.4308	1.0416	.03995	.96005		45
	30	.28402	.71598	3.5209	.29621	3.3759	1.0429	.04118	.95882		30
	45	.28820	.71180	3.4699	.30096	3.3226	1.0443	.04243	.95757		15
17	**0**	**.29237**	**.70763**	**3.4203**	**.30573**	**3.2709**	**1.0457**	**.04370**	**.95630**	**73**	**0**
	15	.29654	.70346	3.3722	.31051	3.2205	1.0471	.04498	.95502		45
	30	.30070	.69929	3.3255	.31530	3.1716	1.0485	.04628	.95372		30
	45	.30486	.69514	3.2801	.32010	3.1240	1.0500	.04760	.95240		15
18	**0**	**.30902**	**.69098**	**3.2361**	**.32492**	**3.0777**	**1.0515**	**.04894**	**.95106**	**72**	**0**
	15	.31316	.68684	3.1932	.32975	3.0326	1.0530	.05030	.94970		45
	30	.31730	.68270	3.1515	.33459	2.9887	1.0545	.05168	.94832		30
	45	.32144	.67856	3.1110	.33945	2.9459	1.0560	.05307	.94693		15
19	**0**	**.32557**	**.67443**	**3.0715**	**.34433**	**2.9042**	**1.0576**	**.05448**	**.94552**	**71**	**0**
	15	.32969	.67031	3.0331	.34921	2.8636	1.0592	.05591	.94409		45
	30	.33381	.66619	2.9957	.35412	2.8239	1.0608	.05736	.94264		30
	45	.33792	.66208	2.9593	.35904	2.7852	1.0625	.05882	.94118		15
20	**0**	**.34202**	**.65798**	**2.9238**	**.36397**	**2.7475**	**1.0642**	**.06031**	**.93969**	**70**	**0**
	15	.34612	.65388	2.8892	.36892	2.7106	1.0659	.06181	.93819		45
	30	.35021	.64979	2.8554	.37388	2.6746	1.0676	.06333	.93667		30
	45	.35429	.64571	2.8225	.37887	2.6395	1.0694	.06486	.93514		15
21	**0**	**.35837**	**.64163**	**2.7904**	**.38386**	**2.6051**	**1.0711**	**.06642**	**.93358**	**69**	**0**
	15	.36244	.63756	2.7591	.38888	2.5715	1.0729	.06799	.93201		45
	30	.36650	.63350	2.7285	.39391	2.5386	1.0748	.06958	.93042		30
	45	.37056	.62944	2.6986	.39896	2.5065	1.0766	.07119	.92881		15
22	**0**	**.37461**	**.62539**	**2.6695**	**.40403**	**2.4751**	**1.0785**	**.07282**	**.92718**	**68**	**0**
	15	.37865	.62135	2.6410	.40911	2.4443	1.0804	.07446	.92554		45
	30	.38268	.61732	2.6131	.41421	2.4142	1.0824	.07612	.92388		30
	45	.38671	.61329	2.5859	.41933	2.3847	1.0844	.07780	.92220		15
23	**0**	**.39073**	**.60927**	**2.5593**	**.42447**	**2.3559**	**1.0864**	**.07950**	**.92050**	**67**	**0**
	15	.39474	.60526	2.5333	.42963	2.3276	1.0884	.08121	.91879		45
	30	.39875	.60125	2.5078	.43481	2.2998	1.0904	.08294	.91706		30
	45	.40275	.59725	2.4829	.44001	2.2727	1.0925	.08469	.91531		15
24	**0**	**.40674**	**.59326**	**2.4586**	**.44523**	**2.2460**	**1.0946**	**.08645**	**.91355**	**66**	**0**
	15	.41072	.58928	2.4348	.45047	2.2199	1.0968	.08824	.91176		45
	30	.41469	.58531	2.4114	.45573	2.1943	1.0989	.09004	.90996		30
	45	.41866	.58134	2.3886	.46101	2.1692	1.1011	.09186	.90814		15
25	**0**	**.42262**	**.57738**	**2.3662**	**.46631**	**2.1445**	**1.1034**	**.09369**	**.90631**	**65**	**0**
	15	.42657	.57343	2.3443	.47163	2.1203	1.1056	.09554	.90446		45
	30	.43051	.56949	2.3228	.47697	2.0965	1.1079	.09741	.90259		30
	45	.43445	.56555	2.3018	.48234	2.0732	1.1102	.09930	.90070		15
26	**0**	**.43837**	**.56163**	**2.2812**	**.48773**	**2.0503**	**1.1126**	**.10121**	**.89879**	**64**	**0**
	15	.44229	.55771	2.2610	.49314	2.0278	1.1150	.10313	.89687		45
	30	.44620	.55380	2.2412	.49858	2.0057	1.1174	.10507	.89493		30
	45	.45010	.54990	2.2217	.50404	1.9840	1.1198	.10702	.89298		15
27	**0**	**.45399**	**.54601**	**2.2027**	**.50952**	**1.9626**	**1.1223**	**.10899**	**.89101**	**63**	**0**
	15	.45787	.54213	2.1840	.51503	1.9416	1.1248	.11098	.88902		45
	30	.46175	.53825	2.1657	.52057	1.9210	1.1274	.11299	.88701		30
	45	.46561	.53439	2.1477	.52612	1.9007	1.1300	.11501	.88499		15
28	**0**	**.46947**	**.53053**	**2.1300**	**.53171**	**1.8807**	**1.1326**	**.11705**	**.88295**	**62**	**0**
	15	.47332	.52668	2.1127	.53732	1.8611	1.1352	.11911	.88089		45
	30	.47716	.52284	2.0957	.54295	1.8418	1.1379	.12118	.87882		30
	45	.48099	.51901	2.0790	.54862	1.8228	1.1406	.12327	.87673		15
29	**0**	**.48481**	**.51519**	**2.0627**	**.55431**	**1.8040**	**1.1433**	**.12538**	**.87462**	**61**	**0**
	15	.48862	.51138	2.0466	.56003	1.7856	1.1461	.12750	.87250		45
	30	.49242	.50758	2.0308	.56577	1.7675	1.1490	.12964	.87036		30
	45	.49622	.50378	2.0152	.57155	1.7496	1.1518	.13180	.86820		15
30	**0**	**.50000**	**.50000**	**2.0000**	**.57735**	**1.7320**	**1.1547**	**.13397**	**.86603**	**60**	**0**
		Cosine	Versin	Secant	Cotan	Tan	Cosec	Covers	Sine	Deg.	Min.

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Table 11.—Natural Trigonometrical Functions—*Continued*

Deg.	Min.	Sine	Covers	Cosec	Tan	Cotan	Secant	Versin	Cosine		
30	**0**	**0.50000**	**0.50000**	**2.0000**	**0.57735**	**1.7320**	**1.1547**	**0.13397**	**0.86603**	**60**	**0**
	15	.50377	.49623	1.9850	.58318	1.7147	1.1576	.13616	.86384		45
	30	.50754	.49246	1.9703	.58904	1.6977	1.1606	.13837	.86163		30
	45	.51129	.48871	1.9558	.59494	1.6808	1.1636	.14059	.85941		15
31	**0**	**.51504**	**.48496**	**1.9416**	**.60086**	**1.6643**	**1.1666**	**.14283**	**.85717**	**59**	**0**
	15	.51877	.48123	1.9276	.60681	1.6479	1.1697	.14509	.85491		45
	30	.52250	.47750	1.9139	.61280	1.6319	1.1728	.14736	.85264		30
	45	.52621	.47379	1.9004	.61882	1.6160	1.1760	.14965	.85035		15
32	**0**	**.52992**	**.47008**	**1.8871**	**.62487**	**1.6003**	**1.1792**	**.15195**	**.84805**	**58**	**0**
	15	.53361	.46639	1.8740	.63095	1.5849	1.1824	.15427	.84573		45
	30	.53730	.46270	1.8612	.63707	1.5697	1.1857	.15661	.84339		30
	45	.54097	.45903	1.8485	.64322	1.5547	1.1890	.15896	.84104		15
33	**0**	**.54464**	**.45536**	**1.8361**	**.64941**	**1.5399**	**1.1924**	**.16133**	**.83867**	**57**	**0**
	15	.54829	.45171	1.8238	.65563	1.5253	1.1958	.16371	.83629		45
	30	.55194	.44806	1.8118	.66188	1.5108	1.1992	.16611	.83389		30
	45	.55557	.44443	1.7999	.66818	1.4966	1.2027	.16853	.83147		15
34	**0**	**.55919**	**.44081**	**1.7883**	**.67451**	**1.4826**	**1.2062**	**.17096**	**.82904**	**56**	**0**
	15	.56280	.43720	1.7768	.68087	1.4687	1.2098	.17341	.82659		45
	30	.56641	.43359	1.7655	.68728	1.4550	1.2134	.17587	.82413		30
	45	.57000	.43000	1.7544	.69372	1.4415	1.2171	.17835	.82165		15
35	**0**	**.57358**	**.42642**	**1.7434**	**.70021**	**1.4281**	**1.2208**	**.18085**	**.81915**	**55**	**0**
	15	.57715	.42285	1.7327	.70673	1.4150	1.2245	.18336	.81664		45
	30	.58070	.41930	1.7220	.71329	1.4019	1.2283	.18588	.81412		30
	45	.58425	.41575	1.7116	.71990	1.3891	1.2322	.18843	.81157		15
36	**0**	**.58779**	**.41221**	**1.7013**	**.72654**	**1.3764**	**1.2361**	**.19098**	**.80902**	**54**	**0**
	15	.59131	.40869	1.6912	.73323	1.3638	1.2400	.19356	.80644		45
	30	.59482	.40518	1.6812	.73996	1.3514	1.2440	.19614	.80386		30
	45	.59832	.40168	1.6713	.74673	1.3392	1.2480	.19875	.80125		15
37	**0**	**.60181**	**.39819**	**1.6616**	**.75355**	**1.3270**	**1.2521**	**.20136**	**.79864**	**53**	**0**
	15	.60529	.39471	1.6521	.76042	1.3151	1.2563	.20400	.79600		45
	30	.60876	.39124	1.6427	.76733	1.3032	1.2605	.20665	.79335		30
	45	.61222	.38778	1.6334	.77428	1.2915	1.2647	.20931	.79069		15
38	**0**	**.61566**	**.38434**	**1.6243**	**.78129**	**1.2799**	**1.2690**	**.21199**	**.78801**	**52**	**0**
	15	.61909	.38091	1.6153	.78834	1.2685	1.2734	.21468	.78532		45
	30	.62251	.37749	1.6064	.79543	1.2572	1.2778	.21739	.78261		30
	45	.62592	.37408	1.5976	.80258	1.2460	1.2822	.22012	.77988		15
39	**0**	**.62932**	**.37068**	**1.5890**	**.80978**	**1.2349**	**1.2868**	**.22285**	**.77715**	**51**	**0**
	15	.63271	.36729	1.5805	.81703	1.2239	1.2913	.22561	.77439		45
	30	.63608	.36392	1.5721	.82434	1.2131	1.2960	.22838	.77162		30
	45	.63944	.36056	1.5639	.83169	1.2024	1.3007	.23116	.76884		15
40	**0**	**.64279**	**.35721**	**1.5557**	**.83910**	**1.1918**	**1.3054**	**.23396**	**.76604**	**50**	**0**
	15	.64612	.35388	1.5477	.84656	1.1812	1.3102	.23677	.76323		45
	30	.64945	.35055	1.5398	.85408	1.1708	1.3151	.23959	.76041		30
	45	.65276	.34724	1.5320	.86165	1.1606	1.3200	.24244	.75756		15
41	**0**	**.65606**	**.34394**	**1.5242**	**.86929**	**1.1504**	**1.3250**	**.24529**	**.75471**	**49**	**0**
	15	.65935	.34065	1.5166	.87698	1.1403	1.3301	.24816	.75184		45
	30	.66262	.33738	1.5092	.88472	1.1303	1.3352	.25104	.74896		30
	45	.66588	.33412	1.5018	.89253	1.1204	1.3404	.25394	.74606		15
42	**0**	**.66913**	**.33087**	**1.4945**	**.90040**	**1.1106**	**1.3456**	**.25686**	**.74314**	**48**	**0**
	15	.67237	.32763	1.4873	.90834	1.1009	1.3509	.25978	.74022		45
	30	.67559	.32441	1.4802	.91633	1.0913	1.3563	.26272	.73728		30
	45	.67880	.32120	1.4732	.92439	1.0818	1.3618	.26568	.73432		15
43	**0**	**.68200**	**.31800**	**1.4663**	**.93251**	**1.0724**	**1.3673**	**.26865**	**.73135**	**47**	**0**
	15	.68518	.31482	1.4595	.94071	1.0630	1.3729	.27163	.72837		45
	30	.68835	.31165	1.4527	.94896	1.0538	1.3786	.27463	.72537		30
	45	.69151	.30849	1.4461	.95729	1.0446	1.3843	.27764	.72236		15
44	**0**	**.69466**	**.30534**	**1.4396**	**.96569**	**1.0355**	**1.3902**	**.28066**	**.71934**	**46**	**0**
	15	.69779	.30221	1.4331	.97416	1.0265	1.3961	.28370	.71630		45
	30	.70091	.29909	1.4267	.98270	1.0176	1.4020	.28675	.71325		30
	45	.70401	.29599	1.4204	.99131	1.0088	1.4081	.28981	.71019		15
45	**0**	**.70711**	**.29289**	**1.4142**	**1.00000**	**1.0000**	**1.4142**	**.29289**	**.70711**	**45**	**0**
		Cosine	Versin	Secant	Cotan	Tan	Cosec	Covers	Sine	Deg.	Min.

From 45° to 60° read from bottom of table upwards.

Table 12.—Logarithmic Trigonometric Functions

Deg.	Sine	Cosec	Versin	Tangent	Cotan	Covers	Secant	Cosine	Deg
0	− ∞	+ ∞	− ∞	− ∞	+ ∞	10.00000	10.00000	10.00000	90
1	8.24186	11.75814	6.18271	8.24192	11.75808	9.99235	10.00007	9.99993	89
2	8.54282	11.45718	6.78474	8.54308	11.45692	9.98457	10.00026	9.99974	88
3	8.71880	11.28120	7.13687	8.71940	11.28060	9.97665	10.00060	9.99940	87
4	8.84358	11.15642	7.38667	8.84464	11.15536	9.96860	10.00106	9.99894	86
5	8.94030	11.05970	7.58039	8.94195	11.05805	9.96040	10.00166	9.99834	85
6	9.01923	10.98077	7.73863	9.02162	10.97838	9.95205	10.00239	9.99761	84
7	9.08589	10.91411	7.87238	9.08914	10.91086	9.94356	10.00325	9.99675	83
8	9.14356	10.85644	7.98820	9.14780	10.85220	9.93492	10.00425	9.99575	82
9	9.19433	10.80567	8.09032	9.19971	10.80029	9.92612	10.00538	9.99462	81
10	9.23967	10.76033	8.18162	9.24632	10.75368	9.91717	10.00665	9.99335	80
11	9.28060	10.71940	8.26418	9.28865	10.71135	9.90805	10.00805	9.99195	79
12	9.31788	10.68212	8.33950	9.32747	10.67253	9.89877	10.00960	9.99040	78
13	9.35209	10.64791	8.40875	9.36336	10.63664	9.88933	10.01128	9.98872	77
14	9.38368	10.61632	8.47282	9.39677	10.60323	9.87971	10.01310	9.98690	76
15	9.41300	10.58700	8.53243	9.42805	10.57195	9.86992	10.01506	9.98494	75
16	9.44034	10.55966	8.58814	9.45750	10.54250	9.85996	10.01716	9.98284	74
17	9.46594	10.53406	8.64043	9.48534	10.51466	9.84981	10.01940	9.98060	73
18	9.48998	10.51002	8.68969	9.51178	10.48822	9.83947	10.02179	9.97821	72
19	9.51264	10.48736	8.73625	9.53697	10.46303	9.82894	10.02433	9.97567	71
20	9.53405	10.46595	8.78037	9.56107	10.43893	9.81821	10.02701	9.97299	70
21	9.55433	10.44567	8.82230	9.58418	10.41582	9.80729	10.02985	9.97015	69
22	9.57358	10.42642	8.86223	9.60641	10.39359	9.79615	10.03283	9.96717	68
23	9.59188	10.40812	8.90034	9.62785	10.37215	9.78481	10.03597	9.96403	67
24	9.60931	10.39069	8.93679	9.64858	10.35142	9.77325	10.03927	9.96073	66
25	9.62595	10.37405	8.97170	9.66867	10.33133	9.76146	10.04272	9.95728	65
26	9.64184	10.35816	9.00521	9.68818	10.31182	9.74945	10.04634	9.95366	64
27	9.65705	10.34295	9.03740	9.70717	10.29283	9.73720	10.05012	9.94988	63
28	9.67161	10.32839	9.06838	9.72567	10.27433	9.72471	10.05407	9.94593	62
29	9.68557	10.31443	9.09823	9.74375	10.25625	9.71197	10.05818	9.94182	61
30	9.69897	10.30103	9.12702	9.76144	10.23856	9.69897	10.06247	9.93753	60
31	9.71184	10.28816	9.15483	9.77877	10.22123	9.68571	10.06693	9.93307	59
32	9.72421	10.27579	9.18171	9.79579	10.20421	9.67217	10.07158	9.92842	58
33	9.73611	10.26389	9.20771	9.81252	10.18748	9.65836	10.07641	9.92359	57
34	9.74756	10.25244	9.23290	9.82899	10.17101	9.64425	10.08143	9.91857	56
35	9.75859	10.24141	9.25731	9.84523	10.15477	9.62984	10.08664	9.91336	55
36	9.76922	10.23078	9.28099	9.86126	10.13874	9.61512	10.09204	9.90796	54
37	9.77946	10.22054	9.30398	9.87711	10.12289	9.60008	10.09765	9.90235	53
38	9.78934	10.21066	9.32631	9.89281	10.10719	9.58471	10.10347	9.89653	52
39	9.79887	10.20113	9.34802	9.90837	10.09163	9.56900	10.10950	9.89050	51
40	9.80807	10.19193	9.36913	9.92381	10.07619	9.55293	10.11575	9.88425	50
41	9.81694	10.18306	9.38968	9.93916	10.06084	9.53648	10.12222	9.87778	49
42	9.82551	10.17449	9.40969	9.95444	10.04556	9.51966	10.12893	9.87107	48
43	9.83378	10.16622	9.42918	9.96966	10.03034	9.50243	10.13587	9.86413	47
44	9.84177	10.15823	9.44818	9.98484	10.01516	9.48479	10.14307	9.85693	46
45	9.84949	10.15052	9.46671	10.00000	10.00000	9.46671	10.15052	9.84949	45
	Cosine	Secant	Covers	Cotan	Tangent	Versin	Cosec	Sine	

From 45° to 90° read from bottom of table upwards.

Table 13.—Fractions Expressed as Decimals of an Inch and of 1 Foot

Inches	1	2	3	4	5	6	7	8	9	10	11
Decimal of 1 ft.	0.0833	0.1667	0.2500	0.3333	0.4167	0.5000	0.5833	0.6667	0.7500	0.8333	0.9167

Fractions of 1 inch

Fraction	Decimal of 1 in.	Decimal of 1 ft.	Fraction	Decimal of 1 in.	Decimal of 1 ft.	Fraction	Decimal of 1 in.	Decimal of 1 ft.	Fraction	Decimal of 1 in.	Decimal of 1 ft.
1/64	0.015625	0.0013	17/64	0.265625	0.0221	33/64	0.515625	0.0430	49/64	0.765625	0.0638
1/32	.03125	.0026	9/32	.28125	.0234	17/32	.53125	.0443	25/32	.78125	.0651
3/64	.046875	.0039	19/64	.296875	.0247	35/64	.546875	.0456	51/64	.796875	.0664
1/16	.0625	.0052	5/16	.3125	.0260	9/16	.5625	.0469	13/16	.8125	.0677
5/64	.078125	.0065	21/64	.328125	.0273	37/64	.578125	.0482	53/64	.828125	.0690
3/32	.09375	.0078	11/32	.34375	.0286	19/32	.59375	.0495	27/32	.84375	.0703
7/64	.109375	.0091	23/64	.359375	.0299	39/64	.609375	.0508	55/64	.859375	.0716
1/8	.125	.0104	3/8	.375	.0313	5/8	.625	.0521	7/8	.875	.0729
9/64	.140625	.0117	25/64	.390625	.0326	41/64	.640625	.0534	57/64	.890625	.0742
5/32	.15625	.0130	13/32	.40625	.0339	21/32	.65625	.0547	29/32	.90625	.0755
11/64	.171875	.0143	27/64	.421875	.0352	43/64	.671875	.0560	59/64	.921875	.0768
3/16	.1875	.0156	7/16	.4375	.0365	11/16	.6875	.0573	15/16	.9375	.0781
13/64	.203125	.0169	29/64	.453125	.0378	45/64	.703125	.0586	61/64	.953125	.0794
7/32	.21875	.0182	15/32	.46875	.0391	23/32	.71875	.0599	31/32	.96875	.0807
15/64	.234375	.0195	31/64	.484375	.0404	47/64	.734375	.0612	63/64	.984375	.0820
1/4	.25	.0208	1/2	.50	.0417	3/4	.75	.0625	....	....	......

EXAMPLE: 9 19/32 in. = 0.7500 + 0.0495 = 0.7995 ft.

FLOW AND CAPACITY EQUIVALENTS

Flow and capacity equivalents, Tables 14 and 15, and the tables of capacity of tanks and cisterns, Tables 16 to 18, are based on the following values.

1 cubic foot = 7.480519 U.S. gallons
1 U.S. gallon = 231 cu. in. = 0.13368056 cu. ft.
1 lb. of water at 62° F. = 0.01603489 cu. ft.

The British Imperial gallon = 277.418 cu. in. (U.S.) = 277.420 cu. in. (British) = 1.20094 U.S. gallons.

The flow equivalents of Table 14 are based on water at 62° F. These flow equivalents for a given weight of water can be converted to flow equivalents for any other fluid by multiplying by the ratio of the density of that fluid to the density of water. For example, the flow per minute of 20 lb. of oil of specific gravity 0.90 (water = 1.0) is

$$(1/0.90) \times 20 \times 0.1200 = 2.67 \text{ gal. per min.}$$

Table 14.—Flow Equivalents

	Cu. ft. per sec.	Cu. ft. per min.	U.S. Gallons per min.	U.S. Gals. per hr.	U.S. Gallons per 24 hr.	Lb. H_2O per min.	Lb. H_2O per hr.
1 cu. ft. per sec. =	1	60	448.83	26,929.9	646,316.8	3741.00	224,460
1 cu. ft. per min. =	1/60	1	7.4805	448.83	10,771.2	62.35	3741.00
1 gal. per min. =	0.002228	0.1337	1	60	1440	8.3350	500.1
1 gal. per hr. =	3713×10^{-5}	0.002280	1/60	1	24	0.1389	8.3350
1 gal. per 24 hr. =	1547×10^{-6}	9283×10^{-5}	6944×10^{-4}	1/24	1	0.005788	0.3473
1 lb. H_2O per min. =	2673×10^{-3}	0.01604	0.1200	7.1986	172.7658	1	60
1 lb. H_2O per hr. =	4455×10^{-5}	2673×10^{-3}	0.0020	0.1200	2.8794	1/60	1

Table 15.—Gallon and Cubic Feet Equivalents

Gallons or cu. ft.	Gallons equivalent to cu. ft.	Cu. ft. equivalent to gallons	Gallons or cu. ft.	Gallons equivalent to cu. ft.	Cu. ft. equivalent to gallons	Gallons or cu. ft.	Gallons equivalent to cu. ft.	Cu. ft. equivalent to gallons
0.1	0.75	0.0134	100	748.0	13.368	100,000	748,051.9	13,368.1
.2	1.50	.0267	200	1,496.1	26.736	200,000	1,496,103.8	26,736.1
.3	2.24	.0401	300	2,244.2	40.104	300,000	2,244,155.7	40,104.2
.4	2.99	.0535	400	2,992.2	53.472	400,000	2,992,207.6	53,472.2
.5	3.74	.0668	500	3,740.3	66.840	500,000	3,740,259.5	66,840.3
.6	4.49	.0802	600	4,488.3	80.208	600,000	4,488,311.4	80,208.3
.7	5.24	.0936	700	5,236.4	93.576	700,000	5,236,363.3	93,576.4
.8	5.98	.1069	800	5,984.4	106.944	800,000	5,984,415.2	106,944.4
.9	6.73	.1203	900	6,732.5	120.312	900,000	6,732,467.1	120,312.5
1.0	7.48	.134	1,000	7,480.5	133.681	1,000,000	7,480,519	133,680.6
2	14.96	.267	2,000	14,961.0	267.361	2,000,000	14,961,038	267,361.1
3	22.44	.401	3,000	22,441.6	401.042	3,000,000	22,441,557	401,041.7
4	29.92	.535	4,000	29,922.1	534.722	4,000,000	29,922,076	534,722.2
5	37.40	.668	5,000	37,402.6	668.403	5,000,000	37,402,595	668,402.8
6	44.88	.802	6,000	44,883.1	802.083	6,000,000	44,883,114	802,083.3
7	52.36	.936	7,000	52,363.6	935.764	7,000,000	52,363,633	935,763.9
8	59.84	1.069	8,000	59,844.2	1,069.444	8,000,000	59,844,152	1,069,444
9	67.32	1.203	9,000	67,324.7	1,203.125	9,000,000	67,324,671	1,203,125
10	74.80	1.337	10,000	74,805.2	1,336.806	10,000,000	74,805,190	1,336,806
20	149.6	2.674	20,000	149,610.4	2,673.61	20,000,000	149,610,380	2,673,611
30	224.4	4.010	30,000	224,415.6	4,010.42	30,000,000	224,415,570	4,010,417
40	299.2	5.347	40,000	299,220.8	5,347.22	40,000,000	299,220,760	5,347,222
50	374.0	6.684	50,000	374,025.9	6,684.03	50,000,000	374,025,950	6,684,028
60	448.8	8.021	60,000	448,831.1	8,020.83	60,000,000	448,831,140	8,020,833
70	523.6	9.358	70,000	523,636.3	9,357.64	70,000,000	523,636,330	9,357,639
80	598.4	10.694	80,000	598,441.5	10,694.44	80,000,000	598,441,520	10,694,444
90	673.2	12.031	90,000	673,246.7	12,031.25	90,000,000	673,246,710	12,031,250

Table 16.—Capacity of Vertical Cylindrical Tanks per 1 ft. Depth

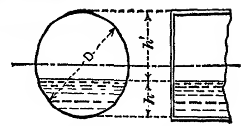
1 U.S. gallon = 231 cu. in. = 0.13368056 cu. ft.; 1 cu. ft. = 7.480519 gal.
1 barrel = 31.5 gal. = 7276.5 cu. in. = 4.2109 cu. ft.
For capacity of pipes, see pp. 5–34 to 5–39.

Diameter, ft.	in.	Capacity, Cu. ft.	U.S. gal.	Bbl.	Diameter, ft.	in.	Capacity, Cu. ft.	U.S. gal.	Bbl.	Diameter, ft.	in.	Capacity, Cu. ft.	U.S. gal.	Bbl.
1	0	0.785	5.87	0.186	18	0	254.47	1903.6	60.43	35	0	962.11	7,197.1	228.48
1	3	1.227	9.18	.291	18	3	261.59	1956.8	62.12	35	3	975.91	7,300.3	231.76
1	6	1.767	13.22	.420	18	6	268.80	2010.8	63.83	35	6	989.80	7,404.2	235.05
1	9	2.405	17.99	.571	18	9	276.12	2065.5	65.57	35	9	1003.8	7,508.9	238.38
2	0	3.142	23.50	.746	19	0	283.53	2120.9	67.33	36	0	1017.9	7,614.4	241.73
2	3	3.976	29.74	.944	19	3	291.04	2177.1	69.11	36	3	1032.1	7,720.6	245.10
2	6	4.909	36.72	1.17	19	6	298.65	2234.0	70.92	36	6	1046.3	7,826.9	248.47
2	9	5.940	44.43	1.41	19	9	306.35	2291.7	72.75	36	9	1060.7	7,934.6	251.89
3	0	7.069	52.88	1.68	20	0	314.16	2350.1	74.61	37	0	1075.2	8,043.1	255.34
3	3	8.296	62.06	1.97	20	3	322.06	2409.2	76.48	37	3	1089.8	8,152.3	258.80
3	6	9.621	71.97	2.28	20	6	330.06	2469.1	78.38	37	6	1104.5	8,262.2	262.29
3	9	11.045	82.62	2.62	20	9	338.16	2529.6	80.30	37	9	1119.2	8,372.2	265.78
4	0	12.566	94.00	2.98	21	0	346.36	2591.0	82.25	38	0	1134.1	8,483.7	269.32
4	3	14.186	106.12	3.37	21	3	354.66	2653.0	84.22	38	3	1149.1	8,595.9	272.89
4	6	15.904	118.97	3.78	21	6	363.05	2715.8	86.22	38	6	1164.2	8,708.8	276.47
4	9	17.721	132.56	4.21	21	9	371.54	2779.3	88.23	38	9	1179.3	8,821.8	280.06
5	0	19.635	146.88	4.66	22	0	380.13	2843.6	90.27	39	0	1194.6	8,936.2	283.69
5	3	21.648	161.93	5.14	22	3	388.82	2908.6	92.34	39	3	1210.0	9,051.4	287.35
5	6	23.758	177.72	5.64	22	6	397.61	2974.3	94.42	39	6	1225.4	9,166.6	291.00
5	9	25.967	194.25	6.17	22	9	406.49	3040.8	96.53	39	9	1241.0	9,283.3	294.71
6	0	28.274	211.51	6.71	23	0	415.48	3108.0	98.67	40	0	1256.6	9,400.0	298.41
6	3	30.680	229.50	7.29	23	3	424.56	3175.9	100.82	40	3	1272.4	9,518.2	302.16
6	6	33.183	248.23	7.88	23	6	433.74	3244.6	103.00	40	6	1288.2	9,636.4	305.92
6	9	35.785	267.69	8.50	23	9	443.01	3314.0	105.21	40	9	1304.2	9,756.1	309.72
7	0	38.485	287.88	9.14	24	0	452.39	3384.1	107.43	41	0	1320.3	9,876.5	313.54
7	3	41.282	308.81	9.80	24	3	461.86	3455.0	109.68	41	3	1336.4	9,997.0	317.36
7	6	44.179	330.48	10.49	24	6	471.44	3526.6	111.96	41	6	1352.7	10,118.9	321.23
7	9	47.173	352.88	11.20	24	9	481.11	3598.9	114.25	41	9	1369.0	10,240.8	325.10
8	0	50.265	376.01	11.94	25	0	490.87	3672.0	116.57	42	0	1385.4	10,363.5	329.00
8	3	53.456	399.88	12.69	25	3	500.74	3745.8	118.91	42	3	1402.0	10,487.7	332.94
8	6	56.745	424.48	13.48	25	6	510.71	3820.3	121.28	42	6	1418.6	10,611.9	336.89
8	9	60.132	449.82	14.28	25	9	520.77	3895.6	123.67	42	9	1435.4	10,737.5	340.87
9	0	63.617	475.89	15.11	26	0	530.93	3971.6	126.08	43	0	1452.2	10,863.2	344.86
9	3	67.201	502.70	15.96	26	3	541.19	4048.4	128.52	43	3	1469.1	10,989.6	348.88
9	6	70.882	530.24	16.83	26	6	551.55	4125.9	130.98	43	6	1486.2	11,117.5	352.94
9	9	74.662	558.51	17.73	26	9	562.00	4204.1	133.46	43	9	1503.3	11,245.5	357.00
10	0	78.540	587.52	18.65	27	0	572.56	4283.0	135.97	44	0	1520.5	11,374.1	361.08
10	3	82.516	617.26	19.60	27	3	583.21	4362.7	138.50	44	3	1537.9	11,504.3	365.22
10	6	86.590	647.74	20.56	27	6	593.96	4443.1	141.05	44	6	1555.3	11,634.5	369.35
10	9	90.763	678.95	21.55	27	9	604.81	4524.3	143.63	44	9	1572.8	11,765.4	373.50
11	0	95.033	710.90	22.57	28	0	615.75	4606.2	146.23	45	0	1590.4	11,897.0	377.68
11	3	99.402	743.58	23.61	28	3	626.80	4688.8	148.85	45	3	1608.2	12,030.2	381.91
11	6	103.87	776.99	24.67	28	6	637.94	4772.1	151.50	45	6	1626.0	12,163.3	386.14
11	9	108.43	811.14	25.75	28	9	649.18	4856.2	154.16	45	9	1643.9	12,297.2	390.39
12	0	113.10	846.03	26.86	29	0	660.52	4941.0	156.86	46	0	1661.9	12,431.9	394.66
12	3	117.86	881.65	27.99	29	3	671.96	5026.6	159.57	46	3	1680.0	12,567.3	398.96
12	6	122.72	918.00	29.14	29	6	683.49	5112.9	162.31	46	6	1698.2	12,703.4	403.28
12	9	127.68	955.09	30.32	29	9	695.13	5199.9	165.08	46	9	1716.5	12,840.3	407.63
13	0	132.73	992.91	31.52	30	0	706.86	5287.7	167.86	47	0	1734.9	12,978.0	412.00
13	3	137.89	1031.5	32.75	30	3	718.69	5376.2	170.67	47	3	1753.5	13,117.1	416.42
13	6	143.14	1070.8	33.99	30	6	730.62	5465.4	173.50	47	6	1772.1	13,256.2	420.83
13	9	148.49	1110.8	35.26	30	9	742.64	5555.4	176.36	47	9	1790.8	13,396.1	425.27
14	0	153.94	1151.5	36.55	31	0	754.77	5646.1	179.24	48	0	1809.6	13,536.7	429.74
14	3	159.48	1193.0	37.87	31	3	766.99	5737.5	182.14	48	3	1828.5	13,678.1	434.22
14	6	165.13	1235.3	39.22	31	6	779.31	5829.7	185.07	48	6	1847.5	13,820.3	438.74
14	9	170.87	1278.2	40.58	31	9	791.73	5922.6	188.02	48	9	1866.5	13,962.4	443.25
15	0	176.71	1321.9	41.97	32	0	804.25	6016.2	190.99	49	0	1885.7	14,106.0	447.81
15	3	182.65	1366.4	43.38	32	3	816.86	6110.6	193.99	49	3	1905.0	14,250.4	452.39
15	6	188.69	1411.5	44.81	32	6	829.58	6205.7	197.01	49	6	1924.4	14,395.5	457.00
15	9	194.83	1457.4	46.27	32	9	842.39	6301.5	200.05	49	9	1943.9	14,541.4	461.63
16	0	201.06	1504.1	47.75	33	0	855.30	6398.1	203.11	50	0	1963.5	14,688.0	466.29
16	3	207.39	1551.4	49.25	33	3	868.31	6495.4	206.20	50	3	1983.2	14,835.4	470.96
16	6	213.82	1599.5	50.78	33	6	881.41	6593.4	209.31	50	6	2003.0	14,983.5	475.67
16	9	220.35	1648.4	52.33	33	9	894.62	6692.2	212.45	50	9	2022.8	15,131.6	480.37
17	0	226.98	1697.9	53.90	34	0	907.92	6791.7	215.61	51	0	2042.8	15,281.2	485.12
17	3	233.71	1748.2	55.50	34	3	921.32	6892.0	218.79	51	3	2062.9	15,431.6	489.89
17	6	240.53	1799.3	57.12	34	6	934.82	6992.9	222.00	51	6	2083.1	15,582.7	494.69
17	9	247.45	1851.1	58.77	34	9	948.42	7094.7	225.23	51	9	2103.3	15,733.8	499.49

Table 17.—Contents of Horizontal Cylindrical Tanks When Filled to Various Depths

Tanks with flat ends. Contents given in U. S. gallons per 1 foot of length

Diam. of Tank, in.	Full Tank	Depth of Liquid, h, inches: 3	6	9	12	15	18	21	24	27	30	33	36	39	42	45	48	51	54	57	60	63	66	69	72
12	5.88	1.15	2.94	.....	.....	.....	.....	.....	.....	.....	.....	.....	.....	.....	.....	.....	.....	.....	.....						
18	13.22	1.45	3.86	6.61	.....	.....	.....	.....	.....	.....	.....	.....	.....	.....	.....	.....	.....	.....	.....						
24	23.50	1.70	4.60	8.05	11.75	.....	.....	.....	.....	.....	.....	.....	.....	.....	.....	.....	.....	.....	.....						
30	36.72	1.91	5.23	9.27	13.72	18.36	.....	.....	.....	.....	.....	.....	.....	.....	.....	.....	.....	.....	.....						
36	52.88	2.12	5.79	10.34	15.43	20.85	26.44	.....	.....	.....	.....	.....	.....	.....	.....	.....	.....	.....	.....						
42	71.97	2.28	6.31	11.31	16.97	23.07	29.47	35.99	.....	.....	.....	.....	.....	.....	.....	.....	.....	.....	.....						
48	94.01	2.45	6.78	12.20	18.38	25.10	32.20	39.54	47.00	.....	.....	.....	.....	.....	.....	.....	.....	.....	.....						
54	118.98	2.60	7.22	13.04	19.68	26.97	34.72	42.80	51.08	59.49	.....	.....	.....	.....	.....	.....	.....	.....	.....						
60	146.89	2.75	7.64	13.82	20.91	28.72	37.06	45.82	54.87	64.11	73.44	.....	.....	.....	.....	.....	.....	.....	.....						
66	177.73	2.89	8.04	14.56	22.07	30.37	39.28	48.65	58.39	68.41	78.59	88.86	.....	.....	.....	.....	.....	.....	.....						
72	211.52	3.02	8.42	15.26	23.17	31.92	41.36	51.32	61.71	72.45	83.41	94.54	105.76	.....	.....	.....	.....	.....	.....						
78	248.24	3.15	8.78	15.94	24.21	33.41	43.34	53.86	64.87	76.27	87.97	99.90	111.97	124.13	.....	.....	.....	.....	.....						
84	287.90	3.26	9.12	16.57	25.24	34.85	45.24	56.29	67.87	79.91	92.30	104.98	117.85	130.87	143.95	.....	.....	.....	.....						
90	330.49	3.43	9.46	17.20	26.20	36.21	47.05	58.61	70.75	83.39	96.43	109.81	123.45	137.28	151.23	165.25	.....	.....	.....	.....	.....	.....	.....	.....	.....
96	376.02	3.50	9.79	17.80	27.13	37.52	48.81	60.84	73.52	86.73	100.39	114.44	128.79	143.40	158.17	173.06	188.01	.....	.....	.....	.....	.....	.....	.....	.....
102	424.50	3.61	10.10	18.37	28.01	39.00	50.49	62.99	76.18	89.94	104.20	118.89	133.92	149.25	164.81	180.53	196.37	212.25	.....	.....	.....	.....	.....	.....	.....
108	476.10	3.71	10.39	18.94	28.90	40.03	52.14	65.09	78.74	93.04	107.87	123.17	138.87	154.89	171.19	187.71	204.37	221.14	238.05	.....	.....	.....	.....	.....	.....
114	530.25	3.78	10.74	19.49	29.75	41.22	53.73	67.10	81.24	96.05	111.43	127.31	143.63	160.33	177.33	194.60	212.05	229.65	247.37	265.13	.....	.....	.....	.....	.....
120	587.54	3.91	10.98	20.02	30.57	42.39	55.26	69.06	83.65	98.95	114.87	131.32	148.25	165.58	183.27	201.24	219.46	237.87	256.43	275.08	293.77	.....	.....	.....	.....
126	647.74	4.01	11.27	20.54	31.38	43.50	56.76	70.96	86.00	101.78	118.21	135.20	152.71	170.66	189.00	207.67	226.61	245.79	265.16	284.65	304.24	323.87	.....	.....	.....
132	355.45	4.11	11.53	21.03	32.17	44.62	58.22	72.82	88.28	104.53	121.44	138.98	157.06	175.61	194.58	213.91	233.56	253.48	273.61	293.93	314.36	334.88	355.45	.....	.....
138	777.00	4.20	11.80	21.54	32.93	45.70	59.64	74.62	90.50	107.20	124.60	142.66	161.28	180.42	200.00	219.98	240.31	260.94	281.81	302.90	324.15	345.54	367.00	388.50	.....
144	846.04	4.29	12.06	22.02	33.67	46.77	61.03	76.38	92.68	109.81	127.69	146.24	165.40	185.11	205.27	225.88	246.88	268.18	289.78	311.62	333.67	355.87	378.18	400.59	423.02



To ascertain the contents of a tank over one-half full: Let h' = depth of unfilled portion. Find from the table the quantity corresponding to a depth h. Subtract this quantity from the contents of a full tank.

Table 18.—Capacity of Rectangular Tanks per 1 Foot of Length

Capacities given in U. S. gallons

Width of Tank	Length of Tank																								
	1'-0"	1'-6"	2-0"	2'-6"	3'-0"	3'-6"	4'-0"	4'-6"	5'-0"	5'-6"	6'-0"	6'-6"	7'-0"	7'-6"	8'-0"	8'-6"	9'-0"	9'-6"	10'-0"	10'-6"	11'-0"	12'-0"	13'-0"	14'-0"	15'-0"
ft. in.																									
1 0	7.48	11.22	14.96	18.70	22.44	26.18	29.92	33.66	37.40	41.14	44.88	48.62	52.36	56.10	59.84	63.58	67.32	71.06	74.81	78.55	82.29	89.77	97.25	104.73	112.21
1 6	....	16.83	22.41	28.05	33.66	39.27	44.88	50.49	56.10	61.71	67.32	72.93	78.55	84.16	89.77	95.38	100.99	106.60	112.21	117.82	123.43	134.65	145.87	157.09	168.31
2 0	....	....	29.92	37.40	44.88	52.36	59.84	67.32	74.81	82.29	89.77	97.25	104.73	112.21	119.69	127.17	134.65	142.13	149.61	157.09	164.57	179.53	194.49	209.45	222.41
2 6	....	....	....	46.75	56.10	65.45	74.80	84.16	93.51	102.86	112.21	121.56	130.91	140.26	149.61	158.96	168.31	177.66	187.01	196.36	205.71	224.41	243.12	261.82	280.52
3 0	....	....	....	....	67.32	78.54	89.77	100.99	112.21	123.43	134.65	145.87	157.09	168.31	179.53	190.75	202.97	213.19	224.41	235.63	246.86	269.30	291.74	314.18	336.62
3 6	....	....	....	....	....	91.64	104.73	117.82	130.91	144.00	157.09	170.18	183.27	196.36	209.45	222.54	235.63	248.73	261.82	274.90	288.00	314.18	340.36	366.54	392.73
4 0	....	....	....	....	....	....	119.69	134.65	149.61	164.57	179.53	194.49	209.45	224.41	239.37	254.34	269.30	284.26	299.22	314.18	329.14	359.06	388.99	418.91	448.83
4 6	....	....	....	....	....	....	....	151.48	168.31	185.14	201.97	218.80	235.62	252.47	269.30	286.13	302.96	319.79	336.62	353.45	370.28	403.94	437.61	471.27	504.93
5 0	....	....	....	....	....	....	....	....	187.01	205.71	224.41	243.11	261.82	280.52	299.22	317.92	336.62	355.32	374.03	392.72	411.43	448.83	486.23	523.64	561.04
5 6	....	....	....	....	....	....	....	....	....	226.28	246.86	267.43	288.00	308.57	329.14	349.71	370.28	390.85	411.43	432.00	452.57	493.71	534.86	576.00	617.14
6 0	....	....	....	....	....	....	....	....	....	....	269.30	291.74	314.18	336.62	359.06	381.50	403.94	426.39	448.83	471.27	493.71	538.59	583.48	628.36	673.25
6 6	....	....	....	....	....	....	....	....	....	....	....	316.05	340.36	364.67	388.98	413.30	437.60	461.92	486.23	510.54	534.85	583.47	632.10	680.73	729.35
7 0	....	....	....	....	....	....	....	....	....	....	....	....	366.54	392.72	418.91	445.09	471.27	497.45	523.64	549.81	575.99	628.36	680.73	733.09	785.45
7 6	....	....	....	....	....	....	....	....	....	....	....	....	....	420.78	448.83	476.88	504.93	532.98	561.04	589.08	617.14	673.24	729.35	785.45	841.56
8 0	....	....	....	....	....	....	....	....	....	....	....	....	....	....	478.75	508.67	538.59	568.51	598.44	628.36	658.28	718.12	777.97	837.82	897.66
8 6	....	....	....	....	....	....	....	....	....	....	....	....	....	....	....	540.46	572.25	604.05	635.84	667.63	699.42	763.00	826.60	890.18	953.76
9 0	....	....	....	....	....	....	....	....	....	....	....	....	....	....	....	....	605.92	639.58	673.25	706.90	740.56	807.89	875.52	942.54	1009.9
9 6	....	....	....	....	....	....	....	....	....	....	....	....	....	....	....	....	....	675.11	710.65	746.17	781.71	852.77	923.84	994.91	1066.0
10 0	....	....	....	....	....	....	....	....	....	....	....	....	....	....	....	....	....	....	748.05	785.45	822.86	897.66	972.47	1047.3	1122.1
10 6	....	....	....	....	....	....	....	....	....	....	....	....	....	....	....	....	....	....	....	824.73	864.00	942.56	1021.1	1099.6	1178.2
11 0	....	....	....	....	....	....	....	....	....	....	....	....	....	....	....	....	....	....	....	....	905.14	987.43	1069.7	1152.0	1234.3
12 0	....	....	....	....	....	....	....	....	....	....	....	....	....	....	....	....	....	....	....	....	....	1077.2	1166.0	1256.7	1346.5
13 0	....	....	....	....	....	....	....	....	....	....	....	....	....	....	....	....	....	....	....	....	....	....	1264.2	1361.5	1458.7
14 0	....	....	....	....	....	....	....	....	....	....	....	....	....	....	....	....	....	....	....	....	....	....	....	1466.2	1570.9
15 0	....	....	....	....	....	....	....	....	....	....	....	....	....	....	....	....	....	....	....	....	....	....	....	....	1683.1

MEASURES AND WEIGHTS COMMONLY USED IN THE UNITED STATES AND GREAT BRITAIN

Abbreviations

bbl.	= barrel	dwt.	= pennyweight	hr.	= hour	quin.	= quintal
bu.	= bushel	fath.	= fathom	Imp.	= Imperial (bu., gal.)	qr.	= quarter
chn.	= chain	ft.	= foot	in.	= inch	qt.	= quart
circm.	= circumference	fl. drm.	= fluid drachm	lea	= league	scr.	= scruple
cir. in.	= circular inch	fl. oz.	= fluid ounce	m. pace	= military pace	sec.	= second
cir. mil.	= circular mil	furl.	= furlong	min.	= minute	sq.	square (in., ft., etc.)
cwt.	= hundred weight	gal.	= gallon (U.S.)	naut. mi.	= nautical mile	stat. mi.	statute mile
deg.	= degree	grn.	= grain	pnch	= puncheon	trce.	= tierce
drm.	= drachm (dram)	hghd.	= hogshead	quad.	= quadrant	yd.	= yard

Table 19.—Measures of Length

Long Measure

12 in. = 1 ft.
3 ft. = 1 yd.
5 1/2 yd. = 16 1/2 ft. = 1 rod, pole or perch
40 poles = 220 yd. = 1 furl.
8 furl. = 1760 yd. = 5280 ft. = 1 mile
4 in. = 1 hand
9 in. = 1 span
2 1/2 ft. = 1 m. pace.

Land, Surveyors, or Gunter's Measure

7.92 in. = 1 link
100 links = 66 ft. = 4 rods = 1 chn.
10 chn. = 220 yd. = 1 furl.
8 furl. = 80 chn. = 1 mile.

Nautical Measure

6080.26 ft. = 1.15156 stat. mi. = 1 naut. mi.
3 naut. mi. = 1 lea.
60 naut. mi. = 1 deg. (at equator)
6 ft. = 1 fath.
120 fath. = 1 cable
1 naut. mi. per hr. = 1 knot.

Table 20.—Measures of Area

Square Measure

144 sq. in. = 183.35 cir. in. = 1 sq. ft.
9 sq. ft. = 1 sq. yd.
30 1/4 sq. yd. = 272 1/4 sq. ft. = 1 sq. rod
160 sq. rods
4840 sq. yd.
640 acres = 27,878,400 sq. ft. = 1 sq. mile
1 acre = $(208.71 \text{ ft.})^2$
1 cir. in. = area of circle 1 in. diam. = 0.7854 sq. in.
1 cir. mil = area of circle 0.001 in. diam. = $0.0_5 7854$ sq. in.
1,000,000 cir. mil = 1 cir. in.

Board Measure

144 cu. in. = 12 × 12 × 1 in. = 1 board ft.

The above definition of a board foot applies to rough green lumber. American practice uses a yard standard of 29/32 in. thick and an industrial standard of 7/8 in. thick for rough dry lumber; for dressed lumber the corresponding thicknesses are 25/32 in. and 13/16 in. For example, a rough dry piece 12 × 12 × 29/32 in. contains 1 board ft. Board ft. in round timber = $(1/4\, c \times d \times l)/144$. c = mean circm., d = diam., l = length, all in ft.

Table 21.—Measures of Weight

(The grain is the same in all systems.)

Avoirdupois Weight

16 drm. = 437.5 grn. = 1 oz.
16 oz. = 7000 grn. = 1 lb.
14 lb. = 1 stone
28 lb. = 2 stone = 1 qr.
100 lb. = 1 quin.
4 qr. = 112 lb. = 1 cwt.
20 cwt. = 2240 lb. = 1 long ton
2000 lb. = 1 short ton.

Troy Weight (for gold and silver)

24 grn. = 1 dwt.
20 dwt. = 480 grn. = 1 oz.
12 oz. = 5760 grn. = 1 lb.
1 Assay ton = 29,167 milligrams = Troy oz. per 2000 lb. ton avoirdupois

Carat Weight (for precious stones)

1 carat = 3.086 grn. = 0.200 gram

Apothecaries' Weight

20 grn. = 1 scr. ℈
3 scr. = 60 grn. = 1 drm. ʒ
8 drm. = 480 grn. = 1 oz. ℥
12 oz. = 5760 grn. = 1 lb.

Table 22.—Measures of Volume

Cubic Measure

1728 cu. in. = 1 cu. ft.
27 cu. ft. = 1 cu. yd.
4 × 4 × 8 ft. = 128 cu.

Liquid Measure

4 gills = 1 pint
2 pints = 1 qt.
4 qt. = 231 cu. in. = 8.3356 lb. H_2O = 1 U. S. gal. = 0.8327 Imp. gal.
1 Imp. gal. = 277.420 cu. in. = 10 lb. H_2O
1.20094 U. S. gal. = 1 Imp. gal.
7.4805 U. S. gal. = 1 cu. ft.

Old Liquid Measure

31 1/2 gal. = 1 bbl.
42 gal. = 1 trce.
2 bbl. = 63 gal. = 1 hghd.
84 gal. = 2 trce. = 1 pchn.
2 hghd. = 126 gal. = 1 pipe or butt
2 pipes = 3 pchn. = 1 tun.

Apothecaries' Fluid Measure

60 minims = 1 fl. drm.
8 drm. = 1 fl. oz.
1 fl. oz. = 1/128 U. S. gal. = 1.8047 cu. in. = 456.3 grn. H_2O at 39° F.
16 fl. oz. = 1 pint
1 fl. oz. (British) = 1.732 cu. in. = 437.5 grn. H_2O at 62° F.

Dry Measure

2 pints = 1 qt.
8 qts. = 1 peck
4 pecks = 2150.42 cu. in. = 1 bu. (struck)
1 1/4 bu. (struck) = 1 bu. (heaped)
105 qts. (struck) 7056 cu. in. = 1 bbl.
8 Imp. gal. 2218.192 cu. in. = 1 Imp. bu.
8 Imp. bu. = 1 qr.

Shipping Measure

100 cu. ft. = 1 Register ton *
40 cu. ft. = 31.143 U. S. bu. = 1 shipping ton (U. S.) †
42 cu. ft. = 32.719 Imp. bu. = 1 shipping ton (British) †

* For measurement of entire internal capacity † For measurement of cargo.

Miner's Inch

A variable unit, defined by statute in different states as the quantity of water that will flow through an orifice of 1 sq. in. area, under a head ranging from 4 1/2 to 6 in. U. S. Reclamation Service defines the miner's inch as 1 cu. in. per sec. The law governing the locality in which the measurement is to be made should be consulted.

Table 23.—Circular Measure

60 sec.	=	1 min.
60 min.	=	1 deg.
90 deg.	=	1 quad.
360 deg.	=	circm.
57.2957795 deg.	=	1 radian
	=	57° 17′ 44.806″
1 radian	=	angle of arc equal to radius.

Table 24.—Time

60 sec. = 1 min.		7 days = 1 week	
60 min. = 1 hr.		52 weeks = 1 year	
24 hr. = 1 day			

1 year (exact) = 365 days, 5 hr., 48 min., 45.6 sec.

A year divisible by 4 is a leap year and contains 366 days, except centesimal years, of which only those divisible by 400 are leap years.

365.24734 mean solar days
= 366.008515 sidereal days
24 hr. mean solar time
= 23 hr. 56 min., 49.9 sec. sidereal time
1 mean solar day
= 1 mean sidereal day + 3 min., 10.1 sec.

THE METRIC SYSTEM

The fundamental standard of length in the metric system is the International Standard Meter, derived from the Metre des Archives, Paris. Its length is defined as the distance between two lines at 0° C. on a platinum-iridium bar deposited at the International Bureau of Weights and Measures near Paris. 1 meter = 39.37 in.

The fundamental standard of mass is the International Standard Kilogram, which is a mass of platinum-iridium, deposited at the International Bureau, whose weight *in vacuo* is the same as that of the Kilogram des Archives.

The fundamental unit of volume is the liter, which is defined as the volume of a kilogram of water at the temperature of maximum density, 4° C., under a pressure of 76 cm. of mercury. The liter originally was defined as a cubic decimeter, but is slightly greater. The generally accepted relation between the cubic decimeter and the liter is 1 liter = 1.000027 cu. decimeters.

The unit of force is the **dyne**, which is the force which acting for 1 second on a mass of one gram will produce a velocity of 1 centimeter per second. 1 dyne = (1/980.665) gram. 1,000,000 dynes = 1.020 kilograms force = 2.248 lb. force. 1 lb. force = 0.4536 kilogram force = 448,800 dynes.

The unit of heat is the **gram-calorie**, which is the quantity of heat required to raise 1 gram of water 1° C. See p. 3-17.

Table 25.—Units of Metric Measures*

Length			Area		
Unit	Symbol	Value in Meters	Unit	Symbol	Value in Sq. Meters
Micron	μ	0.000 001			
Millimeter	mm.	0.001	Sq. millimeter	sq. mm.	0.000 001
Centimeter	cm.	0.01	Sq. centimeter	sq. cm.	0.000 1
Decimeter	dm.	0.1	Sq. decimeter	sq. dm.	0.01
Meter (unit)	m.	1.0	Sq. meter (centiare)	sq. m.	1.0
Dekameter	dkm.	10.0	Sq. dekameter (are)	sq. dkm.	100.0
Hectometer	hm.	100.0	Hectare	ha.	10,000.0
Kilometer	km.	1,000.0	Sq. kilometer	sq. km.	1,000,000.0
Myriameter	Mm.	10,000.0			
Megameter	....	1,000,000.0			

Cubic Measure			Volume		
Unit	Symbol	Value in Cubic Meters	Unit	Symbol	Value in Liters
Cubic kilometer	cu. km.	1,000,000,000	Milliliter	ml.	0.001
Cubic hectometer	cu. hm.	1,000,000	Centiliter	cl.	0.01
Cubic dekameter	cu. dkm.	1,000	Deciliter	dl.	0.1
Cubic meter	cu. m.	1	Liter (unit)	l.	1.0
Cubic decimeter	cu. dm.	0.001	Dekaliter	dkl.	10.0
Cubic centimeter	cu. cm.	0.000 001	Hectoliter	hl.	100.0
Cubic millimeter	cu. mm.	0.000 000 001	Kiloliter	kl.	1,000.0
Cubic micron	μ^3	$0.0_{17}1$			

Weight					
Unit	Symbol	Value in Grams	Unit	Symbol	Value in Grams
Microgram	γ	0.000 001	Gram (unit)	g.	1.0
Milligram	mg.	0.001	Dekagram	dkg.	10.0
Centigram	cg.	0.01	Hectogram	hg.	100.0
Decigram	dg.	0.1	Kilogram	kg.	1,000.0
Gram (unit)	g.	1.0	Myriagram	Mg.	10,000.0
			Quintal	q.	100,000.0
			Ton	t.	1,000,000.0

* A subscript after a figure indicates the number of times it is repeated. Thus 0.0_38 = 0.0008

Table 26.—Metric Equivalents of Length *

English to Metric			Log.	Metric to English			Log.
1 inch =	25.4	mm.	1.404834	1 mm. =	0.03937	in.	$\bar{2}.595165$
=	2.54	cm.	0.404834	=	0.003281	ft.	$\bar{3}.515984$
=	0.0254	m.	$\bar{2}.404834$	=	0.001094	yd.	$\bar{3}.038863$
1 foot =	304.800	mm.	2.484015	1 cm. =	0.3937	in.	$\bar{1}.595165$
=	30.480	cm.	1.484015	=	0.03281	ft.	$\bar{2}.515984$
=	0.3048	m.	$\bar{1}.484015$	=	0.01094	yd.	$\bar{2}.038863$
1 yard =	91.4402	cm.	1.961137	1 meter =	39.37	in.	1.595162
=	0.9144	m.	$\bar{1}.961137$	=	3.2808	ft.	0.515984
=	0.0_3914	km.	$\bar{4}.961137$	=	1.0936	yd.	0.038863
1 rod =	502.9211	cm.	2.701500	=	0.1988	rd.	$\bar{1}.298500$
=	5.0292	m.	0.701500	=	0.04971	chain	$\bar{2}.696440$
=	0.005029	km.	$\bar{3}.701500$	=	0.0_36214	mi.	$\bar{4}.793350$
1 chain =	2011.68	cm.	3.303559	1 kilometer =	3280.833	ft.	3.515984
=	20.1168	m.	1.303559	=	1093.611	yd.	3.038863
=	0.02012	km.	$\bar{2}.303559$	=	198.838	rods	2.298500
1 mile =	1609.344	m.	3.206650	=	49.7095	chains	1.696440
=	1.6093	km.	0.206650	=	0.6214	mi.	$\bar{1}.793350$

Table 27.—Metric Equivalents of Area *

English to Metric			Log.	Metric to English			Log.
1 cir. mil =	0.0_35067	sq. mm.	$\bar{4}.704751$	1 sq. mm. =	1,973.55	cir. mils	3.295249
1 sq. in. =	645.163	sq. mm.	2.809669	=	0.001550	sq. in.	$\bar{3}.190331$
=	6.4516	sq. cm.	0.809669	=	0.0_410764	sq. ft.	$\bar{5}.031968$
=	0.0_36452	sq. m.	$\bar{4}.809669$	=	0.0_51196	sq. yd.	$\bar{6}.077726$
1 sq. ft. =	92,903.41	sq. mm.	4.968032	1 sq. cm. =	0.1550	sq. in.	$\bar{1}.190331$
=	929.0341	sq. cm.	2.968032	=	0.001076	sq. ft.	$\bar{3}.031968$
=	0.0929	sq. m.	$\bar{2}.968032$	=	0.0_31196	sq. yd.	$\bar{4}.077726$
=	0.0_5929	hectare	$\bar{6}.968032$	1 sq. m. =	1,549.9969	sq. in.	3.190331
=	0.0_7929	sq. km.	$\bar{8}.968032$	=	10.7639	sq. ft.	1.031968
1 sq. yd. =	836,130.74	sq. mm.	5.922274	=	1.1960	sq. yd.	0.077726
=	8,361.307	sq. cm.	3.922274	=	0.002471	sq. chain	$\bar{3}.392882$
=	0.83613	sq. m.	$\bar{1}.922274$	=	0.0_32471	acre	$\bar{4}.392882$
=	0.0_4836	hectare	$\bar{5}.922274$	=	0.0_63861	sq. mi.	$\bar{7}.586700$
=	0.0_6836	sq. km.	$\bar{7}.922274$	1 hectare =	107,638.7	sq. ft.	5.031968
1 sq. chain =	404.686	sq. m.	2.607118	=	11,959.85	sq. yd.	4.077726
=	0.04047	hectare	$\bar{2}.607118$	=	24.710	sq. chain	1.392882
=	0.0_34047	sq. km.	$\bar{4}.607118$	=	2.4710	acres	0.392882
1 acre =	4,046.86	sq. m.	3.607118	=	0.003861	sq. mi.	$\bar{3}.586700$
=	0.4047	hectare	$\bar{1}.607118$	1 sq. km. =	10,763,867.38	sq. ft.	7.031968
=	0.004047	sq. km.	$\bar{3}.607118$	=	1,195,985.26	sq. yd.	6.077726
1 sq. mile =	2,589,998	sq. m.	6.413300	=	2,471.050	sq. chains	3.392882
=	259.0	hectare	2.413300	=	247.1045	acres	2.392882
=	2.590	sq. km.	0.413300	=	0.3861	sq. mi.	$\bar{1}.586700$

Table 28.—Metric Equivalents of Weight or Mass *

English to Metric			Log.	Metric to English			Log.
1 grain =	64.797	mg.	1.811568	1 milligram =	0.01543	grain	$\bar{2}.188433$
=	0.0648	gm.	$\bar{2}.811568$	=	0.0_43215	oz. Troy	$\bar{5}.507191$
1 oz. Troy or				=	0.0_43527	oz. avoir.	$\bar{5}.547454$
apothecary =	31,103.5	mg.	4.492809	1 gram =	15.4324	grains	1.188433
=	31.103	gm.	1.492809	=	0.03215	oz. Troy	$\bar{2}.507191$
=	0.03110	kg.	$\bar{2}.492809$	=	0.03527	oz. avoir.	$\bar{2}.547454$
1 oz. avoirdupois =	28,349.5	mg.	4.452546	=	0.0_22679	lb. Troy	$\bar{3}.428010$
=	28.3495	gm.	1.452546	=	0.0_22205	lb. avoir.	$\bar{3}.343334$
=	0.02835	kg.	$\bar{2}.452546$	1 kilogram =	32.1508	oz. Troy	1.507191
1 pound Troy or				=	35.2740	oz. avoir.	1.547454
apothecary =	373.2417	gm.	2.571990	=	2.6792	lb. Troy	0.428010
=	0.3732	kg.	$\bar{1}.571990$	=	2.2046	lb. avoir.	0.343334
1 pound avoirdupois =	453.5925	gm.	2.656666	=	0.0_21102	short ton	$\bar{3}.043304$
=	0.4536	kg.	$\bar{1}.656666$	=	0.0_39842	long ton	$\bar{4}.993086$
=	0.0_34536	metric ton	$\bar{4}.656666$	1 metric ton =	2204.62	lb. avoir.	3.343334
1 short ton =	907.2	kg.	2.957696	=	1.1023	short tons	0.043304
=	0.9072	metric ton	$\bar{1}.957696$	=	0.9842	long ton	$\bar{1}.993086$
1 long ton =	1,016.06	kg.	3.006914				
=	1.0161	metric ton	0.006914				

* A subscript after a figure indicates the number of times it is repeated. Thus $0.0_38 = 0.0008$

Table 29.—Metric Equivalents of Capacity or Volume*

English to Metric			Log.	Metric to English			Log.
1 cu. in. =	16,387.17	cu. mm.	4.214504	1 cu. mm. =	0.0$_4$6102	cu. in.	$\bar{5}$.785496
=	16.3872	cu. cm.	1.214504	=	0.0$_3$2705	fluid dr.	$\bar{4}$.432185
=	0.016387	l.	$\bar{2}$.214504	=	0.0$_4$3381	fluid oz.	$\bar{5}$.529094
=	0.0$_3$16387	hl.	$\bar{4}$.214504	1 cu. cm. =	0.06102	cu. in.	$\bar{2}$.785496
=	0.0$_4$16387	cu. m.	$\bar{5}$.214504	=	0.0$_4$3531	cu. ft.	$\bar{5}$.547951
1 cu. ft. =	28,317.08	cu. cm.	4.452049	=	0.0$_5$1308	cu. yd.	$\bar{6}$.116589
=	28.3169	l.	1.452049	=	0.0$_4$2838	bushel	$\bar{5}$.452972
=	0.2832	hl.	$\bar{1}$.452049	=	0.2705	fluid dr.	$\bar{1}$.432185
=	0.02832	cu. m.	$\bar{2}$.452049	=	0.03381	fluid oz.	$\bar{2}$.529094
1 cu. yd. =	764,559.5	cu. cm.	5.883411	=	0.001057	quart	$\bar{3}$.023944
=	764.5595	l.	2.883411	=	0.0$_3$2642	gallon	$\bar{4}$.421884
=	7.6456	hl.	0.883411	1 liter =	61.02398	cu. in.	1.785496
=	0.7646	cu. m.	$\bar{1}$.883411	=	0.035313	cu. ft.	$\bar{2}$.547951
1 bushel =	35,239.3	cu. cm.	4.547027	=	0.0013079	cu. yd.	$\bar{3}$.116589
=	35.2393	l.	1.547027	=	0.028377	bushel	$\bar{2}$.452972
=	0.3524	hl.	$\bar{1}$.547027	=	1.0567	quart	0.023944
=	0.03524	cu. m.	$\bar{2}$.547027	=	0.2642	gallon	$\bar{1}$.421884
1 fluid drachm =	3,696.7	cu. mm.	3.567815	1 hectoliter =	6,102.398	cu. in.	3.785496
=	3.6967	cu. cm.	0.567815	=	3.5313	cu. ft.	0.547951
1 fluid oz. =	29,573.7	cu. mm.	4.470906	=	0.13079	cu. yd.	$\bar{1}$.116589
=	29.5737	cu. cm.	1.470906	=	2.8377	bushels	0.452972
1 quart =	946.359	cu. cm.	2.976056	1 cu. meter =	61,023.38	cu. in.	4.785496
=	0.9464	l.	$\bar{1}$.976056	=	35.3133	cu. ft.	1.547951
=	0.0$_3$946	cu. m.	$\bar{4}$.976056	=	1.3079	cu. yd.	0.116589
1 U. S. gallon =	3,785.43	cu. cm.	3.578116	=	28.3773	bushels	1.452972
=	3.7854	l.	0.578116	=	1,056.682	quarts	3.023944
=	0.003785	cu. m.	$\bar{3}$.578116	=	264.170	gallons	2.421884

Table 30.—Metric Equivalents of Density*

English to Metric	Log.	Metric to English	Log.
1 lb. per cu. in. = 27.680 gm. per cu. cm.	1.442162	1 gm. per cu. cm. = 0.03613 lb. per cu. in.	2.557838
= 27,679.7 kg. per cu. m.	4.442162	= 62.430 lb. per cu. ft.	1.795381
1 lb. per cu. ft. = 0.01602 gm. per cu. cm.	$\bar{2}$.204619	= 8.3454 lb. per U. S. gal.	0.921450
= 16.0183 kg. per cu. m.	1.204619	1 kg. per cu. m. = 0.0$_4$3613 lb. per cu. in.	$\bar{5}$.557838
= 0.01602 metric ton per cu. m.	$\bar{2}$.204619	= 0.062430 lb. per cu. ft.	$\bar{2}$.795381
1 lb. per cu. yd. = 0.5933 kg. per cu. m.	$\bar{1}$.773254	= 1.6856 lb. per cu. yd.	0.226746
= 0.0$_3$5933 metric tons per cu. m.	$\bar{4}$.773254	= 0.0$_2$8345 lb. per U. S. gal.	$\bar{3}$.921450
1 lb. per U. S. gal. = 0.1198 gm. per cu. cm.	$\bar{1}$.078550	1 metric ton per cu. m = 62.4286 lb. per cu. ft.	1.795381
= 119.826 kg. per cu. m.	2.078550	= 1685.487 lb. per cu. yd.	3.226746
1 short ton per cu. yd. = 1.1865 metric tons per cu. m.	0.074285	= 0.8428 short ton per cu. yd.	$\bar{1}$.925715
1 long ton per cu. yd. = 1.3289 metric tons per cu. m.	0.123502	= 0.7525 long ton per cu. yd.	$\bar{1}$.876498

Table 31.—Metric Equivalents of Velocity*

English to Metric	Log.	Metric to English	Log.
1 in. per sec. = 2.54001 cm. per sec.	0.404834	1 cm. per sec. = 0.3937 in. per sec.	$\bar{1}$.595165
= 0.0254 m. per sec.	$\bar{2}$.404834	= 0.03281 ft. per sec.	$\bar{2}$.515984
= 1.5240 m. per min.	0.182985	= 1.9685 ft. per min.	0.294136
1 ft. per sec. = 30.4801 cm. per sec.	1.484015	= 0.02237 mi. per hr.	$\bar{2}$.349653
= 0.3048 m. per sec.	$\bar{1}$.484015	= 0.01943 knot	$\bar{2}$.288367
= 18.2880 m. per min.	1.262166	1 m. per sec. = 39.37 in. per sec.	1.595165
= 1.0973 km. per hr.	0.040318	= 3.2808 ft. per sec.	0.515984
1 ft. per min. = 0.5080 cm. per sec.	$\bar{1}$.705864	= 196.8500 ft. per min.	2.294136
= 0.00508 m. per sec.	$\bar{3}$.705864	= 2.2369 mi. per hr.	0.349653
= 0.3048 m. per min.	$\bar{1}$.484015	= 1.9426 knots	0.288367
= 0.018288 km. per hr.	$\bar{2}$.262166	1 m. per min. = 0.6562 in. per sec.	$\bar{1}$.817014
1 mile per hr. = 44.704 cm. per sec.	1.650348	= 0.05468 ft. per sec.	$\bar{2}$.737833
= 0.4470 m. per sec.	$\bar{1}$.650348	= 3.2808 ft. per min.	0.515984
= 26.8222 m. per min.	1.428495	= 0.03728 mi. per hr.	$\bar{2}$.571503
= 1.6093 km. per hr.	0.206650	= 0.03238 knot	$\bar{2}$.510218
1 knot = 51.4791 cm. per sec.	1.711631	1 km. per hr. = 0.9113 ft. per sec.	$\bar{1}$.959681
= 0.5148 m. per sec.	$\bar{1}$.711631	= 54.6806 ft. per min.	1.737833
= 30.8875 m. per min.	1.489782	= 0.62138 mi. per hr.	$\bar{1}$.793357
= 1.8532 km. per hr.	0.267933	= 0.5396 knot	$\bar{1}$.732072

* A subscript after a figure indicates the number of times it is repeated. Thus 0.0$_3$8 = 0.0008

Table 32.—Metric Equivalents of Pressure*

English to Metric	Log.	Metric to English	Log.
1 lb. per sq. in. = 0.7031 gm. per sq. mm.	$\bar{1}.846996$	1 gm. per sq. mm. = 1.4223 lb. per sq. in.	0.153004
= 70.3066 gm. per sq. cm.	2.846996	= 204.8170 lb. per sq. ft.	2.311366
= 0.07031 kg. per sq. cm.		1 gm. per sq. cm. = 0.01422 lb. per sq. in.	$\bar{2}.153004$
= 0.7031 metric ton per sq. m.		= 2.0482 lb. per sq. ft.	0.311366
= 0.07031 metric atmosphere†		1 kg. per sq. cm. = 14.2234 lb. per sq. in.	1.153004
1 lb. per sq. ft. = 0.004882 gm. per sq. mm.		= 2048.1696 lb. per sq. ft.	3.311366
= 0.4882 gm. per sq. cm.	$\bar{1}.688598$	1 metric ton per sq. m. = 1.4223 lb. per sq. in.	0.153004
= 0.0_34882 kg. per sq. cm.	$\bar{4}.688598$	204.8170 lb. per sq. ft.	2.311366
= 0.004882 metric ton per sq. m.	$\bar{3}.688598$	0.1024 ton per sq. ft.§	$\bar{1}.010342$
= 0.0_34882 metric atmosphere†		1 metric atmosphere† = 14.2234 lb. per sq. in.	1.153004
1 ton per sq. ft.† = 9.7648 metric tons per sq. m.	0.989663	2048.1696 lb. per sq. ft.	3.311366
= 0.9765 metric atmosphere†	$\bar{1}.989663$	1.0241 tons per sq. ft.	0.010342
1 atmosphere ‡ = 1.0335 metric atmospheres †	0.014310	0.9676 atmosphere ‡	$\bar{1}.985696$

† 1 metric atmosphere = 1 kg. per sq. cm. ‡ 1 atmosphere = 14.7 lb. per sq. in. § 1 ton = 2000 lb.

Table 33.—Metric Equivalents of Force*

English to Metric	Log.	Metric to English	Log.
1 lb. = 444,800 dynes	5.648165	1 dyne = 0.0_52248 lb.	$\bar{6}.351796$
= 453.6 grams	2.656673	= 0.0_47233 poundal	$\bar{5}.859318$
= 0.04448 joule per cm.	$\bar{2}.648165$	= 0.0_21020 gram	$\bar{3}.008600$
= 0.4536 kg.	$\bar{1}.656673$	= 0.0_51020 kg.	$\bar{6}.008600$
= 32.17 poundals	1.507451	= 0.0_61 joule per cm.	$\bar{7}.000000$
1 poundal = 13,826 dynes	4.140696	1 gram = 0.0_22205 lb.	$\bar{3}.343409$
= 14.10 grams	1.149219	= 0.07093 poundal	$\bar{2}.850830$
= 0.0_213826 joule per cm.	$\bar{3}.140696$	= 980.7 dynes	2.991536
= 0.01410 kg.	$\bar{2}.149219$	= 0.001 kg.	$\bar{3}.000000$
= 0.03108 lb.	$\bar{2}.492481$	= 0.0_49807 joule per cm.	$\bar{5}.991536$
		1 joule per cm. = 22.48 lb.	1.351796
		= 723.3 poundals	2.859318
		= 10,000,000 dynes	7.000000
		= 10,200 grams	4.008600
		= 10.20 kg.	1.008600
		1 kg. = 2.205 lb.	0.343409
		= 70.93 poundals	1.850830
		= 980,700 dynes	5.991536
		= 1000 grams	3.000000
		= 0.09807 joule per cm.	2.991536

Table 34.—Metric Equivalents of Power, Work and Energy*

English to Metric	Log.	Metric to English	Log.
1 ft.-lb. = 0.1383 kg.-m.	$\bar{1}.140683$	1 kilogram-meter = 7.2331 ft.-lb.	0.859318
= 0.0_63766 kw.-hr.	$\bar{7}.575880$	= 0.0_53653 Hp.-hr.	$\bar{6}.562651$
= 0.0_65121 cheval-vapeur-hr.‡	$\bar{7}.709698$	= 0.0_29303 B.t.u.	$\bar{3}.968622$
= 0.0_33241 kg.-calorie	$\bar{4}.510660$	1 kilogram-meter per sec. = 7.2331 ft.-lb. per sec.	0.859318
1 ft.-lb. per sec. = 0.1383 kg.-m. per sec.	$\bar{1}.140683$	= 0.01315 Hp.	$\bar{2}.118929$
= 0.0_21383 poncelet †	$\bar{3}.140683$	1 kilogram-meter per min. = 7.2331 ft.-lb. per min.	0.859318
= 0.0_21843 cheval-vapeur ‡	$\bar{3}.265619$	= 0.0_32192 Hp.	$\bar{4}.340809$
= 0.0_33241 kg.-cal. per sec.	$\bar{4}.510660$	1 kilowatt-hour = 2,655,180 ft.-lb.	6.424082
1 ft.-lb. per min. = 0.1383 kg.-m. per min.	$\bar{1}.140683$	= 1.3341 Hp.-hr.	0.125189
= 0.0_21383 poncelet †	$\bar{3}.140683$	= 3411.5 B.t.u.	3.532946
= 0.0_43072 cheval-vapeur ‡	$\bar{5}.487469$	1 cheval-vapeur ‡ = 542.48 ft.-lb. per sec.	2.734384
1 Hp. = 76.0398 kg.-m. per sec.	1.881040	= 32,548.57 ft.-lb. per min.	4.512532
= 4562.384 kg.-m. per min.	3.659191	= 0.9863 Hp.	$\bar{1}.994018$
= 0.7604 poncelet †	$\bar{1}.881040$	1 cheval-vapeur-hour = 1,952,914 ft.-lb.	6.290683
= 1.01390 cheval-vapeur ‡	0.016616	= 2513.56 B.t.u.	3.400294
1 Hp.-hr. = 273,742.9 kg.-m.	5.437436	1 Poncelet † = 723.31 ft.-lb. per sec.	2.859318
= 0.7456 kw.-hr.	$\bar{1}.872516$	= 43,398.10 ft.-lb. per min.	4.637471
= 641.6895 kg.-cal.	2.807325	= 1.3151 Hp.	0.118929
1 B.T.U. = 0.2520 kg.-cal.	$\bar{1}.401401$	1 kilogram-calorie = 3085.60 ft.-lb.	3.489360
= 107.4925 kg.-m.	2.031123	= 0.0_21558 Hp.-hr.	$\bar{3}.192675$
= 0.0_32931 kw.-hr.	$\bar{4}.467016$	= 3.9686 B.t.u.	0.598637
= 0.0_33978 cheval-vapeur-hr.‡	$\bar{4}.599711$	1 kilogram-calorie per sec. = 3085.60 ft.-lb. per sec.	3.489360
		= 5.610 Hp.	0.748982

* A subscript after a figure indicates the number of times it is repeated. Thus $0.0_38 = 0.0008$.
† 1 poncelet = 100 kg.-m. per sec. ‡ 1 cheval-vapeur = 75 kg.-m. per sec. = 1 metric horsepower.

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HANDBOOK OF ENGINEERING FUNDAMENTALS

Edited by Ovid W. Eshbach

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